Eleventh International Technical Conference on Experimental Safety Vehicles

11

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of Transportation

**National Highway
Traffic Safety
Administration**

United States

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**National Highway
Traffic Safety
Administration**

Foreword

This report of the proceedings of the Eleventh International Technical Conference on Experimental Safety Vehicles was prepared by the National Highway Traffic Safety Administration, U.S. Department of Transportation.

We wish to thank the authors and all those responsible for the excellence of the material submitted, which aided materially in the preparation of this report.

For clarity and because of some translation difficulties, a certain amount of editing was necessary. Apologies are, therefore, offered where the transcription is not exact.

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Contents

CONTENTS

SECTION 4: TECHNICAL SESSIONS
Chairman: Ian D. Neilson, United Kingdom

Technical Session One: Occupant Protection for Side Impact

CONTENTS

CONTENTS

CONTENTS

CONTENTS

CONTENTS

Section 1
Opening Ceremonies

Welcoming Address:

Diane K. Steed,
Administrator,
National Highway Traffic Safety
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I am honored to welcome you to the eleventh annual international conference on experimental safety vehicles—the first held in this country since 1976, our bicentennial year.

Over the past two decades—thanks in no small part to the efforts of many here today—we have seen significant technological advances in automotive safety. Many of the safety improvements that are on today's vehicles were made possible through the research of the governments, manufacturers and suppliers, and individuals represented here today.

At the National Highway Traffic Safety Administration we take our mandate to save lives and reduce injuries very seriously. In one sense, we are trying to save the same life in as many ways as possible. Let me take a few moments to explain what I mean. Picture for a moment, a hypothetical traffic fatality.

It is night. It is raining. The crash takes place on a rural road with poor alignment. The driver is a young salesman returning home from a dinner meeting where he'd been drinking. He is driving too fast, in a vehicle with worn tires. The driver brakes on a turn and the car skids out of control. It crashes into a tree and the driver, who wasn't wearing his safety belt, smashes into the steering column and ultimately is ejected from the car as it rolls over. The police arrive on the scene and they call an ambulance. The driver is taken to the nearest hospital where he dies the next morning of internal injuries.

The outcome is all too familiar. But what is the cause of death? There is no single answer, although a lot of people will try to identify one. The police officer might cite speed or drinking on his report as the cause. The brake engineer might wonder if a car with antilock brakes would have skidded. The biomechanics expert could blame the fatality on the failure to develop the next generation of collapsible steering columns because there is still not enough known about soft tissue impact tolerance. The state motor vehicle inspector might blame worn tires. The truth is, we in highway safety need to be concerned about *all* factors that contributed.

We believe that effective intervention in one or more of the areas I just cited on that long list of contributing factors can make the difference in preventing the crash fatality I just described.

Ours must be a broad focus, a balanced program that considers equipment as but one area of improvement in our continuing quest to save lives. Nevertheless, safety research into design and equipment improvements is and will remain a vital part of our program.

At earlier conferences, we have considered the value of such safety devices as air bags and automatic safety belts, windshield glazing to protect against facial lacerations, high-mounted rear brake lights to reduce the chance of rear-end collisions, and anti-lock braking systems to enhance vehicle control on wet roads.

We have also worked to harmonize safety regulations with other countries to see that safety innovations are more readily available to all.

To further *that* safety goal, I announced at Oxford my intention to form a Motor Vehicle Safety Research Advisory Committee. Today, that intent is a reality and appointments to the Committee will be announced over the next few months. This Committee will represent a unique opportunity for research people from government and private industry. Through this program we will be able to avoid duplication of effort and foster greater harmonization.

In just five years we have already seen the benefits of efforts to work together in the use of common symbols on vehicle control displays and recent changes in lighting standards, and we are close—very close—in our efforts to harmonize braking standards for passenger cars.

By working together to eliminate conflicting regulations that make it burdensome for manufacturers with customers in both foreign and domestic markets, we are helping to remove the non-tariff trade barriers that restrict international commerce. Harmonization also helps reduce costs to consumers and the industry. We are also able to promote safety by upgrading standards and encouraging the sharing of safety technology.

We've entered a new era in highway safety, for this is the year when automatic crash protection became more than regulatory rhetoric.

How times have changed!

Back in 1956 the Ford Motor Company introduced a safety package that included a deep-dish steering wheel, padded dash, and safety belts. Unfortunately that car was surpassed by a competitor and, didn't set any sales records, giving rise to the notion that "safety doesn't sell."

Over the years, that notion became the prevailing wisdom. As the Chairman of General Motors, Roger Smith, said recently, "Back in the early '70's, GM was the first auto manufacturer to design, build, and sell an air-cushion restraint system—we were the world leader—but we may have been a little ahead of our customers."

"But," he added, "people's tastes change."

And, as we begin four days of what I know will be a positive exchange of information, it is obvious that he is right.

On my recent trip to Japan and South Korea, I was pleased to see a number of safety regulatory programs underway and to learn that nearly all the Japanese companies are hard at work on air bags. Honda will offer driver side air bags on the Acura Legend beginning next month and other companies have plans to introduce this technology in the relatively near future.

In the parking lot across the street from this hotel are 18 vehicles that demonstrate what automakers around the world are offering in the way of automatic crash protection. At 12:30 some of you joined with people from around the world in viewing an air bag demonstration.

Just one of a number of interesting discussions and demonstrations you will experience this week.

Your work is of great benefit. We eagerly anticipate the reports, the panels, and the private conversations and exchanges of information that lie ahead.

Now, I am proud to have the opportunity to present someone who has made transportation safety a national and international issue, a lady who has made a positive difference in all our lives, Elizabeth Hanford Dole.

Keynote Address

The Honorable Elizabeth Hanford Dole,
Secretary of Transportation,
United States

I'm delighted to participate in the opening of this 11th International Technical Conference on experimental safety vehicles. On behalf of the American delegation, I bid you welcome to Washington. I'm confident that this will be a worthwhile and informative conference.

Since the first conference in 1970, tremendous strides have been made in the technology of motor vehicle safety—advances toward which all nations have contributed, and from which every nation has benefitted.

The nations which participate in this conference share a common goal—to increase automobile safety for the traveling public. Since the automobile is the preferred means of transportation throughout the industrialized world, developing ever safer motor vehicles is of paramount importance to our nations. Motor vehicle deaths transcend national boundaries; no nation is immune from the tragedies of deaths and crippling injuries due to automobile crashes.

In fact, we inhabit a world where national boundaries are no longer seen as natural barriers, to culture or commerce. And so we at DOT are deeply involved in a worldwide campaign called Harmonization. We seek to harmonize American vehicle safety standards with those in other lands—not only to lower consumer costs but also to enhance our ability to compete in foreign markets and to ensure that we devote every ounce of strength we have to remove all foreign trade barriers to our products. Even now, we are looking for areas where we can coordinate with European governments the adjustment of standards. We've already succeeded in putting common symbols on vehicle control displays; now we are looking to make similar rules for passenger cars and brakes.

That's just the beginning of changes I see on the American road. Even more striking examples come to mind. Who would have thought that the American public would so readily accept mandatory child safety seat laws? The seed was planted in the state of Tennessee in 1978 with the first of such safety seat laws. Today, every state in the union, along with the District of Columbia, has similar legislation on the books. Usage rates for children under five has jumped from just 15 percent in 1979 to 75.8 percent last year. And our studies show that child seats, when properly installed, reduce the risk of death or serious injury for young children by about 70 percent.

Many European countries have had a long and successful history of mandatory safety belt laws, but it took a great deal of effort to get the ball rolling here. Indeed, the biggest single challenge I faced when arriving at the Department four years ago was to review and settle, once and for all, the U.S. regulation on automatic crash protection for passenger car occupants. Our decision in 1984 has contributed much to a nationwide awakening on occupant protection. Twenty-seven states and the District of Columbia have now passed safety belt laws. And last fall, we officially entered not just another model year, but also a new era in highway safety—the year when automatic crash protection became more than just regulatory rhetoric. This year, the auto industry is manufacturing some one million cars which will offer either automatic safety belts or air bags. And by 1990, automatic protection will be standard equipment in all new passenger cars unless states representing two-thirds of the population of the United States have enacted effective mandatory seat belt use laws. While I don't have to tell you the significance of this safety milestone, I can't help but note that for the first time in the long 15-year history of this rule, we're looking at reality—not just a prototype of the future—and lives are being saved.

Who would have thought, just a few short years ago, that one would open *Time* or *Newsweek* and find two-page advertising spreads touting auto safety initiatives? A nationwide NHTSA survey found over three-quarters of Americans favoring safety belt laws for the driver and front seat passenger. And in states where mandatory laws have been enacted, an even higher percentage want them to remain on the books.

I've spoken of changed expectations as well as changed designs. Before I leave this subject, may I point to the single most encouraging example of grassroots citizens leading their government toward safer highways. For while engineers were responsible for anti-lock brakes, improved steering columns and anti-lacerative windshields, it was citizens by the millions who changed the way we view the drunk driver in this country. Some of our European friends have had much tougher drunk driving laws, but America is finally beginning to catch up. A decade ago, too many Americans regarded a drunk driver as only a nuisance. Today, we see him as a potential killer—and rightfully so. DOT is working hand in hand with aroused groups of citizens and state legislators across the country to change attitudes and laws. And we will not rest until we get every last drunk driver off the roads and highways of this country, nor will we accept toothless laws and lenient judges. This is one change still unfolding, and we have a way to go on this front.

In 1984, we vigorously supported—and President Reagan signed into law—a bill encouraging states to set 21 as their legal minimum drinking age. Forty-seven states have now done so. Although we normally defer to the states on traffic law issues, as the President said, a uniform drinking age will do away with "blood borders," where teenagers have a positive incentive to drink and drive, to cross state lines to take advantage of lower drinking age laws and then make the return trip home "under the influence."

Statistics show that setting 21 as the legal minimum drinking age works. While drunk driving remains the leading cause of death for our young people, the proportion of teenage drunk drivers has dropped from 28 percent in 1982 to 20 percent in 1985, a significant and encouraging decrease.

The human factor forms but the first leg of what I call the safety triad for our highways. The second rests on the condition of our highways and bridges. There's progress there as well. Our interstate highway system—the safest, most efficient highway network in the world—is almost complete, and we're rehabilitating and repairing roads and bridges at record rates. We're preserving and protecting the system of highways we depend on so heavily, both for our commerce and our travel.

Then there is the third and final leg in the safety triad—the one which many of you share in your daily work. I speak, of course, of motor vehicle design. My Department joins with our auto industry in looking for vehicle safety improvements that are practical and cost effective. One new feature resulting from that search is the high-mounted stop lamp, now standard equipment on new cars. I approved that requirement in 1983, after years of research, field testing, and careful consideration of costs and benefits. Just last week, we reported that vehicles equipped with the high-mounted stop lamp were 22 percent less likely to be struck in the rear by another vehicle while braking. We are very pleased that the results so far confirm our earlier determination that this simple, inexpensive safety feature is an effective means of preventing many of the rear-end collisions that occur each year. We estimate that once installed throughout the nation's fleet, the high-mounted stop light will prevent roughly 900,000 accidents a year and save 40,000 injuries and nearly half a billion dollars in property damage.

Meanwhile, research and development continues on a wide array of safety technologies. To date, at least six manufacturers have announced plans to provide air bags as standard or optional equipment on some or all of their lines in the years just ahead. And the automakers are committed to produce millions of air bags by the early 1990's. While the air bag is undeniably useful, it is most useful in conjunction with safety belts. But whatever the final range of systems offered, it is safe to say that consumer demand will play a large part in determining what the future will hold.

Since he first entered office, President Reagan has made plain his allegiance to market forces. For the fact is, that we in the Reagan Administration look upon transportation—which contributes $800 billion to our GNP—as the engine of the American economy. Make a car or truck one percent more efficient, and the added sales stagger the imagination. What better reason to experiment with lightweight, easy-to-mould engine parts in place of a metal engine? Or computer-driven brakes, now installed in some top-of-the-line models? Or engines no bigger than those which now power motorcycles, and which self-adjust to different grades of fuel?

If windshield wipers will know when to wipe and drivers can command cruising speed at the sound of their voice—it won't be because government mandated these things. On the other hand, a government that is sensitive to the creative dynamic—one that recognizes many years lead time—such a government can foster an atmosphere wherein individual genius can merge with corporate resources. A century ago, it was backyard inventors, like Selden, Goodyear, and Edison, who changed the face of industry. Today, it is teams of exceptional engineers who advance the frontiers of design and safety. Our commitment to safety remains paramount—safety must never be deregulated. What also hasn't changed is the need for government to clear the deck of burdensome economic regulation and reflect in its own actions some of the same experimental energy which translates a dream from the drawing board to the auto showroom.

We will, for example, continue to remove impediments to technological innovation. We intend our safety standards to encourage new safety technology and designs, not stifle them. And where the auto companies or any group can suggest ways to streamline and update our standards, I am eager to listen. To cite just one example, we believe that our vehicle lighting standards can and should be simplified. We have already made progress to permit new types of headlamps to be used in the U.S. If we could move toward a truly performance-oriented standard for headlamps for both the U.S. and other nations, it would reduce excessive design restrictions on auto manufacturers without compromising essential safety.

In the years just past, great strides have been made. Through both individual innovation and international cooperation, we now have the capacity to design automobiles that can withstand higher-impact collisions; that have better vehicle control and braking systems; greater occupant protection and restraint systems, and overall structural safety improvements. Out of forums such as this have come cars that burn less fuel, that combine safety and style. You have explored the outer reaches of modern technology, and in doing so, challenged the conventions of the field. Here in Washington, it is all too easy to fall into a mistaken line of reasoning, to see no further than the morning headlines or the evening newscast. One can soon begin to believe that people's lives are affected most exclusively by what happens at today's hearing or tomorrow's staff conference. In truth, lives are shaped by those who invent or manufacture a product as much as those who make a regulation.

I'm reminded of a story about the great American Justice, Oliver Wendell Holmes, who once found himself on a train, but couldn't locate his ticket.

While the conductor watched, smiling, the 88-year old Justice Holmes searched through all his pockets without success. Of course, the conductor recognized the distinguished justice, so he said, "Mr. Holmes, don't worry. You don't need your ticket. You will probably find it when you get off the train and I'm sure the Pennsylvania Railroad will trust you to mail it back later."

The Justice looked up at the conductor with some irritation and said, "My dear man, that is not the problem at all. The problem is not, where is my ticket. The problem is, where am I going?"

Where, indeed, we might very well ask ourselves the same question. But however we get to the future, one thing is for sure—we'll get there on four wheels. And however the vehicles of the future look, they will doubtless be the product of your imagination. Along the road, there will be plenty of fresh changes. For every custom was once an eccentricity; every idea was once a dream, including democracy—and democracy's favorite transport—the automobile. Through this forum, we've traveled a long way together. The road ahead looks more promising yet—and the United States Government looks forward to continuing the journey with you.

Awards Presentations

Awards for Engineering Excellence

Chairwoman: Diane K. Steed

In recognition of and appreciation for extraordinary contributions in the field of motor vehicle safety engineering and for distinguished service to the motoring public.

Federal Republic of Germany

Prof. Dr. Bernd Friedel
Federal Highway Research Institute

Professor Friedel, as a medical doctor, has addressed the problems of passive vehicle safety at the Federal Highway Research Institute. Together with other scientists, he founded the European joint biomechanical research project and has performed as technical advisor to the European Communities and the Economic Commission of Europe. In 1980 Professor Friedel was elected chairman of the European Experimental Vehicles Committee. For his major contribution to vehicle safety and his active participation in the International Experimental Safety Vehicles Program, Professor Friedel is especially recognized by this award for safety engineering excellence.

Dipl.-Ing. Rüdiger Schmidt
Volkswagen AG

Mr. Schmidt has been active in automotive safety engineering since 1971 when he joined Volkswagen in vehicle technology research. His development of a lightweight automotive diesel powerplant received worldwide attention because of its performance, reliability, and fuel efficiency. We recognize Mr. Schmidt with this award for safety engineering excellence.

France

Claude Robert Chillon
Centre d'Études PSA of Peugeot and Citroen

Mr. Chillon has been an active participant in automotive safety design and engineering since 1957. His work has addressed vehicle structures and aerodynamics, primary and secondary safety research and testing, aggressivity, side impact protection, and rigorous accident data analysis. In addition, Mr. Chillon has been an active participant in the International Experimental Safety Vehicles Program.

Georges Stcherbatcheff
Renault France

The results of Mr. Stcherbatcheff's safety research work in biomechanics, pedestrian protection, mathematical simulations, passive safety, motorcycle safety, side impact protection, and accident data analysis have truly been impressive. He has published numerous technical papers on these subjects and has been recognized worldwide for his extensive contribution to vehicle safety improvements.

Italy

Dr. Pier Luìgi Ardoino
Fiat Safety Center

Dr. Ardoino was appointed Director of the Fiat Safety Center in 1985 and in this position is responsible for all safety research conducted by the Fiat SPA. Prior to this appointment Dr. Ardoino was especially active in Fiat innovative research programs addressing the problems associated with passive safety, and he was responsible for the development of improved test methodology, test protocol and test equipment now currently in use by Fiat.

Japan

Dr. Masaru Igarashi
Suzuki Motor Co., Ltd.

Dr. Igarashi has made a significant contribution to automotive safety by his efforts to develop and apply simulations to improve small car crashworthiness and safety. The results of his research work advanced the use of crashworthiness simulation techniques to improve 3-point belted occupant protection. For his major contributions to vehicle safety, Dr. Igarashi is deserving of special recognition.

Sadao Taniguchi
Japan Automobile Mfrs. Assn., Inc.

Mr. Taniguchi has been on the JAMA technical staff in charge of safety matters for over 20 years. One of the many results of his efforts was a draft of the Traffic Safety and Nuisance Research Institute's Automobile Type Approval Test Standards.

Mr. Taniguchi has been an active participant in all Experimental Safety Vehicle Conferences and has functioned as the JAMA focal point for all Japanese arrangements in support of the International ESV Program.

Sweden

Hugo Mellander
Volvo Car Corporation

Since Mr. Mellander joined Volvo in 1974 as a specialist in biomechanics of impact, he has made major contributions to Volvo's effort to gain increased knowledge in biomechanical injury and protection criteria. One particularly noteworthy contribution was his initiation of a project which led to the development of a load-sensing face for crash dummies.

In 1983 he became Group Manager for Volvo's Advanced Engineering in Traffic Safety and has been active in engineering and in the introduction of new safety features in Volvo production cars.

United Kingdom

Anthony H.K. Denniss
Norton Motors (1978) Ltd.

For over 20 years Mr. Denniss has played an important part in the acceptance of safety features as essential components in motorcycles. Such features include mechanical antilocking brakes, unspillable fuel tanks, and daytime running lights. Currently, Mr. Denniss' research has involved the development of leg guard protection for motorcycle fairings. The demonstration safety motorcycle ESM3 presented at this conference is the result of Mr. Denniss' work.

Stanley J. Edge
Scheller-Clifford Ltd.

Since 1965 Mr. Edge has been responsible for improving the safety of steering wheels at Scheller-Clifford Ltd. In 1984 Mr. Edge produced a prototype steering wheel which met the pendulum impact tests developed to reduce facial and brain injury to drivers wearing safety belts. His first design was successful despite the fact that almost all existing production steering wheels failed. Regulations to require safety levels demonstrated by Mr. Edge's design improvements are now under discussion in Europe.

William T. Lowe
Leyland Trucks Ltd.

Mr. Lowe is currently head of the Leyland Technical Center and is recognized for his work as chief engineer, designer, and project manager for the development of improvements in the safety performance of Leyland Trucks. Mr. Lowe is responsible for the TX 450 Leyland Technology Demonstrator, a heavy truck which incorporates many advanced safety features. For his outstanding efforts to improve heavy truck safety Mr. Lowe receives the award for safety engineering excellence.

United States

Alfred G. Beier
Navistar International Corporation

Mr. Beier made significant contributions to truck safety through his active participation and leadership as chairman of the truck brake committees for the Society of Automotive Engineers and the Motor Vehicle Manufacturers Association. Most notable among his contributions was the significant upgrading of medium duty hydraulic brake system performance initiated in the early 1980's which led the industry to the use of more effective brake systems.

Nancy A. Bundra
General Motors Corporation

Ms. Bundra headed the effort to develop data describing static and dynamic anthropometric measurements, functional reach, and functional accommodation preferences of the United States truck driver population. These data were translated into recommended test procedures and design guides for truck driver accommodation/packaging, including seat position steering wheel position and arm and foot reach. For her dedication to the design of truck cab environments to ensure a nonfatigued, alert driver, Ms. Bundra is deserving of special recognition.

United States

Peter Every
Kelsey Hayes, Inc.

Mr. Every has significantly advanced the safety of light trucks through his efforts to develop a low cost and highly effective rear axle antilock brake control system. The improved stopping performance, controllability during emergency braking, and cost effectiveness of this system has encouraged its adoption as standard equipment on many new model light trucks produced in the United States. Mr. Every is to be commended for his efforts to improve vehicle safety and as such is highly deserving of this special recognition.

John Repp
Ford Motor Company

Mr. Repp has made outstanding engineering contributions in the development of air bags for production cars. He led the technical team that developed the driver and passenger air bags installed in the 1981 Lincoln test fleet and more recently was responsible for developing the driver side air bag currently available on the Ford Tempo and Mercury Topaz. This work is particularly noteworthy since it demonstrates the application of air bag technology in a small, affordable car.

Section 2
Government Status Reports

Chairman: Howard M. Smolkin, United States

United Kingdom

David Lyness,
Head of Vehicle Standards and
Engineering Division,
Department of Transport

Recent trends in road deaths are set out in the following table

	1983	1984	1985	1986 (provisional)
Car occupants	2,019	2,179	2,061	2,245
Pedestrians	1,914	1,868	1,789	1,848
Motorcyclists	963	967	796	758
Cyclists	323	345	286	272
Bus and Coach Occupants	38	37	32	25
Others (mainly lorry or van drivers)	188	203	201	250
	5,445	5,599	5,165	5,398

The figures for total deaths on the road in Great Britain have shown no clear trend over the last few years. The provisional number of deaths in 1986 was 5,400 which is also the average of the preceding 3 years. That is slightly less than 10 per 100,000 of our population. Of the 5,400 deaths in 1986, 42% were car occupants, 34% pedestrians, and 14% motorcycle riders. This distribution is more orientated to car occupants than motorcyclists than was the case in previous years. This is largely explained by the fact that whereas car traffic continues to grow, motorcycle traffic has declined over that period. In 1986 there were 0.9 car occupant deaths per million car kilometres. The comparable rate for motorcycle riders was 14.

Compulsory Seat Belt Wearing

At last year's ESV conference I described the initial success of the regulation we introduced in 1983 to require drivers and front seat passengers to wear their seat belts. That regulation had to be renewed by Parliament within 3 years and the necessary debate took place towards the end of 1985. The evidence of public acceptance and effectiveness in saving many lives led Parliament to make the regulation permanent. We are particularly pleased with the very high rate of compliance with the law. This has varied very little throughout the 4 1/2 year period to date and continues at around 95%.

Seat Belt Fitment and Performance

Success with front seat belts has led us to introduce fitting requirements for rear seat belts in all new cars from April 1987. The regulations allowed the user to fit restraints for children or for disabled people instead of the normal adult belts, of which either two 3 point or three lap belts are permitted. We shall also be requiring lap belts for the exposed forward facing passenger seats in long distance coaches. These exposed seats are those which have no other high backed forward facing seat or suitable restraining barrier directly in front of them.

We are looking closely at the design and installation of seat belts and in our view small changes here could make a considerable improvement in acceptability, comfort and performance. We have also been looking closely at design of child restraints, and the UK has contributed a paper to the Conference suggesting some detailed improvements.

Improved Design of Steering Wheels

Improved design of steering wheels has clear potential to reduce facial injuries to drivers wearing seat belts. This was the subject of a report by TRRL to the Oxford Conference. Since then user trials have shown that padded steering wheels meeting an acceptable performance standard are also acceptable to users. Individual vehicle manufacturers have taken up the idea and discussions in the European Community are underway so as to establish a type approval specification.

Accident Studies

We have a programme of in depth accident investigation involving Department of Transport vehicle examiners, the TRRL, and Birmingham and Loughborough Universities which will continue until 1989. At present we have detailed computer access to some 1439 accidents involving 1618 vehicles and 2720 occupants for analysis purposes. Each accident contains 7 vehicle listings and 6 occupant listings containing between 23-114 variables in each listing. This will allow a very comprehensive range of questions to be addressed and TRRL is presenting a paper on the results from the first analysis of the data base.

Car Occupant Protection in Frontal Impact

Seat belt wearing has had a major effect in reducing the effects of many types of impact. This has exposed the effects of intrusion and lack of passenger compartment integrity in many medium to high energy impacts. Possible improvements are suggested in another one of the papers from TRRL being presented later this week, and some of these improvements are incorporated in the demonstration car ESV 87.

Side Impact Protection

We continue to regard this as an important priority for international cooperation to determine all the elements of a standard. Work since Oxford has brought us much nearer agreement on a usable dummy and on the specification of the barrier.

Pedestrian Protection

We in Britain are especially concerned with measures to reduce the risks to pedestrians who comprise about one third of all road user deaths. More than half of these clearly result from the pedestrian being hit by the front of a car.

The proposals for a simplified test procedure, put to you at the Oxford Conference and based on mathematical simulation, have now been supported by tests using dummies. Our demonstration car ESV 87 incorporates the design changes to reduced pedestrian injuries which were demonstrated in 1985.

Motorcycles

Casualty rates for motorcycle riders are far higher than for any other category of road user, and it is particularly tragic that so many young people are killed and seriously injured in motorcycle accidents.

On the vehicle safety side we believe much more effort is needed by all concerned to make motorcycles safer. The technology is available to improve both active and passive safety. The TRRL's latest ESM motorcycle presented here this week shows what can be done particularly as regards anti-lock braking and leg protection.

The anti-lock braking system demonstrated on that motorcycle is now half way through extensive service trials with police forces in the United Kingdom; first reactions are highly favourable.

Since the Oxford Conference TRRL have continued their work on leg guards to include moving vehicle tests. This is reported in the papers circulated this week. We are very close to having a practical specification which could form the basis of regulations.

Buses and Coaches

Buses and coaches already provide a high degree of protection for passengers. But efforts continue to improve the safety of buses and coaches in a number of aspects. We are now introducing regulations to require new coaches to be constructed so that their superstructure meets ECE Regulation 66. We shall be requiring speed limiters to be fitted to ensure that no coach can exceed 70 miles per hour (112 kilometres per hour) which is the legal speed limit for coaches on British motorways.

We are also working to formulate European standards for flammability of materials used in coaches and for the strength of their seats. Finally, we hope that the European Community will agree on proposals for mandatory fitment of anti-lock brakes to coaches and buses used on inter-urban services.

Goods Vehicles

At Oxford I reported a number of regulations being introduced on lorries and trailers to require sideguards, rear underrun guards and spray suppression devices. These regulations are now bearing fruit as an increasing proportion of heavy lorries on roads in Britain carry the new equipment. We have recently introduced the latest amendment to the European Community Braking Directive Requirements into United Kingdom legislation which gives an improved minimum braking performance of more than 10%. A further change in the European Community is being discussed which will allow Member States to require anti-lock brakes on heavy goods vehicles with trailers. Looking further ahead there are promising developments in front underrun protection described in one of the TRRL papers for this Conference and we are also researching new suspensions which we hope will not only reduce road damage but further improve braking and stability.

Vehicle Lighting

Finally I should like to draw attention to a requirement we have just introduced for all new vehicles to be fitted with dim dip lighting. This requirement is satisfied by a change to the vehicle's wiring so that when the sidelamps and the ignition are activated the headlamps are illuminated at around one-tenth their normal power. This prevents vehicles being driven on sidelamps alone, and provides a level of illumination to make vehicles conspicuous without causing glare—it is especially appropriate in conditions of good street lighting. We are very hopeful that this simple change will have a positive effect on road safety.

Federal Republic of Germany

Rolf Stamm,
Senior Advisor,
Federal Ministry of Transport,
Federal Republic of Germany

Nearly 11 years have gone by since the 6th International Technical Conference on Experimental Safety Vehicles, which took place here in Washington, D.C., in October 1977. Important aspects of our work may have changed in this decade, but our determination to cooperate on an international basis and exchange research findings and development data has remained the same and is also the moving force behind our gathering here once more in the hope to advance motor vehicle safety even further.

In 1984 and 1985 the trend of road traffic accidents in the Federal Republic of Germany was marked by a drastic decrease in the number of fatalities. There were 11,732 fatalities in 1983 compared to 8,400 in 1985, i.e. a decrease of nearly 30%, although overall mileage figures in both years rose by about 3%. The number of occupant fatalities decreased from 6,038 in 1983 to 4,182 in 1985, i.e., a decline of more than 30%.

In 1986, however, the fatality figures were clearly higher compared with those in 1985, that is to say they rose by 6.5% to about 8,950 deaths. Based on the mileage figures, this amounts to a percentage increase by 1.7%. Car occupant fatalities rose by 9.9%. However these values are still considerably below those for 1984.

As was already pointed out at the 10th ESV Conference in 1985, it is especially important in this connection that the noncompliance with the belt usage law in front seating positions incurs a fine of DM 40 in the Federal Republic since 1984. Wearing rates are encouragingly high since that time. The motorway wearing rates amounted to 98% in September 1986, the wearing rates on other rural roads to 96% and in cities to 93%. This applies to drivers and front seat passengers. Since July 1986, a fine for noncompliance has also been introduced for rear seat occupants. For the time being the wearing or securing rates in rear seats have still not reached a satisfactory level. This applies to adults and children. In September 1986, 51% of the adults in rear seating positions of passenger vehicles equipped with rear seatbelts (presently about 80% of all cars) were observed as having worn their belts. Although the rate has doubled compared with the time before the introduction of the fine for noncompliance, the increase in absolute terms is not as high as in the case of the front seat wearing rates after the penalty was introduced. Above all the level attained is incomparably lower. The overall securing rate for children in rear seating positions (sum of seatbelt wearing rate and child restraint usage rate) also hardly reached the 50%-mark in September 1986.

Autobahns (German motorways) continue to be the safest routes in the Federal Republic of Germany. The motorway accident rate (accidents per million vehicle km) amounts to 0.16, while it amounts to about 0.6 on other roads outside built-up areas. Within the framework of a comprehensive exhaust gas emission test the effect of a short term 100 km/h speed limit regulation on accidents was investigated. However, the results did not induce the Federal Government to introduce a speed limit on motorways.

The comprehensive program for the improvement of road safety, which the Federal Government presented in 1984, is being gradually put into effect by introducing a variety of measures. In the meantime, the two-stage licensing of motorcyclists and the probationary licensing of novice drivers have been enacted. Mofa riders are required by law to wear crash helmets. Other measures are being prepared, e.g., the introduction of antilocking systems (ALS) for heavy trucks and coaches, improved mirror systems for trucks to afford a better view of pedestrians and cyclists in the area immediate to the truck. The envisaged model experiment "Fewer Traffic Signs" described in the 1984 Road Safety Program of the government is in progress. The purpose of this model experiment is to reduce the number of signs or improve signposting in built-up areas, the conditions and the criteria under which this could be accomplished. In all, 39 cities applied to participate in the experiment; in three cities measures to reduce and improve signposting were taken and accomplished by the end of 1986. The related accident studies are expected to take another two to three years so that a final report on the evaluation of the safety effects of these measures will not be available before 1989.

In the field of accident research, activities have continued to emphasize the questions of occupant safety and active safety of passenger cars since the last ESV Conference. In particular, the activities of the Federal Government, car industry, and motor vehicle insurers have to be mentioned.

With respect to the motor vehicle insurers' accident research (HUK), the following items are to be looked at:

An investigation of 800 passenger car accidents with respect to "Injuries despite belt wearing" revealed that head injuries, above all to the driver, are still dominating and require intensive efforts to further improve steering column and wheel. By means of design modifications it should be possible to prevent steering columns becoming displaced in an accident and moving inwards and upwards. All measures possible should additionally be taken to eliminate or mitigate head impacts, such as, e.g., safety belt tensioners and airbags. The investigations revealed no major head injuries, not even in accidents resulting in extreme car decelerations, unless the head impacted on a rigid part. Attention is now focussed on setting up a data base of current accident data on the effects of airbags, safety belt tensioners, and modified steering column designs.

The wearing of seatbelts in the rear of cars is not only a vital measure of self-protection but also a measure to protect the other occupants in the car. A research project, which has been completed in the meantime, revealed that in about one out of six accidents occupants run the risk of mutual injuries caused by the occupants who do not wear their belts.

Investigations into child accidents are planned for 1987.

On the sector of active car safety, a study of 2,000 accidents (300 cars equipped with ALS) involving cars of the same model has been undertaken to compare the behavior of ALS-controlled cars with that of cars without ALS. The study is not yet completed, but the safety effects of ALS in critical situations are confirmed by the findings thus far. The accident involvement of ALS-equipped motor vehicles still needs to be investigated. Special studies on 300 truck accidents and 200 bus accidents confirmed the predicted benefits of ALS, namely prevention of about 5% and mitigation of about 15% with respect to accident consequences.

Comprehensive accident material is now available for the first time with respect to truck accidents:

To supplement an investigation based on 1,100 truck accidents into the exterior safety of trucks and partner protection, a study on truck occupant safety has been undertaken, again in collaboration with FAT (Research Association in Automotive Engineering), by investigating 800 truck accidents resulting in injuries to the driver or damage to the driver's cab. Truck safety priorities can thus be rated based on concrete facts.

The car-truck collision tests of the German Motor Vehicle Insurers now point to a solution to the problem of truck front protection which appears to promise success: a supporting structure with a configuration of deformation elements affixed to it.

New tests on motorcycle safety have confirmed the HUK-Association concept that the trajectory can be influenced by special design features, such as, e.g., leg protection, optimized position of fuel container, steering system and sliding feature. In high-speed crashes, a trajectory causing the least possible harm can be achieved by features enabling on-the-spot separation of rider and vehicle, preventing the rider impacting the other vehicle in the crash to the widest possible extent. In minor crashes, grazing collisions or falls, the concept ensures better leg protection. A new series of tests confirmed that airbags, also for motorcycles, can be of considerable advantage—the practical application would however still require tests on the safety of the activation characteristics and the effectiveness of airbags in this field. These questions will be dealt with in a series of tests beginning in 1987.

The accident research institutions of the government continued work on a number of important projects and specified new major research areas.

Since 1984, the on-site accident investigations have been modified based on a new random sampling procedure. It is now possible to also investigate accidents occuring during the night and over the weekend. The accidents investigated are exclusively accidents in the investigation area involving at least one injured accident victim, irrespective of the severity of the injury suffered. A number of special evaluations has also been carried out, e.g., with respect to the protective clothing of motorcyclists, the reconstruction of multiple car crashes, problems regarding the field of vision from commercial vehicles and the comparative effectiveness of different antilocking systems for trucks. A study on the technical defects of bicycles revealed a wide field where action is needed. Within the framework of the on-site accident investigations, a preliminary study on the feasibility of eye tests on accident victims has also been undertaken. The study revealed that the incorporation of such screening tests into the on-site investigations is technically no problem. A main study is expected to provide information on the extent to which the vision of accident victims had been impaired and possibly been the cause of accident. The on-site investigations furthermore included tests on accident victims with respect to various drugs and alcohol.

The Federal Ministry for Research and Technology continues to sponsor research and development projects aimed at improving passive and active safety features of motor vehicles. After the successful completion of the research project "Design of cars affording optimum occupant protection from the viewpoint of the economy"—this project was reported at the last ESV Conference—the projects now in progress are focussing on the improvement of active safety by means of modern information technologies. With the aid of a new driving simulator, the analysis and evaluation of the performance of active car safety features prior to use will be undertaken.

The project also includes especially the optimization of an on board route guidance system and of cockpit design, the survey and analysis of the effects of the route guidance system on driver behavior, and the analysis and evaluation of driving situations.

Within the framework of the European EUREKA research initiative, the car industry, information technology sector, science, and administration have combined resources in the comprehensive joint PROMETHEUS research project—Programme for a European Traffic with Highest Efficiency and Unprecedented Safety—to utilize the possibilities which information technologies are offering to pave the way for a safe, environmentally acceptable, and efficient future road traffic system. The main objective is the real improvement of active safety. This objective should be achieved by means of collecting all information relevant to safety, and especially the information items which are liable to support drivers in critical situations. The broad spectrum of development possibilities has been subdivided into the following three main areas of research:

- development of computerized systems of car features capable of actively/passively assisting drivers with their task of driving (solutions representing autonomous on board systems without outside control)
- development of communication networks among motor vehicles providing a range of electronic "vision" far beyond a driver's normal visual perception range
- development of communication and information systems between roadside and on-board computers for an optimal private car traffic management system.

In a comprehensive project, the possible contribution of the ALI Scout automatic driver information and control system to the improvement of road safety and the reduction of noise and exhaust gas emissions is being investigated. In the city of Berlin, the infrastructural measures necessary for this investigation are being implemented, and it is planned to equip about 900 motor vehicles with the information displays. The bidirectional exchange of data between the infrastructure and the vehicle is achieved by means of infrared signalling devices installed beside the controllers of traffic lights.

In this connection, the development of an accident data recorder also needs to be mentioned. The recorder enables the exact reconstruction of accidents. For this purpose, only the data relevant to accidents are recorded within a time interval of about 60 seconds before an accident and transferred to destruction-proof and manipulation-proof storage devices. Equipment of this nature will especially help the courts. The standard equipment of cars with accident data recorders was called for on several occasions by the Annual Conference of Traffic Court Judges. In view of the European legislation governing equipment parts, this demand has hardly any chance of success. But it may be possible to persuade drivers to use such devices on a voluntary basis for reasons of self-protection.

Still another research project concentrates on the development of a tire which remains operable after a breakdown, i.e., it does not only ensure the operation of the vehicle when it becomes deflated but also retains performance characteristics ensuring the safety of car operation-safe steering characteristics and driving stability. The development efforts are based on a new safety tire concept: the tire bead lies on the inside of the rim channel and not as at present on the outside of the rim flange.

The development of TOPAS, a tractor-trailer based on a new safety concept with optimized passive and active safety features—in this case a tanker truck—has been successfully completed. The essential design feature is the 30 cm lower center of gravity which considerably improves the overturning limit. The knowledge acquired in this project points to the following possibilities of improving the safety of future tanker trucks, e.g.:

- improved driving dynamics by lowering the center of gravity
- introduction of antilocking systems
- monitoring the performance of the safety features of various types of trucks by electronic devices
- monitoring loading and unloading procedures by electronic devices
- measures to improve the fitness of truck drivers
- automatic route guidance systems and automatic headway warning devices
- side guards for trucks.

With respect to the driver and vehicle behavior in critical situations, the work undertaken to systemize and classify critical traffic situations has been completed. Accidents involving passenger cars have been subdivided into groups which differ in the number and combination of disturbed loop components (driver, vehicle, environment). Another project concentrates presently on the vehicle characteristics which are relevant to the accident regime. The characteristic driving maneuvers in accidents are evaluated based on on site accident studies to enable computer simulations to be carried out.

Since the vehicle component is to be considered more than has been done hitherto in the interpretation of accident causes, the joint analysis of vehicle and accident data is required. As a first step in this direction, the accident data records of a Federal state have been combined with vehicle data, such as, e.g., kind, model, and design of vehicle. The absolute number of vehicles in each category of vehicle types involved in accidents has been qualified by relating it with the total number of registered vehicles. First analyses for example focused on the influence of engine power and wheel suspension systems of cars. The type of car accident taken into exemplary consideration in this project is an accident on a rural road caused by speeding, and the resulting loss of control over the vehicle.

As a continuation of the investigations on side collisions, the deformable element of a movable barrier has been slightly modified to comply better with specifications. The element can now be considered as ready for application. By means of a dynamometric force measuring wall, the stiffness of the front structure of current production cars is measured. Based on the resulting findings, the force-deflection characteristic of the barrier element will be adjusted, if necessary.

In respect of side collisions, a project has been undertaken to study the effect of the speed of the struck vehicle. Further details on this project are reported in another conference contribution.

A comparison of tests using the EUROSID dummy in collisions with the EEVC barrier in crabbed mode and with a barrier in 90° mode revealed that the forces generated in the crabbed test constellation are lower. Having subjected the EUROSID dummy to a series of full scale tests, it can now be considered as nearly checked-out for application.

All parties concerned now agree that clear evidence with respect to elimination of submarining should be incorporated into a regulation on frontal collisions. It has therefore been proposed to require the measurement of the forces in the lap belt of dummies in addition to the strain gauges, or submarining detectors used. It has been demonstrated that the additional interpretation of the lap belt force would be a better proof of the elimination of submarining. However, it still needs to be checked whether or not this requirement is sufficient to solve the problem.

With respect to pedestrian safety, a research project for component tests has been started after proposals for a full scale test have been worked out.

At a symposium on rear seat belts which was held jointly with the German car industry it has been pointed out that in view of better wearing rates and the penalty for noncompliance, the geometrical design of these belts and their interaction with the rear seat need to be improved. Such improvements have already been incorporated in the design of some car series recently gone into production.

Studies in the field of the emergency medical service (EMS) system revealed that, among other factors, an accident involvement of about 3,500 rescue vehicles on call will have to be expected on an annual basis including 50 serious injury accidents and 14 fatal accidents. In general rescue vehicles using priority vehicle lights and a special signal horn while on call run an accident risk which compared with other motor vehicles is higher by a factor of 8.

A research project, which is currently in progress and cosponsored by the car industry, deals with the safety relevance of the indicators, controls and switches in motor vehicles.

The range of application of the risk compensation theory is being studied and analyzed under controlled field test conditions. In this connection, the effects of an antilocking system on the accident regime and the behavior of taxi drivers will be observed over a period of several years.

The technical inspection intervals for passenger cars have also been studied. In this connection the possible effects of reducing inspection intervals for older cars were assessed from the economic viewpoint. The study revealed that a reduction of the current two-year intervals—three years in the case of new cars—to one year is not justified. The resulting maximally possible benefits of safety improvements are far from covering the costs of technical inspection of passenger cars.

A study on buses in connection with an ECE draft regulation revealed that the evaluation of the load on passengers in the proposed test method requires that the seats be firmly anchored in the passenger compartment.

The braking test regulations for passenger cars, which have been harmonized between the U.S. and Europe, have to be practicable and satisfying from the viewpoint of the inspection practice. In some points, further improvements should be possible.

The stability of motorcycles at high speeds is still a largely unresolved problem. The influencing parameters have therefore been subjected to a systematic analysis. Suggestions were presented as to how to reduce the weave mode caused by the rider by means of design modifications.

A number of the projects described were undertaken in close cooperation between state authorities and the car industry.

The main research activities of the car industry are the topic of numerous other contributions to this conference.

Environmental impact problems with regard to the car are characterized above all by noise pollution and exhaust emissions. In EEC countries, the noise emission limits for motor vehicles were reduced in the past years and it is intended to reduce them still further. The maximum reduction, applying to passenger cars, buses and trucks, has already been fixed for 1988/89.

As an additional measure, a definition of "noise-controlled motor vehicle" has been incorporated into the German Vehicle Code (StVZO) by the Federal Government. Apart from tougher drive noise limits, limits for other sources of noise have been fixed for trucks. The inclusion of the noise-controlled motor vehicle definition into the StVZO has paved the way for the introduction of user benefits. This fact together with recommendations to environmentally responsible operators (especially those of the state) is hoped to accelerate the commercial availability of "noise-controlled trucks".

Further reductions in the noise pollution of residential areas are expected from traffic restraint measures (e.g., the introduction of a 30 km/h speed limit and design alterations to create traffic or speed retarders, elimination of through traffic), appeals to drivers to drive at low speed, elimination of manipulations and the use of low noise road surfacings.

In the Federal Republic of Germany, the percentage share of motor vehicles in the total of emissions in 1984 was as follows: 58% in the case of nitrogen monoxide (NO_x), 48% in the case of hydrocarbons (HC) and 57% in the case of carbon monoxide (CO). In 1983 the government had already urged that car emissions must be reduced fast and drastically.

A comprehensive exhaust gas emission test with a speed limit of 100 km/h on motorways did not result in the reduction of NO_x emissions expected by various sides. After difficult negotiations, the emission limits shown in the table below were passed by the EEC in 1985 (based on the European car emission test method—ECE Regulation 15) :

Car emission limits in EEC countries

Category of motor vehicles (engine displacement)	Applicable for new models/new motor vehicles as of	Emission limits (g/test)	
> 2l	1 October 1988 - 1 October 1989	OC: HC + NO_x: NO_x:	25 6,5 3,5
1.4 ≥ 2l	1 October 1991 - 1 October 1993	CO: HC + NO_x:	30 8
< 1.4l	A: 1 October 1990 - 1 October 1991 B: 1 October 1992 - 1 October 1993	CO: HC + NO_x: NO_x: To be fixed in 1987	45 15 6

These European limits, just like the currently applicable U.S. emission limits, have already become part of the German legislation in the definition of the "emission-controlled car". This is also the precondition for the official introduction of this car which is planned by the Federal Government as a two-stage measure:

- At the first stage, incentives are offered to promote the purchase (on a voluntary basis) of emission-controlled cars. Car tax law amendments enable savings in car taxes of up to DM 2,200.
- At the second stage the emission limits shown above will become legal on the dates specified. In addition, there will be law amendments to offer tax incentives to car owners willing to have subsequent alterations to the combustion system of used cars undertaken to obtain emission reductions.

The introduction of unleaded fuel, which is the condition for the introduction of the catalytic converter, has been accelerated by a series of measures, the most important being the mineral oil tax concessions of DM .07/l unleaded fuel till April 1, 1987 and DM .06/l at present. As a result of this measure unleaded fuel is now DM .03/l cheaper than the fuel containing lead. More than 11,500 filling stations (out of about 17,500) sold unleaded fuel at the end of 1986, which reached a market share of about 15% at that time.

After the private car traffic, truck traffic comes second in causing emissions. The Federal Government in its "commercial vehicle concept" of 1985 therefore decided on the limitation and reduction of the gaseous pollutants and particulate matters emitted by these vehicles. Negotiations on the introduction of limits in respect of the gaseous pollutants are now being conducted in the EEC. The ECE Regulation 49, which has been in force for a number of years, should be adopted as EEC directive for this purpose. As a first step, it is planned to reduce the limits of this Regulation by 20% (CO and NO_x) and 30% (HC). As a consequence of a voluntary agreement of the German car industry, the 20% lower emission values (compared with the ECE Regulation 49) have already been observed since the beginning of 1986 in the case of new engines.

Since the last ESV Conference, a large number of research projects has been completed, continued, or taken up. The objective of further road safety improvement is a central issue of the joint European PROMETHEUS research project. We hope that the possibilities offered by the information technology will lead to improvements of active safety, and help make road traffic more efficient and environmentally acceptable.

France

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In the field of road safety you can never take anything for granted: in 1985, the reduction of the number of road accident victims was very important, which led us to estimating that "the prolongation of the trend observed in 1985 during the next years will allow us to attain our objective (defined in 1982: reduction by a third in five years) in August 1987, which will then represent a yearly number of deaths inferior to 8,500".

This estimation was certainly based not only upon the good results in 1985 but also and above all upon the *significant improvement* obtained since 15 years back: in fact, since 1972, the black year of road traffic (16,617 killed, 388,067 injured, 274,476 accidents with personal injury), progress of safety had been notable though irregular; fifteen years later on, at the end of 1986, we stated that the number of deaths had regressed by 34% compared with 1972, the number of injured and the number of accidents with personal injury going down by an equivalent percentage (−33%), while the number of registered motor vehicles had increased by 56% during the same period (28 millions in 1986) and the overall traffic flow by 50% (360 milliards of kilometers in 1986).

Thus the road user's accident risk per km run had been divided by 2.3 in 15 years. Nevertheless France remains at the bottom of the twelve EEC countries: 10th for the number of deaths per 100,000 inhabitants, 7th for the number of deaths per 10,000 motor vehicles.

However, facts in 1986, as they appear in the following table, are worrying:

	Fatal accidents	Deaths	Injured
1984 1985 1986	199,464 191,096 184,626	11,525 10,447 10,961	282,485 270,745 259,009
Evolution 85/84	−4.2%	−9.4%	−4.2%
Evolution 86/85	−3.4%	+4.9%	−4.3%

Certainly the reduction of the number of accidents with personal injury and that of the number of injured have continued; this is encouraging.

But the increase in the number of persons killed is considerable after the exceptional improvement in 1985; it expresses a *new increase in the average severity* of accidents with personal injury (6 deaths in 100 accidents with personal injury in 1986 compared with 5.5 in 1985). If the growth of traffic on main roads and motorways represents 6.2% compared to 1985, the *death rate* for 100 million vehicles × km attains 2.92, marking thus a *growth of 1.2%* compared with 1985.

A remarkable fact is that this aggravation does not seem to be specifically French; in fact, various data allow us to state that Belgium, Spain, the Netherlands, the FRG, as well as other European countries have recorded a significant increase of fatalities: a cruel paradox made that the European Road Safety Year became a year of increase in accident risk.

In France, according to a first analysis, three main causes seem to be at the source of this regress:

- the deterioration of obedience to speed limits (the average speeds increased by 1-3 km/h according to the road types, the percentages of exceeding speed limits by 2-8% according to the road types!);
- a still important amount of blood alcohol contents;
- the deterioration of the rates of safety belt use during the last few years, in spite of a clear recovery at the end of the year after a specific campaign.

Thus, whether it is a matter of drawing up the balance sheet or giving the main possible explanations, the people in charge of Road Safety state that Road Safety policy should manifest itself by a *reinforced effort* and *continuous management* having become indispensable by the inadmissibly high cost in human life and the very heavy economic cost of road unsafety (almost 82 milliard French francs).

Road Safety Action

So it is in a perspective of continuity and reinforcement that we can describe the Road Safety policy implemented during the last few years. Let us state the most significant facts:

Decentralized Action

The worrying standstill between 1979 and 1981 already stated had led us to considering that, in the field of Road Safety, decentralization prospects should be fully reached so that local action could relay and specify central action. The programmes REAGIR and −10% CONTRACTS tried to increase the "responsibilization" of citizens and social actors, above all territorial authorities and especially local councils. These programmes suppose duration: to the statement of unsafety facts (enquiries into several thousands of serious accidents), to the mobilization of teams of numerous experienced analysts, we should add the necessary prolongation of grouping and co-ordinating actions of public and private partners and the implementation of a departmental *plan* of Road Safety elaborated and proposed by Departmental Commissions.

Various operations and programmes are in keeping with the decentralized action of Road Safety (PLAN Contracts between Government and some Regions, the Programme "Town with more safety, districts without accidents" devoted to car traffic layout experiments, and others).

As far as the philosophy behind the decentralized action is concerned, it was the subject of two events that attracted attention: the EVALUATION 85 Symposium gathering together many international researchers and engineers and the EUROPEAN FORUM held in Aix-en-Provence in 1986 allowing the comparison of decentralized policies and the presentation of very significant case studies.

Social Communication

The mobilization of many actors in decentralized Road Safety policy requires the uniting of conditions of a diversified and active social communication in which the very institutional communication takes place without any veto.

The year 1985 was marked by the organization of the "ROUND-TABLE Conference on Road Safety: New Initiatives" directing its works towards great actors potentially interested in motor vehicle traffic and Road Safety: analysis of the part that the *insurance* can play in the enterprise of accident prevention; action taken by *HEALTH professions* not only in the therapeutic field but also in a perspective of accident prevention and road user training; speech and actions of communication professionals, *advertising agencies*, marketing specialists, mass media people.

In 1986, the organization in Paris of the international congress on "Road Unsafety" by the ATEC allowed us to go beyond the framework of exchange between specialists offering free speech to practicing engineers and allowing to associations of road users and accident victims to contribute to the progressive consciousness that is the base of durable action.

Regulations

However rich the whole set of regulations may be, progress in road safety requires in this field above all updating and operative translation of regulations, but also adoption of new regulations:

- the progressive introduction of means of *technical vehicle control* should allow the diagnosis of thousands of vehicles over five years old on the occasion of a transaction. This law is completed by a decree foreseeing the confiscation of the registration document of a severely damaged vehicle and its restitution after verifying the good quality of the repairs made.

The problems raised by these regulations were dealt with in a round-table conference. On the other hand, in the perspective of the project of EEC regulations in the field of periodic technical motor vehicle control, France is making national consultations in order to prepare a whole project of mandatory repairs of the main safety components.

- In the field of the fight against *Drinking-and-Driving*, the main innovation besides testing and introducing new control means (ethylotests and ethylometers) consists in the preparation and adoption of a new law (17 January 1986) allowing the *immediate confiscation of the driving license* for a maximum duration of 72 hours in case of detection of blood alcohol content in the driver, of manifest drunk driving or refusal to undergo detection, and the firm suspension of the driving license, by an administrative decision, as soon as the detection result is confirmed; this law also allows the immobilization of the vehicle if there is no qualified driver.
- The Law of 6 January 1986 on emergency medical assistance and ambulance transport creates a committee at the departmental level having the mandate to control the quality of distribution of emergency medical assistance and its adjustment to needs; this law foresees the organization of the SAMU medical services (determined in a decree) and in particular the operation of the centres for reception and control of calls.
- Driver training is dealt with in two particular actions:
    - on the one hand, extension to 22 departments of the experimental "anticipated learning to drive" as soon as 16. This is a new type of training aiming at allowing the young driver to discover progressively the driving situations in an atmosphere of confidence and cautiousness in order to facilitate his adaptation to driving after having obtained his driving license.
    - On the other hand, the reform of the training of driving instructors by the creation of a Professional Certificate of Driving and Road Safety, which as soon as 1986 replaced the CAPEC certificate. This new Certificate will raise very considerably the level of training to be given by institutions presenting reinforced conditions of approval.

On the other hand, let us note the recent decision to develop a "national programme of driver training", which will serve as a framework and a guide to driving instructors who, moreover, will increase their qualifications by participating in refresher courses.

- As far as penalties are concerned, we note that they are not dealt with in special regulations or rules, but that the concern of making the road users respect the main safety rules leads to a reinforcement of control and penalties by the rise of penalty rates, the simplification of the administrative confiscation of the driving license, the generalization of equipment of Police and Gendarme forces. In parallel, it was decided to modernize the national data bank of driving licenses. A new bill being carried in Parliament multiples by 2 and 4 imprisonment and fines for drunk drivers in cases of accidents.

Technical Action

The action aiming at improving infrastructure and the road network, a permanent action, continued in 1985 and 1986. Besides the detection and suppression of "black accident spots", it contains: a policy of systematic taking into account of safety in the "coordinated strengthening" programme; layout of numerous isolated points (especially round-abouts), conventional or new road equipment (for instance installation of emergency exit beds on motorways); a specific action of installing 2,000 speed-reducing humps in special areas. Government also accelerated the rhythm of motorway construction, which will go beyond 200 km/year.

We should also note the organization of a great study cycle on the theme "Infrastructure and Road Safety", in which a great number of engineers and technicians of departments and cities participated.

In the field of the motor vehicle, the year 1985 was marked by inviting tenders for "Aids to Driving" allowing in the perspective of inciting to innovation to select technical research works in various fields: in-vehicle road information devices, devices of detection of lowered vigilance, various detection and alarm systems (anti-collision radar devices, insufficient tire pressure, and others). In 1986, this interest in using the possibilities of electronics led the car manufacturers and other industry partners to participate in the first phases of European projects elaborated in the framework of the EUREKA programme (PROMETHEUS, EUROPOLIS, CARMINAT, and others).

Research

The fusion of ONSER and IRT allowed the creation of a new organization, INRETS, which is thus the main French research institute for road safety.

In the field of secondary safety, in which the ESV Conference takes a special interest, we will note some striking facts:

- The French researchers have actively collaborated in the definition of the testing procedure using the EUROSIDE anthropometric dummy aiming at works on side collisions. The project of a European regulation is being made.
- The studies on protection systems form the subjects of works aiming at verifying wear resistance and conditions of maintaining protection.
- Experimental research works on the protection of pedestrians by means of an adaptation of the forms of the fronts of motor vehicles are being made and may be susceptible of developments at the European level;
- Finally, the last phase of elaboration of the experimental lorry VIRAGES has begun.

Prospects and Proposals

We stated above the paradox of a European Road Safety Year during which appeared a considerable increase of accident risk. This fact has the meaning of an alarm signal and should attract the attention of all the Road Safety partners and especially of decision-makers, on the one hand, and researchers, on the other hand.

In the framework of this international conference, it is useful to think about our common conditions of progress and, taking into account European perspectives, about necessary convergence of some efforts. Among others, we will mention three fields of co-operation for research and for action.

- *Management of speed limits* is a permanent and capital element of a Road Safety policy; it is time to admit this and draw all the useful consequences of it. Especially in this field, co-operation governs success;
- Development of *road transport of goods* has many positive aspects for economy; but it supposes a concerted device of surveillance and the control of the road unsafety it generates. Any evolution of regulations, organization, economics or finances should be examined drawing up an inventory of its potential effects in the field of safety.
- Passenger car or professional driver training depends on educational systems and institutions whose means finally determine the effectiveness and conscience of operators; a socioeconomic analysis of driver training made on a comparative basis commands the attention of all those who consider Road Safety as a major objective.

Italy

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Since the last ESV Conference, many improvements have been introduced in the field of motor vehicle safety. In particular, in Europe and also in Italy, the year 1986 was devoted to traffic safety, during which some noteworthy initiatives were developed. Among these I would like to mention the establishment of the EEC ERGA Safety Working Group which not only had an active role during the "Year of Safety" but is still working at problems of more difficult solution. After more than one year of meetings between Government and test laboratory experts, some concrete results, as well as indications for the future, have been provided by the Group to the European Common Market Commission. The impression I received—based also on the reports by my colleagues—is that in this type of discussion there is often the desire to conclude the work undertaken at any cost. This partly neglects the fundamental objective, which is: the achievement of scientifically sound tradeoffs between the practical feasibility and needs of new generation vehicles. Therefore, it is important for the enhancement of safety to abandon extreme positions even if they sometimes appear valid in the light of available, but sometimes limited, knowledge. After more careful and thorough consideration, it appears that those test methods or legislative requirements which might be desirable now could be much more effective if they are harmonized with the actual needs. These considerations are not meant to reduce the importance of establishing, beforehand, performance criteria capable of simulating, with sufficient accuracy, the behaviour of both man and vehicle: and this is particularly true when considering the impact tests criteria.

In establishing performance criteria for a safety standard two elements, namely man and vehicle, should be considered together. It is important that this need be recognized, otherwise regulations will be in conflict with one another, and will not be based on the reality of actual accidents. Legislative improvisation in fact is no longer an appropriate method of writing safety standards.

It is vital that the effectiveness of safety standards be assessed and that every single safety regulation be meaningful, effective and formulated in objective terms. In order to find out whether a legal measure meets this criterion a parallel research is required where the feedback from accident analysis is obtained to evaluate the extent of the benefits and of the disadvantages connected with the introduction of any safety related requirements. However, a proper implementation of this research depends on adequate knowledge coming from multi-disciplinary accident analysis and experimental biomechanical studies. Both these areas of research require more extensive development before the present generation of safety standards can be soundly based.

As regards the results obtained so far, due note should be taken of the many advances in the field of active and passive safety. For active safety it is very difficult to quantify the benefits which have been achieved. However, it is enough to drive a modern car or truck and then to drive one dating from the early 1970s to realize that very substantial improvements have occurred. These relate to improved fields of view, better mirror systems, heated rear windows, rear window wipers, and improved lighting and signalling systems.

A second area where advances have been made and in which Italian vehicles have gained recognition for their performance is in braking and handling. Tire adhesion, particularly under wet conditions, has improved greatly in the last decade, as well as the braking and handling of modern cars. Brake fade is no longer a significant problem and antilock braking systems are becoming more widely available, particularly on commercial vehicles.

Improved ergonomics and comfort, and reduced noise have safety related benefits in terms of a reduction in driver fatigue. Many of these factors are behind the reductions in crash rates. Most of these measures have not been the result of regulations, but have already been introduced by vehicle manufacturers as a result of in-house research and development.

In the field of passive safety, we have seen very significant improvements in seat belt technology. All these improvements and the structural characteristics of the passenger compartment and the deformable structures around it can be used to provide optimum levels of protection for the occupant, only if he or she is correctly wearing a seat belt.

However in a severe collision even a correctly restrained driver risks injuries to his face and head, because contact with the steering wheel is still possible. A great deal of research has to be done to define a more biomechanically correct approach in this area.

As regards the voluntary adoption of safety devices in our country, we must recognize that European vehicles are now fitted for the most part with head restraints. However, I would like to point out that devices such as these are useless if passengers are not aware of the importance of a correct adjustment. The same consideration also applies to the seat belt anchorage adjusting systems being investigated in Europe and likely to be fitted to some vehicles.

The last decade has seen many improvements in vehicle design. Further improvements are intrinsically more complex because they involve the basic structure of the car, both in terms of its dynamic crash characteristics and the energy absorbing characteristics of the interior of the passenger compartment. For an improved pedestrian safety, the exterior shape and structure of the car should be considered as well. These developments often require new materials and new production methods as well as sophisticated design. For these reasons further levels of crash protection will be increasingly expensive to achieve.

This equally applies to commercial vehicles (for example lateral protective devices), where an adequate compromise between conflicting requirements for the protection of other road users and the operability of the vehicle must be found.

In summary it should be objectively recognized that in the past decade significant improvements in traffic safety have been achieved due to numerous changes in vehicle design. Further improvements will only occur in minor steps, notwithstanding the introduction of extensive vehicle design changes.

In contrast, the European highway infrastructure offers many opportunities for major improvements in road safety which would be extremely cost effective. In addition improvements can be achieved in the area of road user behaviour.

There are relatively few proven programs which can be shown to actually improve road user performance so that accident involvement is reduced. Dealing with alcohol abuse is one of such programs. In the long term traffic education at school level is necessary for effective and safe use of the highway system.

New interesting possibilities are offered by modern technology, in particular by the use of electronics. For example, an electronic control system that senses road conditions, driver's behaviour, and vehicle load conditions can immediately vary suspension spring and shock absorber characteristics. It is thus possible to make suspensions rigid and safe for the driver who negotiates curves in a sporty manner, or soft and comfortable for relaxed driving. Even more important is the application of electronics to brakes. Wheel antilock systems, already fitted to fast and luxury cars, will become more popular as cost decreases.

For commercial vehicles a partial antilock system—namely, a system not operating on all wheels but nevertheless capable of preventing vehicle skidding—could be fitted.

Great care shall be taken not to create a "mistaken" sense of safety which will result in a demand for higher vehicle performance because of the new devices. For example, a wheel antilock system with sophisticated performance should not lead the motorist to believe that he is a racing-car driver and that performance of his vehicle is much higher than standard, thus inducing him to exceed his own limits and those of the vehicles.

I would like to mention also several provisions, introduced by the Italian Government in the field of safety. To begin with, there is the mandatory use of the crash helmet for motorcyclists, and new rules for the certification of two-wheeled vehicles. Attention was also given to heavy vehicles such as buses and trucks which make up a large part of total road traffic. For these vehicles, there is a stage-by-stage plan for the adoption of speed limiting devices, antiskid devices, and side and rear backward-reflecting indicators. We shall also be looking carefully at the information gained from this conference, to examine what part it may play in the adoption of new legislations.

Nevertheless, my government is convinced that all this praiseworthy research into vehicle safety should not continue to be seen as a separate factor, but should be included in a more general overview of transport policy. Indeed, our Italian 1984 law provided for drawing up of a General Transport Plan, which was prepared and approved by the Government in May, 1985. This plan points out: the need for a rationalization of the institutional structure governing the transport sector, the course of action to be taken to fit action to need, and the necessary laws and legislation; it also deals with a series of problems related to the social aspects of the transport problem, including safety. The plan contains precise information regarding the policies to be followed for road safety, understood as a component of the national transport system.

It is in this context that the Italian Government is following the proceedings of this ESV Conference with keen interest.

Sweden

Prof. Bertil Aldman,
Department of Traffic Safety,
Chalmers University of Technology,
and
Olle Eklund,
Head, Vehicles Department,
Swedish Road Safety Office

Accident Statistics

The positive trend in Sweden, indicated by decreasing numbers of police-reported road accidents since 1970, was unfortunately broken 5 years ago. From 1968 to 1982 the number of fatalities decreased by 40% but has since then increased again by 10%. For nonfatal cases there was a 20% decrease in that period followed by an increase of 15%.

The most negative trend during the last 3 to 5 years was found among car occupants. Driver fatalities and injuries increased by 35% and passenger casualties only slightly less. During the last 5 years the increase in traffic volume was 13%.

Studies have shown that the average speeds on the Swedish roads increased during the eighties and this is believed to be the reason for this negative tendency. However, stricter enforcement of speed limits seems to have changed this unfortunate development during the last months.

For motorcyclists and pedal cyclists the negative trend which began around 1980 seems to have been broken in recent years.

In 1985 the total number of people killed in road accidents in Sweden was 808; 20,671 were injured, and 5,814 of these suffered serious injuries. The complete official statistics for 1986 are not yet available but 844 fatalities have been confirmed.

At the end of 1986 the number of cars in use was nearly 3.5 million.

A New Traffic Policy

Within the Swedish Government work on a revision of the current traffic policy is in progress. The ambition is to make a proposal to the national parliament in 1988.

In this work road safety and environment matters have a central position. Among the items which will be given particular consideration, the following can be mentioned:

- Driver education,
- Speed adaption,
- Anti-locking brakes on heavy vehicles,
- Requirements on tires,
- High mounted stop lamps, and
- Road safety measures within the road maintenance program.

A number of authorities, organizations as well as companies and private persons are now working together in Sweden to achieve better road safety. For some of the institutions road safety is their main objective; others are involved in this work to a lesser degree.

Coordination and Planning

The Swedish Road Safety Office is the central administrative authority for matters related to safety on the roads. A few years ago the office was given increased powers for the task of coordinating the efforts of all institutions involved in this kind of work.

In the fall of 1984 the Swedish Government decided that a special council linked to the office should be appointed for the task of planning and coordination of road safety activities in Sweden. It was also decided that a new staff function, a planning department, should be established.

The new directives also stipulated that the Swedish Road Safety Office should produce a traffic safety plan every year.

The council for coordination and planning consists of representatives from nine leading authorities and organizations in the field. Its function is an advisory one but it is expected that each authority and organization should carry out its share of jointly decided activities.

In this context the word coordination can be defined as the total long-term planning of all the activities of the parties involved.

The parliament has formulated the objectives for this work. These include a progressive reduction of the total number of casualties in all road user categories as well as a progressive reduction of the risk factors to which these people are exposed.

The new staff function, the planning department within the Road Safety Office, was formed at the end of January 1986. This department will serve the council with the information necessary for its decisions.

The 1987 Traffic Safety Plan

In accordance with its instructions the Road Safety Office has published a traffic safety plan for 1987. It addresses the following problems:

1. Speed adaption
2. Unprotected road users
3. New driver license holders
4. Drunk driving

5. Disabilities among road users
6. Drivers' perception
7. Children in traffic.

This plan will be the basis for all activities of the authorities and organizations represented in the council as well as for all other efforts in this field.

Regulations for Vehicles

The Road Safety Office is also responsible for regulations concerning vehicles and vehicle components. Among the activities in this area the following can be mentioned:

- Regulations regarding type approval of protective clothing for motorcyclists have been established and were set in force in December 1985.
- Measures have been taken to improve the visibility of motorcycles.
- The use of wild-bars (bull-bars) on vehicles has been prohibited due to their aggressivity to unprotected road users.
- The use of seat belts is mandatory in the rear seat from July 1, 1986.
- Methods for measuring the tire/road noise have been developed in cooperation with a working group within the Economic Commission for Europe (ECE).

Road Safety Research

The Swedish Transport Research Board has published a long-term program in which priority is given to projects aiming at the construction of models and formulation of theories regarding the function of man as a road user. The need for this kind of project is indicated particularly in behavioral research and in biomechanics.

Among the research activities in progress in Sweden a few will be mentioned here.

A comprehensive study on the use and effectiveness of different types of child restraints has been conducted jointly by The Folksam Group, Chalmers University, and VTI. The analysis of the results has begun and a complete report will be published later this year.

The injury mechanisms in rear-end impacts and the effect of large head motions on the central nervous system in car accidents are the subjects of a joint clinical and experimental study at Chalmers University and the University of Göteborg. A comparison is also made between long-term complaints after traumatic loading of the neck in car accidents and similar symptoms from static loading among industrial workers.

Many accident avoidance problems with heavy goods vehicles are related to their great dimensions and variations in weight. A study conducted in four Nordic countries in 1986/87, with measurements of decisive quantities in the air brake systems and the braking characteristics of these vehicles, will be described at the international conference "Roads and Traffic Safety on Two Continents" organized by TRB and VTI in Göteborg in September 1987.

During the winter periods 1985-1987 investigations were carried out by VTI in a program for improved ECE regulations for antilock braking systems to ensure acceptable performance under winter conditions.

Within the long term road accident research program at Chalmers University of Technology an attempt has been made to design a low cost device which can be mass produced and is capable of indicating in a simple way the severity of a car accident. A report of the function and feasibility of this device will be presented at the 1987 International IRCOBI Conference, September 8–10, in Birmingham, United Kingdom.

The European Experimental Vehicles Committee

Prof. Dr. Bernd Friedel,
Director,
Bundesanstalt für Strassenwesen,
The European Experimental Vehicles Committee

As I mentioned at the 10th ESV Conference, 1986 was declared road safety year in Europe by the European Communities. A large number of activities were undertaken in the different member states to promote traffic safety on the roads. It is difficult to assess the benefits of such a comprehensive programme—but there is no doubt that road safety is an issue which demands the continuous attention of the world's governments and administrations, of the public, and last but not least, of the research sector. Against this backdrop, there can be no denying the necessity for further improvements in motor vehicle safety.

Since 1985, the EEVC has therefore continued its work on car safety, especially with regard to side impact protection. At the Kyoto conference five years ago, an EEVC report was published on Structure-

Improved Side Impact Protection in Europe. Reviewing the accident data available, recommendations were made at that time for a suitable test procedure consisting of a full scale test in which the car being tested remained stationary while being struck in the side by a movable deformable barrier with a 90° lateral impact. Since that time considerable research has been conducted in different countries to develop the test procedure so that it would be available for consideration for legislation in Europe. For example, two different types of barrier face have been developed on the basis of the EEVC specification. Various activities of European institutions towards the proposed test procedure have been sponsored by the Commission of the European Communities (EEC). The results of all this work have been presented by the EEVC on such occasions as the NHTSA Public Meeting on Side Impact held one year ago here in Washington, D.C.

The developed barrier faces worked well in crash tests with real cars. At the end of 1986 the specifications of the stiffness of the barrier were reviewed. It was also recognized that the initial inertia effects of the masses of components at the very front of cars give rise to initial peaks. The problem of determining the appropriate height of the barrier face above the ground level was also tackled. The results of this reconfirmation will be presented in detail at this conference in the technical session devoted to side impact protection. The proposed test is still a 90° test at 50 km/h as announced in 1982. The ground clearance of 300 mm is now preferred if the aim of the regulation is intended to improve the safety of today's cars. It seems some deviations from the force/deflection corridors are unavoidable. In general, the work on the test conditions has been finalized. Therefore our results have been handed over to the Economic Commission for Europe (ECE) as well as to the Commission of European Communities (EEC) as scientific input for drafting appropriate regulations and/or directives.

At the last ESV Conference in 1985, we mentioned our efforts to develop a European Side Impact Dummy (EUROSID). Guidance on the overall features of the design of this dummy has been provided for many years now by the EEVC Ad Hoc Working Group on Side Impact Dummies under the chairmanship of Mr. Neilson from the Transport and Road Research Laboratory (TRRL) in the United Kingdom. First prototypes have been built. A validation programme was conducted which was finalized at the end of last year with the financial support of the EEC and in very close cooperation with INRETS and the Association of Peugeot and Renault from France, TNO from the Netherlands, BASt from the Federal Republic of Germany, TRRL from the United Kingdom and several associated organisations. This programme was set up to provide information about a large number of aspects of the dummy's performance such as biofidelity, scaling of injury criteria values, repeatability of response to similar impacts, reproducibility, and sensitivity of response and durability.

At this conference EEVC will now present the results of about 500 tests (e.g., impactor, sled, and full scale tests) conducted on these aspects and referring to the different body areas injured in side impacts. EUROSID has been shown to be at a very satisfactory state of development. The EEVC main committee has therefore decided that the dummy in its present form is ready to be evaluated in laboratories and testing institutions around the world. A few changes in detail may be necessary after the tests have been carried out by these institutions. The dummy is specified for use as a test measuring tool. The above-mentioned EEVC Working Group will continue its work until the end of this year, particularly in suggesting appropriate injury criteria and protection criteria values and evaluating possible design adjustments. A consortium of three companies was set up last year to produce EUROSID. The dummy and spare parts are marketed by TNO, Netherlands.

At the end of 1986, a seminar on EUROSID was conducted in Brussels in order to present the dummy to interested parties in the national administrations, automobile and component industry, and research and test organisations.

The Commission of the European Communities has started discussions among the twelve member states to develop a new directive on side impact protection using the EEVC proposal for both the barrier and EUROSID. The Commission has repeatedly expressed a firm wish for international harmonization. The EEC would like the governments of the United States of America and Japan to enter into international negotiations with the European governments concerning this issue of lateral protection.

Working in close cooperation with the Commission, the EEVC has loaned one of the prototypes to NHTSA for test purposes. EUROSID is on display at the exhibition being held during this conference.

International cooperation is also under way with CCMC, MVMA, JAMA, and Transport Canada regarding the European Side Impact Dummy. In all these cooperative efforts EEVC is looking to facilitate an exchange of scientific knowledge and to discuss the results of research and development.

Since the 10th ESV conference, EEVC has conducted a study on Heavy Goods Vehicles, also under the chairmanship of Mr. Neilson (TRRL). We are pleased to present our results at this conference. They relate particularly to the accident statistics for our different member countries and to safety measures which should improve accident avoidance and injury protection.

In 1986 the Government of Spain decided to join the EEVC. Seven European governments are now represented on this committee. The European Commission is also involved in our work as an observer.

From the time of its inception, the European Experimental Vehicles Committee has put its effort into coordinated research and has relied upon international cooperation which has always been forthcoming. This also holds true for this 11th ESV Conference.

Japan

Kenzo Inagaki,
Acting Deputy Director,
Automobile Division,
Ministry of International Trade and Industry

It is my great pleasure to be given this opportunity to deliver this status report by the Japanese Government on the issue of automobile safety. Nearly two years have elapsed since the opening of the 10th International Technical Conference on Experimental Safety Vehicles at Oxford, Britain, in July 1985, and in the cause of making automobiles a truly integral part of human society, it is significant that experts in automobile safety technology from many countries have gathered here to share and discuss their research findings.

When we think of automobiles, we are apt to focus our attention on designs and performance, but I believe that the safety issue has the most importance in our attempt to win a full acceptance of automobiles in society. I should add that traffic safety is a vital ingredient for the development of both our industry and our culture.

For this reason, in this status report from Japan, I will focus my attention on new developments in safety issues in Japan since the last conference two years ago.

Current Status of Traffic Accidents in Japan

In 1986, there was a total of 9,317 people killed in traffic accidents in Japan. This was a 0.6% increase compared to 1985. It is a sad fact that the number of traffic deaths has increased moderately but at a steady pace for the past several years, despite our efforts to reduce traffic accidents. However, if we note that the number of automobiles in use has increased by 4.2% and the number of traffic accidents has increased by about 4% in the past one year, the 0.6% increase in traffic deaths is statistically consistent with these trends.

One of the features of traffic accidents in Japan today is an increase in the number of accidents involving people 70 years or older. This is a result of an increase in the number of aged drivers, in parallel with the aging of the Japanese population as a whole. Since the percentage of senior citizens will continue to increase for many years to come, the Japanese Government is engaged in various activities to promote safer driving by older citizens.

For instance, the Government is discussing the possible introduction of more easily recognizable traffic signals, and has introduced a self-checking system for the aged to measure their own aptitude for driving, while expanding traffic safety education programs for senior citizens.

As a factor contributing to the small increase in traffic deaths, I should mention that it has become mandatory for drivers and passengers to use seat belts. The use of seat belts became mandatory from September 1985 for driving on highways, and from November 1986 for driving on ordinary roads. As a result of this enforcement, the rate of seat belt usage on the highways climbed sharply from 52.3% before the enforcement to 95.3% after it came into force; the corresponding figures for ordinary roads were 56.8% before and 95.9% after.

Since then, the seat belt usage rate has further improved, and today the rate is 99.2% for highway driving. I believe that the current level of seat belt usage in Japan is very high compared to other countries.

Automobile Safety Standards

Japan has been expanding the scope of its automobile safety standards according to the "Second Program Plan for Future Automobile Safety Standards" formulated in October 1980. As you are aware, at the previous 10th International Technical Conference on Experimental Safety Vehicles, we reported on the three safety standards that we planned to strengthen. This plan was adopted in September 1985, after notification to GATT. Let me now briefly explain the standards we actually strengthened.

(1) Seat Belts
First, it has become mandatory in Japan to furnish all automobiles with safety belts on all seats, and to equip the driver's seat belt with an emergency lock retractor device. The only exceptions to this are route buses which do not run on highways.

(2) Windshields
Second, it has become mandatory to use HPR laminated glass for automobile windshields.

(3) Fuel Leakage in Collisions
Third, new requirements have been introduced with regard to fuel leakage caused by rear-end collisions, whereas the previous requirements for fuel leakage pertained only to front-end collisions. In addition, trucks no larger than 2.8 tons GVW have been made subject to these fuel leakage regulations.

International Harmonization of Automobile Standards

Automobile standards to secure safety and pollution control differ from one country to another, reflecting different traffic and social conditions in different countries. Nevertheless, if we consider the internationality of automobiles as a commodity distributed on a worldwide scale, there is no doubt that automobile standards must be internationally harmonized as much as possible, while taking the different traffic conditions of various countries into account.

For this reason, on the basis of the Action Program decided in July 1985, the Japanese Government took steps to improve the automobile certification system, and in addition, implemented various actions for the international harmonization of automobile standards.

For example, the Japanese Government actively participates in the ECE WP29 meetings, where in March this year, Japan proposed a worldwide harmonization of installation requirement for lighting and light-signalling devices.

For another example, at the forum of the Council for Transport Technology, members of the Council heard the opinions of people knowledgeable in the field, including people from other countries, and held discussions on the issues of international harmonization. Furthermore, Japan recently organized a Symposium of the International Harmonization of Motor Vehicles Regulations in Tokyo.

In addition, as a step toward the harmonization of standards between Japan and Western countries, Japan has abolished the requirement to furnish speed warning devices and parking lights, and is planning to create a new center specialized in the promotion of international harmonization some time in the current fiscal year 1987.

I hope that you will understand our active commitment to the harmonization issue, and I would like to ask for your support, for Japan is determined to continue its efforts to promote such harmonization.

Automobile Safety Measures

With respect to safety measures relating to vehicle structure, I should mention that safety structures protecting against front-end and rear-end collisions have already been incorporated into newly produced cars, in accordance with the ESV and RSV experiences. As a measure to protect passengers in accidents, automobiles equipped with air bags are now under production.

Since the prevention of traffic accidents is the basis for safety, Japanese automobile manufacturers are working hard to develop methods of preventing accidents by using electronics technology. In addition to antilock braking systems, the manufacturers have already put to practical use electronically controlled suspensions, electronically controlled real time four-wheel drive systems, and four-wheel steering systems.

Also, research is underway to develop warning systems for maintaining a proper distance between automobiles and preventing the driver from dozing. Furthermore, studies are in progress into the possibility of utilizing high-level communications systems to support the driver by providing traffic information from outside the automobiles.

Research for Automobile Safety

In Japan, the Government and automobile manufacturers are actively engaged in research into automobile safety. The collection and analysis of data on traffic accidents are most important for determining the future course of safety policies and research activities, and statistical analyses of these data are performed by each institution conducting the research.

For passenger cars, studies are conducted to determine relationships between the driver's steering behavior and the occurrence of accidents, the importance of steering stability in four-wheel steering, and the effects of worn-out tires, among other research subjects.

For large trucks, investigations are carried out into passengers' behavior upon collisions, the running performance of trailers, and other subjects.

Concerning motorcycles, research subjects include steering stability, antilock braking, and passenger protection. For both two- and four-wheelers, research is underway to formulate a standard for headlight performance and to discover more about the visibility of rear lighting devices.

At the present eleventh conference, from tomorrow, Japanese automobile manufacturers and other Japanese researchers are scheduled to report on research activities in Japan, and in ending this status report from the Japanese Government, I would like to ask all participants to share with us their frank opinions and insights, in order to gain the greatest benefits from this conference in the interests of traffic safety.

Canada

Dr. Gordon Campbell,
Director General,
Road Safety and Motor Vehicle Regulation Directorate,
Transport Canada

Canadian Accident Environment

Since the early seventies, the number of persons killed annually in Canada in traffic-related accidents has declined dramatically. This number peaked in 1973 when approximately 6,700 traffic fatalities were recorded. By comparison, fewer than 4,100 fatalities were recorded in 1986, the lowest number recorded in Canada since 1962. The reduction is all the more impressive if one takes vehicle travel into account. Between 1973 and 1986, the traffic fatality rate per 100 million vehicle kilometres has declined by almost 50 percent, from 4.2 to 2.2. The 1986 traffic fatality rate is the lowest ever recorded in Canada.

The last decade has seen great emphasis placed in Canada on increasing the wearing rate of seat belts and on the use of child restraint systems. The use of federally-approved child restraint systems is now compulsory in all of Canada's ten provinces for infants and younger children travelling in motor vehicles. The use of seat belts by older children and by adults is presently compulsory in eight provinces. Within the year or so, however, it is anticipated that seat belt use legislation will be introduced in both of the two remaining provinces. Although seat belt use has not yet been made mandatory in either of Canada's two territories, the passage of legislation in one jurisdiction appears imminent.

National estimates of seat belt use by drivers are obtained annually by direct observation in roadside restraint use surveys conducted by Transport Canada. While the long-standing goal of an 80% national restraint use rate has yet to be realized in Canada, substantial progress towards it has been made. The last decade has seen more than a three-fold increase in the usage rate of seat belts by drivers, from less than 20 percent in the mid-seventies to a level now exceeding 60 percent. The restraint use rate of drivers continues to increase steadily. In the context of the above-noted goal, it is particularly encouraging that the latest restraint survey data show seat belt usage by drivers exceeding 80 percent in a number of major cities in Canada.

Crashworthiness Research

Transport Canada's research efforts to improve occupant safety in frontal collisions were outlined in the Canadian Government Status Report at the last ESV conference, held in England in 1985. To date, these research efforts have produced two test devices which are in an advanced stage of development.

The first of these devices is the Belt Test Device (BTD) which permits seat belt fit to be quantified on the basis of a simple in-vehicle static test. The test device consists of a standard SAE H-point machine modified to accept pelvic and thoracic body forms. Each form incorporates a series of scales which allow the position of the seat belt to be defined in relation to specific anatomical landmarks. A small number of kits which allow the H-point machine to be converted to a BTD have been fabricated, and are presently being made available to interested agencies for evaluation purposes.

The second piece of hardware developed is a modified G.M. Hybrid III head form. The modified head form incorporates a frangible facial insert formed from methyl methacrylate. The insert is designed to fracture at impact energy levels based on available cadaver data. To further assess the appropriateness of the design of the modified head form, additional head impact response and facial fracture tolerance data will be gathered this summer as part of a cooperative research project with the Collision and Biomechanics Laboratory of INRETS (Institut National de Recherche sur les Transports et leur Sécurité) in France.

In addition, the research efforts produced a modified Hybrid III chest assembly that more closely resembles the human thorax. Comparative sled and barrier crash testing of the modified and unmodified chest assemblies revealed little difference in the peak deflections measured at the mid-sternum location when the dummies were restrained by three-point seat belt systems. Overall, the testing suggested that the existing Hybrid III chest produces human-like defections at the mid-sternum under loading rates represented in a 48 km/hr barrier crash when the dummy is restrained by a three-point seat belt.

Over the next two years, it is anticipated that a steadily increasing proportion of the Department's research activities in the areas of accident data analysis and vehicle crash testing will focus on the issue of side impact protection. Specifically, the Department will be initiating a comparative crash test programme using anthropometric test devices and testing procedures developed in the U.S. and Europe to assess side impact performance on the basis of a dynamic test of a vehicle when impacted in the side by a moving deformable barrier. The upcoming test programme and accompanying accident data analysis are intended to assist the Department in evaluating the appropriateness, in the context of the Canadian accident situation, of the various regulatory options advanced to date to assess side impact performance.

In addition, the Department is presently examining the potential benefits and costs associated with the mandatory installation of manual three-point seat belt assemblies in the two rear outboard seating positions of passenger vehicles.

Crash Avoidance

Recent developments in Canada in the area of crash avoidance relate primarily to vehicle lighting. Consideration by the Department of the potential benefits of improvements in vehicle conspicuity prompted the promulgation of two additional federal requirements for vehicle lighting.

The first of these pertains to the fitment of a centre high-mounted stop lamp in all passenger vehicles sold in Canada after January 1, 1987. It is anticipated that this measure will reduce the number of passenger car rear-end collisions by some 25 percent. The technical requirements associated with this regulation are identical to those introduced in the U.S.

The second regulation recently introduced pertains to the mandatory installation of a daytime running lights system on all new vehicles sold in Canada, commencing December 1, 1989. It is estimated that this measure will reduce daytime multiple vehicle collisions by some 10 to 20 percent. To promote international harmonization, the technical requirements associated with this regulation are compatible with those in Sweden, Norway, and Finland, where the use of daytime running lights is also compulsory. In this context, we are very pleased by the response of the U.S. administration to our initiative. Measures are in hand to permit the operation of vehicles equipped with daytime running lights in the United States, and we are presently cooperating with them in the development of an evaluation programme for the standard.

Research efforts are also in progress to further improve heavy vehicle safety, particularly in the area of braking and stability. A national survey of the condition of the braking system of heavy vehicles was completed recently by the Department. A study of injury-producing accidents involving heavy trucks is also in progress to establish and rank causal factors. The potential benefits of improved heavy truck brake performance resulting from the mandatory installation of front axle brakes are also being examined.

The Department also participated in a major cooperative research project, funded jointly by federal and provincial governments and industry, on the effects of vehicle weight and dimensional variations on the stability and control characteristics of commercial vehicles and their effects on the strain and deflection response of roadway pavement. The study was intended to provide objective data to support the adoption of a more uniform set of vehicle weight and dimensional regulations across Canada, while maintaining operational safety and preserving the highway infrastructure. The results of the study are currently being used by an implementation committee charged with the responsibility for updating the uniform Canadian operating standards for motor vehicle size and weight.

United States

Michael M. Finkelstein,
Associate Administrator for Research and Development,
National Highway Traffic Safety Administration

Since the Experimental Safety Vehicles (ESV) Conference held in Oxford in July 1985, the National Highway Traffic Safety Administration's (NHTSA) motor vehicle safety program has made substantial progress in a wide variety of areas.

When we last reported on the status of motor vehicle safety in the United States, we were just beginning to see laws enacted requiring the use of safety belts. As a result of safety belt use laws now in effect in approximately half the States, more than 1,500 Americans are alive today. They have been saved by the belt. With respect to passive restraints, almost one million passenger cars equipped with either air bags or automatic safety belts will be sold to

American consumers this model year. Restraint system technology that was developed by the research community is now available in auto showrooms all over the nation.

In addition to our success in increasing the use of restraints by motor vehicle users, we are also succeeding in reducing the number of alcohol impaired drivers on our highways. Efforts throughout the United States have reduced the proportion of intoxicated drivers involved in fatal crashes and we are now expanding our research to cover drugs and impaired driving.

Two years ago, we reported on the National Academy of Sciences publication of *Injury in America.* This marked the completion of a study sponsored by NHTSA to examine the need for a well-coordinated, national trauma research program. Today, we can report on a 10 million dollar injury prevention research program managed by the Centers for Disease Control in conjunction with NHTSA. This program has recently funded five centers of excellence to pursue trauma research and has awarded more than 30 research grants (selected from more than 400 applications).

All in all, we have achieved much in the two years since the last meeting. But very clearly, more has to be done if we are to alleviate the deaths and injuries that are still occurring on our nation's highways.

The Accident Environment

Since the early 1980's, the fatality rate has been dropping. Fatalities per 100 million vehicle miles travelled were 3.34 in 1980 and had declined to 2.47 in 1985. Based on preliminary data for 1986, 46,000 persons died in traffic crashes, an increase of about 2,000 from the 1985 level. This fatality increase is attributable to a significant increase in travel. The fatality rate still remained at 1985's all-time low of 2.47 per 100 million vehicle miles travelled (Table 1).

On average, in each of the last five years, there have been approximately 6 million police-reported accidents from a fleet of 180 million vehicles travelling nearly 1.8 trillion miles. These crashes involved more than 10 million vehicles and injured more than 3.3 million people. Our best estimate is that serious injuries in motor vehicle crashes still exceed 160,000 each year, with fatalities averaging slightly in excess of 44,000 each year (Table 2).

In the U.S., passenger car occupants account for the largest proportion of crash fatalities—approximately 54 percent in 1986. And in 1986, fatalities among light truck and van occupants almost equalled the number of pedestrian fatalities—each group accounting for 16 percent of the highway death toll. The largest remaining group are the motorcycle riders, who accounted for 10 percent of fatalities last year (Figures 1, 2, and 3).

Table 1. Motor Vehicle Traffic Fatalities 1980, 1985, and 1986.

Type of Vehicle	1980	1985	1986*
Passenger Cars	27,455	23,198	24,890
Light Trucks/Vans	7,486	6,690	7,383
Medium Trucks	285	156	170
Heavy Trucks	977	821	773
Busses and Others	580	587	694
Motorcyclists	5,144	4,570	4,530
Nonoccupants	9,164	7,773	7,560
Total Fatalities	51,091	43,795	46,000
Fatality Rate	3.34	2.47	2.47

*Preliminary

Table 2. Magnitude of the Highway Safety Problem 1982–1985.

	1982	1983	1984	1985
Reg. Motor Vehicles	165,253	169,446	171,997	177,135
Licensed Drivers	150,310	154,221	155,391	156,868
U.S. Population	231,534	233,981	236,158	238,740
Vehicle Miles of Travel	1,593	1,658	1,717	1,775
All Reported Accidents	18,100	18,300	18,800	19,300
Police Reported Accidents	5,825	5,861	5,908	6,081
Tow-Away Accidents	2,130	2,221	2,314	2,331
Injury Accident	2,158	2,310	2,372	2,248
Property Damage Accident	3,667	3,551	3,534	3,833
Involved Vehicles	9,875	9,869	10,093	10,452
Involved People	15,318	14,852	15,473	16,108
Fatalities	43,945	42,584	44,241	43,795
Injured People	3,192	3,371	3,563	3,363

Note: All values in thousands except Vehicle Miles of Travel in billions and Fatalities in units.

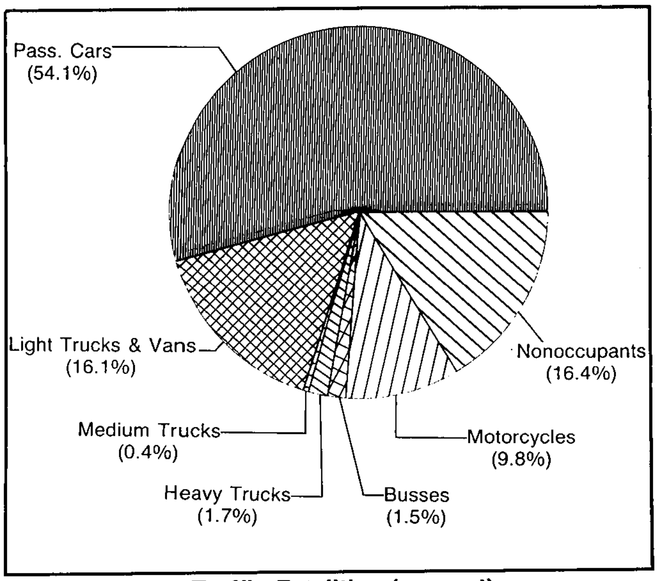


Figure 1. 1986 Traffic Fatalities (percent)

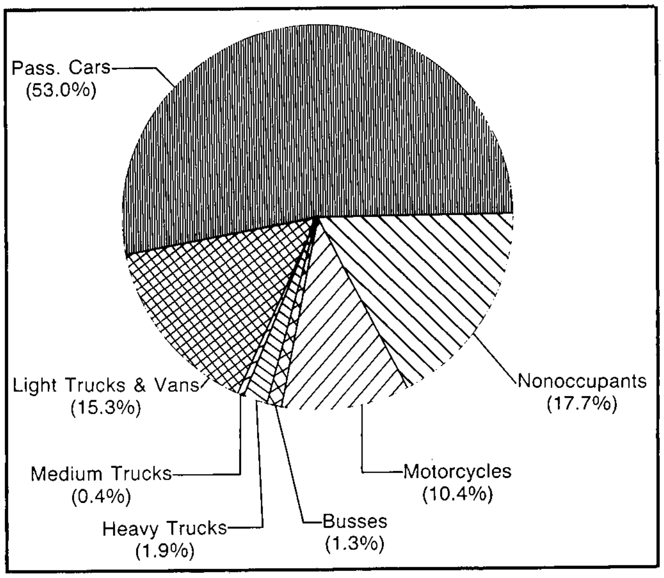


Figure 2. 1985 Traffic Fatalities (percent)

Accident Data Collection and Analysis

Most of the accident data cited throughout this report are derived from the Fatal Accident Reporting System (FARS) and the National Accident Sampling System (NASS).

FARS is a computerized data base containing information on all fatal motor vehicle accidents occurring in the United States. The system became operational in 1975 and currently has records on almost 560,000 fatalities. FARS data are acquired directly from each State's records. The FARS file provides the most comprehensive, detailed, and accurate data on the U.S. national motor vehicle fatality toll.

NASS is a network of trained accident investigation teams that collect data on a nationally representative sample of police reported accidents. Data are compiled and evaluated from detailed accident site inspections, measurements and assessments of damaged vehicles, driver interviews, medical records, autopsy data, and other pertinent records. The NASS system collects information in significantly greater detail, both in the number of variables and the precision of observations, than is available from police records. NASS presently contains over 65,000 cases from 1979 through 1986.

Since the Oxford ESV Conference, we have thoroughly reviewed our data needs, and as a result are in the process of making substantial changes to the NASS system. We will be concentrating NASS on what it does best—the investigation of injury producing crashes—and look to other sources for general estimates of the traffic environment and for information on crash avoidance. The revised system will be in place by January 1988.

The newest of our accident data analysis systems relies on police accident records compiled by the States. Work continues on the development and refinement of this Crash Avoidance Research Data File (CARDFile). CARDFile now uses the three most recent years of data from six States. The file consists of nearly four million police accident records. At this conference we will report on the use of CARDFile in our crash avoidance research program.

Crashworthiness Research

Since Oxford, NHTSA's crashworthiness research has concentrated on the mitigation of injuries sus-

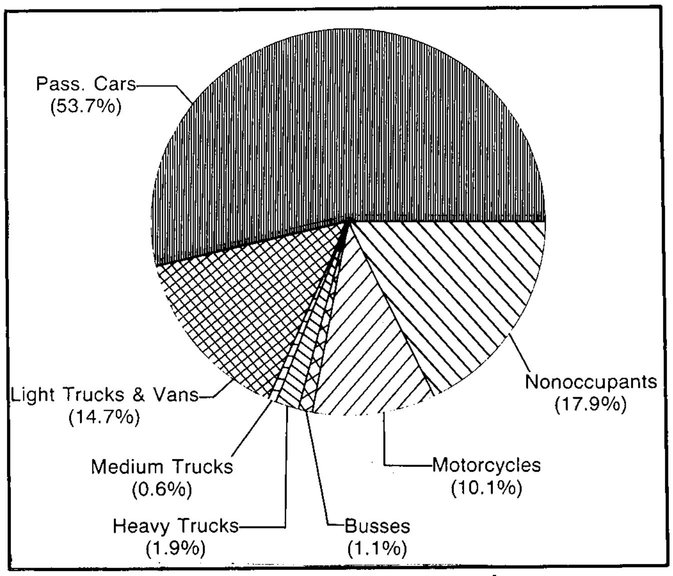


Figure 3. 1980 Traffic Fatalities (percent)

tained in side and frontal crashes for restrained and unrestrained vehicle occupants and on increasing pedestrian protection.

Side Protection

Since 1985, full vehicle side impact crash testing procedures and injury criteria have been refined; and detailed procedures for side impact testing have been published.

An extensive experimental crash test program was conducted to evaluate a variety of production (baseline) and modified cars. In addition, research into alternate test procedures using head and thorax component devices is currently being pursued.

We have also been addressing vehicle aggressiveness in a program which assesses striking vehicle structural characteristics to determine how they affect crash survival in side-struck vehicles.

Frontal Protection

Our frontal crashworthiness research program seeks improved protection of both restrained and unrestrained vehicle occupants involved in frontal impacts. The steering assembly, dashboard, windshield, and A-pillars are the main sources of injury in frontal crashes and our research is examining potential safety improvements in these areas.

We have concentrated most of our effort on steering assemblies, where we are currently testing designs to reduce facial injuries to belted drivers; and abdominal, chest, and head injuries to unbelted drivers. We are also evaluating windshields and side glazing designed to reduce facial lacerations and ejections and we are examining the effects of padding A-pillars to reduce head injury.

Pedestrian Research

Pedestrian protection research, which addresses pedestrian upper body injury, has been progressing on schedule. Since the last conference, further analyses of the accident environment have been performed, experimental methods of simulating pedestrian head and thorax impacts against vehicle surfaces have been developed, injury criteria linking experimental dynamic responses to real-world injury have been derived, and current production vehicles have been tested to identify design features that might play a role in reducing pedestrian injury.

Biomechanics

All of our crashworthiness research is based upon an increasing understanding of the biomechanics of injury. Work continues on the investigation of injury causation, with our current emphasis on frontal impacts for both the driver and passenger.

The biomechanics research has also included the evaluation, modification, and testing of production and prototype dummies developed both by the U.S. and foreign governments as well as by private industry. The dummies were developed for both frontal and side impact crash testing. Research to develop injury criteria for head, neck, thoracic, and abdominal body regions to use in conjunction with these dummies is in progress.

Air Bag Fleet Demonstration Programs

Since 1983, NHTSA has supported the development, procurement, and evaluation of driver-side air bag equipped vehicles in State police fleets and in the Federal fleet. NHTSA financed the development and installation of retrofit air bag systems for more than 500 State police cars. These vehicles were equipped with retrofit driver air bags using conventional air bag technology. NHTSA also supported the General Services Administration (GSA) in its purchase of 5,000 1985 Ford Tempos equipped with manufacturer installed driver-side air bags.

These two fleets have accumulated an estimated 200 million miles of highway travel. These vehicles have been involved in a total of approximately 750 crashes of which 107 crashes were sufficiently severe to cause an air bag deployment. In all cases, the systems worked as expected and these fleets continue to accumulate data on air bag field performance.

We are still working on the development of a retrofit driver air bag with a self-contained, all-mechanical crash sensor. The crash testing to date has been very encouraging. This system appears capable of accurately sensing a wide variety of crash conditions. Currently, the units are undergoing environmental qualification testing.

Crash Avoidance Research

Crash Avoidance Research is being pursued in a variety of areas. We are working on lighting and visibility, handling and stability, and heavy truck research.

Lighting and Visibility

Our major work in lighting and visibility is concentrated on efforts to develop a simplified performance-based vehicle headlighting standard. Our goal is the development of a system which will reduce design restrictions without degradation of seeing distance or increased glare for oncoming drivers. We will be working to expand and upgrade the capability of the existing headlighting computer simulation assessment models. We will then be paying particular attention to enhancing the model's capability to evaluate the effects of glare and seeing distance.

We are also evaluating the Center High-Mounted Stop Light (CHMSL), a requirement of FMVSS No. 108 beginning with MY 1986, and a product of NHTSA's research program. Preliminary data indicate that the CHMSL reduces the likelihood of a rear end crash involving braking vehicles by 22 percent.

Handling and Stability

In the handling and stability area, NHTSA has developed complex test equipment for the accurate measurement of inertial and suspension properties of passenger cars, light trucks, and vans. These data will allow us to make more reliable use of computerized vehicle dynamic simulations to study the relationship between vehicle handling characteristics and crash involvement.

We are turning our attention to the examination of the handling and stability problems of light trucks and vans. We will thoroughly characterize the handling and stability of this class of vehicles during the next few years.

Heavy Duty Vehicle Research

Even though they comprise less than 4 percent of the motor vehicle fleet, heavy trucks continue to be involved in approximately 10 percent of the fatal accidents. The goal of the Heavy Duty Vehicle Safety Research Program is to improve the accident avoidance capabilities and crashworthiness of vehicles with gross vehicle weight ratings in excess of 10,000 lb. through improvements in vehicle performance and driver/vehicle interaction.

Current heavy truck research is continuing to focus on high priority crash avoidance programs aimed at improving the dynamic performance of heavy vehicles in braking and steering maneuvers. Tractor-trailer brake system compatibility problems have been studied in a joint government/industry effort and the data developed is now being utilized. Research is also being initiated to evaluate the performance of second generation anti-lock brake systems through a multi-year in-service evaluation fleet program.

Driver and Pedestrian Safety Research

While this conference is focused on the motor vehicle safety research, NHTSA does have an active program which addresses the behavioral components of traffic safety. Our major efforts continue to concentrate on impaired driving (alcohol and drugs) and on safety belt and child restraint use.

Alcohol and Drug Safety Research

The goal of the alcohol/drug safety program is to develop countermeasures which effectively reduce alcohol/drug impaired driving and related accidents.

Alcohol/drug safety research is focused on four areas:

- General Deterrence—Development of combined enforcement, public information, adjudication, and licensing programs designed to increase the public's perception of the risk of being detected if driving while intoxicated or impaired.
- Specific Deterrence—Development of programs directed at identified impaired drivers and designed to prevent future offenses.
- Prevention—Development of education, public information, and local community training programs designed to instill responsible attitudes towards alcohol/drug use and driving, and to promote cooperative action for avoiding DWI offenses.
- Intervention—Development of programs designed to identify techniques that will enable and motivate third parties (e.g., bartenders, hosts/hostesses, companions) to take action in an alcohol/drug use situation that will deter potential DWI incidents.

Safety Belt Use Research

Since the last meeting two years ago, the number of safety belt use laws has grown steadily. Twenty-seven States and the District of Columbia have enacted safety belt use laws since Secretary Dole's July 1984 occupant protection decision.

NHTSA's research is concentrating on the development of programs that will result in the most effective implementation of these laws. We learned a great deal from our participation in the 1985/86 OECD project on safety belt use laws and our current research concentrates on: (1) assessing the impact of belt use laws through the analyses of crash data and tracking belt use rates; (2) exploring the situational and demographic variables associated with nonuse of belts; and (3) field testing and evaluating both traditional and innovative techniques to achieve higher belt use rates.

Rulemaking

Much of our research has as its objective the modification of motor vehicles to enhance their safety. This is achieved through regulation and information dissemination—the two principal responsibilities of NHTSA's rulemaking organization. Since the Tenth ESV Conference, some significant regulatory actions have occurred.

Crashworthiness

In the crashworthiness area, we issued three important amendments to FMVSS No. 208. In November 1985, we amended the comfort and convenience requirements for both automatic and manual safety belts and delayed the effective date until September 1, 1986, to coincide with the effective date for installation of automatic restraints.

In July 1986, we issued a final rule adopting the Hybrid III test dummy as an alternative to the Part 572 test dummy. Manufacturers have the option of using either the Part 572 or the Hybrid III test dummy until August 31, 1991. After that date, the Hybrid III will replace the Part 572 dummy as NHTSA's means of determining a vehicle's conformance with the performance requirements of FMVSS No. 208.

Finally, on March 30, 1987, we issued an amendment to FMVSS No. 208 providing a one-car credit to a manufacturer that produces a car with a non-belt automatic restraint system for the driver and a dynamically-tested manual lap/shoulder belt for the right front passenger, until September 1, 1993. This amendment to Standard No. 208 was in response to a petition from the Ford Motor Company to extend the one-car credit beyond the phase-in period. This limited extension was granted to encourage the orderly development and production of passenger cars with full-front air bag systems.

Crash Avoidance

In the crash avoidance area, regulations have been promulgated addressing lighting, braking, and controls and displays.

In the lighting area, amendments to FMVSS No. 108 were issued to permit two additional standard replaceable light sources, the HB-3 and HB-4. Action is being initiated by the United States to have these light sources, as well as the HB-1, approved for use in Europe. We are submitting a formal proposal to WP29 to modify all appropriate Economic Commission of Europe (ECE) standards. Simultaneously, we are proceeding with the rulemaking analysis concerning possible use of the HB-2 light source in the United States.

Another amendment was issued which permits the use of a new sealed beam headlamp configuration designated as Type F. Another change to that standard allows a new simplified mounting construction for headlamps. A further rule change now permits the use of modulated headlamps on motorcycles during daylight hours to improve motorcycle safety.

In an effort to increase the daytime conspicuity of passenger automobiles, the agency has issued a notice of proposed rulemaking to allow such vehicles to be equipped with daytime running lights. This would allow vehicles produced in conformance with the proposed Canadian Motor Vehicle Safety Standard to be sold in the United States. Final action on this proposal is pending.

A major rulemaking program to reform the headlighting requirements of FMVSS No. 108 has been initiated. The objective of this program is to develop new headlighting requirements which are vehicle oriented and performance oriented, and thus reduce many of the design constraints which are imposed on manufacturers. A request for comments has been published and notices of proposed rulemaking are scheduled for this summer and next summer.

FMVSS No. 101 has been amended to remove design restrictions and permit greater flexibility in the illumination and identification of controls and displays, and to accommodate new display technologies, including sequencing and retrieving of messages.

With respect to splash and spray reduction devices for heavy trucks, the U.S. Congress has recently amended the statutory requirements. The law now requires that spray suppression devices be mandated unless there is no available technology which can significantly reduce splash and spray and significantly improve visibility of drivers. In view of this change in the law, the agency is planning to hold a public meeting this summer to examine these issues.

New Car Assessment Program (NCAP)

The rulemaking organization is also charged with providing consumers with comparative information on the crashworthiness, damageability, and repairability of motor vehicles.

Under the New Car Assessment Program (NCAP), the agency has tested a total of 211 vehicles since the program began in 1979. Since the Oxford meeting, we have experimented with the use of a Deformable Moving Barrier (DMB) in the NCAP tests. This was initiated to better illustrate the effect of vehicle mass and structure on occupant injury levels. During 1986, nine head-on impacts between a DMB and a production 1985 vehicle were run, each moving at 35 mph (a closing velocity of 70 mph). The DMB was 3,000 pounds and the mass of the nine vehicles tested in the program varied from 2,000 to 3,750 pounds. The DMB and the barrier test data for these nine vehicles are being analyzed to examine the relationships that may exist between the two crash test modes and to compare the test results with real-world accident data. The results will be available later this year.

International Harmonization

Finally, in bringing this status report to a close, it is appropriate to restate NHTSA's commitment to fostering international harmonization of motor vehicle safety standards. This policy is, of course, governed by our legal and procedural requirements and by our overriding concern that motor vehicle safety in the United States not be compromised.

Our current efforts have been directed toward brakes, lighting, and side impact protection for passenger car occupants. A great deal has transpired in each of these areas since the 10th ESV Conference.

Brakes

The effort to harmonize brake standards for passenger cars on a world-wide basis began in the early 1980's. At the last ESV Conference, we reported the issuance of an NPRM to establish a new standard, FMVSS No. 135, Passenger Car Brake Standards. Extensive comments were received from the industry, both domestic and foreign, as well as from WP-29's Group of Rapporteurs on Brakes and Running Gear (GRRF). Review and analysis of those comments and some further testing resulted in a Supplemental Notice of Proposed Rulemaking in January 1987. A nine-month-long comment period was established to permit thorough examination of the latest proposal. The Group of Experts on Construction of Vehicles (WP29) has recently agreed to convening an informal meeting of the GRRF in early July 1987 for the purpose of reviewing and commenting on this latest proposal.

Lighting

At the last ESV Conference we reported on the amendment of FMVSS No. 108 which lowered the minimum permitted mounting heights of headlamps, adopted the 19-point grid of European standards for measuring the photometrics of stop, tail, turn signal, and parking lamps, and lowered the minimum intensity of yellow rear turn signals. Following that action, we made several proposals to the ECE for further harmonization of lighting standards. Included are:

- That the photometric requirements of yellow rear turn signal lamps (ECE Regulation No. 6) be amended to increase the maximum intensity permitted.
- That the test procedures for measuring the photometrics of stop, tail, turn signal, and parking lamps be harmonized by adopting the 5 zone alternative test of FMVSS No. 108. This would involve the amendment of ECE Regulation No. 7, Red Rear Lights and Stop Lights.
- That the ECE develop and issue a new regulation that would permit the installation of center high-mounted stoplamps. This proposed new regulation would be based upon FMVSS No. 108 requirements for such lamps.
- In its efforts to simplify FMVSS No. 108 and at the same time to achieve wider harmonization of headlamp requirements, the NHTSA has proposed that the Europeans and Japanese join with us in agreeing on a harmonized passing beam pattern for headlamps. Such agreement would lead to simplification of headlamp standards and make such standards more performance oriented.
- With regard to replaceable bulb headlamps, the U.S. delegate to the Group of Rapporteurs on Lighting (GRE) distributed copies of drawings for the HB-1, HB-3, and HB-4 headlamp bulbs used in U.S. replaceable bulb headlamps at the 16th session of that group in November 1986. This was done to alert the GRE of a future U.S. proposal that the ECE modify its replaceable bulb headlamp regulations to permit such bulbs. The NHTSA has submitted that proposal to WP-29 for consideration at its 82nd session in June 1987.

Side Impact Protection

NHTSA, the ECE and the Common Market have all been actively engaged in the subject of side impact protection for occupants of passenger cars for several years. The ECE, through its Group of Rapporteurs on Crashworthiness (GRCS), has drafted a proposed regulation, but important provisions dealing with injury criteria, test dummies, and movable deformable barrier have yet to be specified. The Common Market (EEC) has been working on a deformable barrier and a dummy for use in such a regulation. NHTSA has performed a large body of research on the same topics and the automobile industry, both foreign and domestic, has expended resources toward the same objective—arriving at practicable and reasonable requirements to provide better protection to occupants in the event of a lateral collision.

To further the possibility of achieving at least a harmonized test procedure for U.S. and European standards in this area, NHTSA conducted a public meeting in Washington in May 1986, to discuss the various facets of this work. Representatives of European governments and manufacturers, Japanese manufacturers, and U.S. manufacturers participated. The questions of the dummy to be used, the injury criteria to be applied, and the movable deformable barrier to be used in a systems test were discussed during the two-day meeting.

NHTSA has performed some testing of a prototype EUROSID and the results will be published later this year. NHTSA is awaiting delivery of two additional EUROSID dummies, modified according to the latest European design, for further testing. Finally, NHTSA still wants to test an agreed upon European barrier using NHTSA's current procedures as a means of comparing the performance of the various barriers.

Section 3
Results of the International Experimental Safety Vehicle Program

Chairman: Michael M. Finkelstein, United States

Panel One: *ESV/RSV Original Goals and Objectives*

Volkswagen's Participation at ESV Conferences

Prof. Dr. Ulrich W. Seiffert,
Volkswagen AG,
Federal Republic of Germany

Beginning with the first ESV Conference 1971 in Paris under CCMS-sponsorship, the idea of an international program for increased vehicle safety was initiated. North America, Europe and Japan reached a consensus for international cooperation. Because of government activities, the efforts of research institutes and work by the automobile companies and suppliers, the spirit of 1971 is still alive. Very often the question is raised, whether ESV conferences should continue to be held. In my very personal opinion, the success of the conferences can be seen in the existence of a forum which permits an open and progressive discussion of all questions associated with automotive safety. This sometimes is more important than the development and construction of demonstration prototype vehicles. Although the time sequence of the ESV conferences should be defined in relation to research results.

Volkswagen like other automobile manufacturers has actively participated in the ESV Programme. The following short descriptions and pictures show the highlights and some of the results of the different projects:

ESVW I, ESVW II, IRVW I-III, M.I.V.

The various cars or concepts also demonstrate the different priorities in vehicle safety research. In the beginning the 50 mph crash test led to unrealistically heavy vehicles characterized by excessive costs and fuel consumption. Although the vehicles made by some manufacturers more closely approximated the vehicle population then in production, benefit-cost ratios, effectiveness of special features, accident analysis, economy, recycling, compatibility, increase in comfort and active safety exerted considerable influence on the design of production vehicles.

Volkswagen, with the ESVW II and RSV research projects showed alternatives to the results achieved for the ESVW I-vehicle.

At the same time, the first components such as the passive seat belt system (VW-RA) and structural reinforcements became available on production cars.

The most important work with respect to production vehicles were the IRVW I to III vehicles ("Integrated Research Volkswagen"). In addition to questions of vehicle safety, fuel economy, noise and exhaust emissions were also optimized.

Along with the modified concept cars, the ESV-Conference changed its content. Increasingly, questions related to overall traffic safety and the results of vehicle research and not just issues related to passive safety became the subject of valuable discussions. Included also were accident analysis and legislative questions.

At the last conference in Oxford, the question of side impact had a high priority. Volkswagen has worked together with NHTSA on the M.I.V. Project to optimize two contradictory design considerations—the greatest possible reduction in dummy loadings at the lowest possible vehicle weight increase with the precondition that the design be suited to mass production. NHTSA's design goals were the 35 mph head-on fixed barrier impact and the 30 mph side impact with the new deformable crabbed barrier and the new HSRI side impact dummy.

There again it could be demonstrated that not individual requirements, but overall accident performance must be taken into consideration. Based on the M.I.V. work and results of considerable previous research, we are able to present now a new test procedure concept for the evaluation of the performance of passenger cars involved in side impacts.

This new COMPOSITE Lateral Test Procedure combines the advantages of component and full-scale tests without accepting their respective disadvantages. It consequently pursues the two principal features of a meaningful assessment of lateral protection,

- structural integrity
- compartment padding.

In principle, the test procedure consists of the following 4 steps:

Step I

Quasi static structure test by means of a deformable element such as the face bar of a moving barrier. The stroke of the ram is temporarily stopped after contact of interior door panel with the front seat.

Step II

Occupant/door interaction by application of a simplified human torso surrogate to the interior side door structure.

Step III

Continuation of the deformation of the exterior side door structure up to the point of equivalent energy dissipation according to a 30 mph deformable barrier impact.

Step IV

Computer simulation of a full-scale impact test on, e.g. PC.

Input data: Force/Deflection characteristics measured in the structure and padding tests, Steps I to III.

Results: Computed dummy loads and possibly prediction of injury severities.

This new COMPOSITE Lateral Test Procedure provides a reproducible assessment of vehicle performance as well as occupant loads without complicated, expensive and time consuming test methods. It avoids test dummies in the process of development, the most critical element for compliance verification.

Another aspect to be considered is the question of accident avoidance. A wide variety of research has been performed to date in this field. Increased comfort and visibility, handling characteristics, reliability, antiskid braking and fourwheel drive concepts have positively influenced vehicle performance. In addition, progress in accident avoidance characteristics of today's vehicles has led to new research to further reduce the probability of accidents. Most European vehicle manufacturers and many universities and research institutes are participating in the research project PROMETHEUS (Program for a European Traffic with Highest Efficiency and Unprecedented Safety).

The objective of this 8 year research program is to find a global and integrated solution for the total highway traffic system. The highway traffic of the future will become cooperative, conflict free and more compatible. In detail this means enhanced safety, improvement of environmental compatibility, minimizing energy consumption and increased comfort for the individual. To achieve the objective, more intelligence must be installed in future vehicles by means of microelectronics and AI methods so that the vehicle will no longer be isolated in the traffic flow, but will communicate with nearby cars and the environmental infrastructure by means of new communication networks. These new technologies will help the driver avoid accidents and assist him in critical situations. In this manner many human deficiencies may be eliminated although the driver must always have the final responsibility for his car and the ultimate decision made.

While this European project deals with global aspects of traffic including the car as only one factor, future tasks for this type of conference should nevertheless continue to concentrate on the side impact, pedestrian accidents, motorcycle safety and investigation of accident avoidance. Specifically in the field of the interrelationship between man, vehicle, road and environment many unknowns must be explored. The ESV Conference is an excellent forum for the international discussion in these important fields.

Consideration might be given to extending the interval between conferences to more than two years to permit greater progress to be achieved and demonstrated.

ESVW I (1972)

Passive Safety: ○ Crashworthiness in tests:
- 50 mph frontal fixed barrier
- 75 mph head-on car-to-car
- 30 mph car-to-car side impact
- 40 mph rear-end car-to-car
- pole, roof and bumper tests

○ First passive restraint system and preloader (shoulder and knee belt)
○ Benefit/cost analyses of standards

Active Safety: ○ Anti skid system
○ "Silent co-pilot"

Engine and Transmission: ○ 4-cylinder air-cooled rear engine (100 DIN hp)
○ Automatic gear box
○ 30 - 70 mph/ 16,3 sec

Emissions: ○ Complies with 1973 US standards

VOLKSWAGEN Research

ESVW II (1974)

Passive Safety: ○ Crashworthiness in tests:
- 40 mph frontal fixed barrier
- 60 mph head-on car-to-car
- 30 mph car-to-car side impact
- 30 mph rear-end car-to-car
- 43 mph rollover
- pole and bumper tests

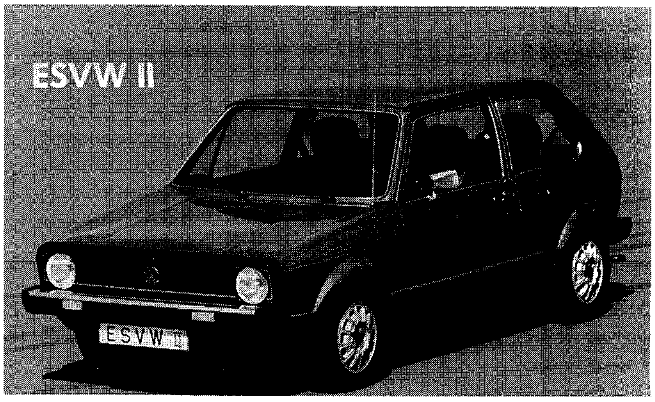
○ Passive preloaded restraint system (shoulder belt and knee bar)
○ Pedestrian protection
○ Hydraulic bumper

Active Safety: ○ Meets all US requirements

Engine and Transmission: ○ 4-cylinder water-cooled front engine (70 DIN hp)

Emissions and Fuel Economy: ○ Complies with 1973 – US standards

VOLKSWAGEN Research



IRVW 1 (1977)

Passive Safety: ○ According ESVW II

Active Safety: ○ Meets all US requirements

Engine and Transmission: ○ 4-cylinder Diesel engine turbocharged (70 DIN hp)
○ 5 gear box
○ 0 - 60 mph: 13.5 sec

Emission: ○ Exhaust:
- 0.23 g/m HC
- 0.83 g/m CO
- 0.96 g/m NO_x

○ Noise 71 dB (A)

Fuel Economy: ○

City	Highway	Composite
55 mpg	69 mpg	60 mpg

VOLKSWAGEN Research

RSVW (1975)

Passive Safety: ○ System-analyses:
- US traffic and accident projections
- Economic and automobile usage trends
- Possible safety measures supported by benefit/cost analyses
- Principle of "Consistent Conditions"
- Compatibility study

Active Safety: ○ Crash avoidance study:
- VW driving simulator
- Real vehicles
- Development of specifications
- Measures to improve active safety

Engine and Transmission: ○ Power plant system-analyses

Emissions and Fuel Economy: ○ Power plant system-analyses

VOLKSWAGEN Research

IRVW 2 (1980)

Passive Safety:
- ○ Crashworthiness in tests:
  - – 40 mph frontal fixed barrier
  - – 30 mph car-to-car side impact
  - – 30 mph rear-end barrier-to-car
- ○ Three-point belt restraint system with preloader

Active Safety:
- ○ Torsion-beam rear axle with track-correcting bearings

Engine and Transmission:
- ○ 1,3 l 4-cylinder engine (75 DIN hp):
  - – 0 - 100 km/h: 15 sec
  - – top speed: 169 km/h
  - – 3 + E gear box

Emissions:
- ○ Exhaust:
  - – 15.85 g/test CO 67 g/test
  - – 20.3 g/test HC+NO_x 20.5 g/test
- ○ Noise 73 dB (A)

Fuel Consumption:

	City	90 km/h	120 km/h	combined (1/3)
○	8	5.3	7.4	6.9 l/100 km/h

- ○ Measures:
  - – Engine-transmission managem. (fuelsaver, stop/start)
  - – High compression with knock control 13 : 1
  - – Optimized aerodynamics $c_D = 0.33$

VOLKSWAGEN Research

IRVW 3 (1984)

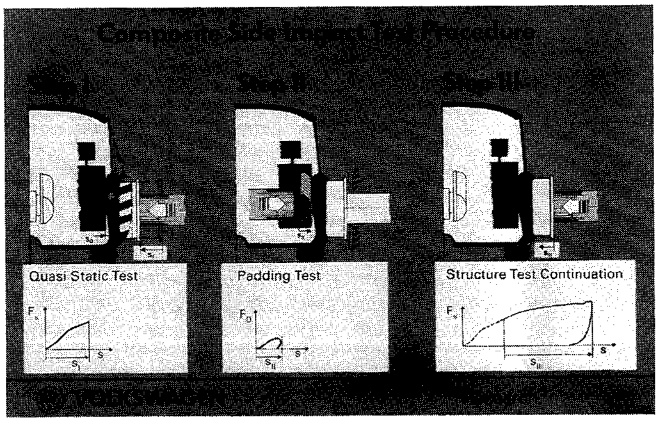
Passive Safety:
- ○ Complies with all US standards
- ○ Three-point belt with clamps at the height adjustable upper anchor point

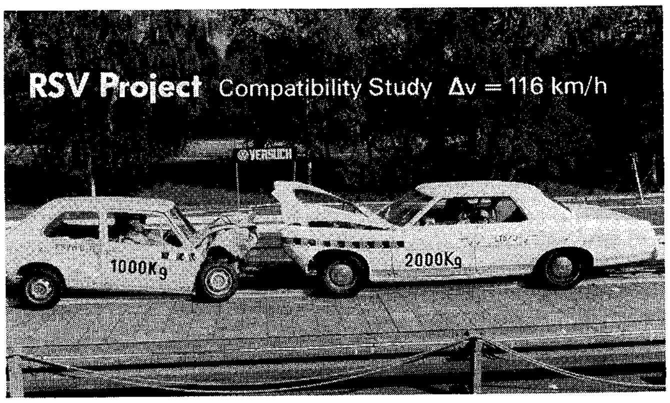
Active Safety:
- ○ Air springs at the front and rear axles, controlling ride height, comfort and drag coefficient
- ○ Load dependent damping at the rear axle
- ○ Anti skid system
- ○ Anti slip system
- ○ Navigation system
- ○ Speed dependent power steering
- ○ Tires with emergency running property

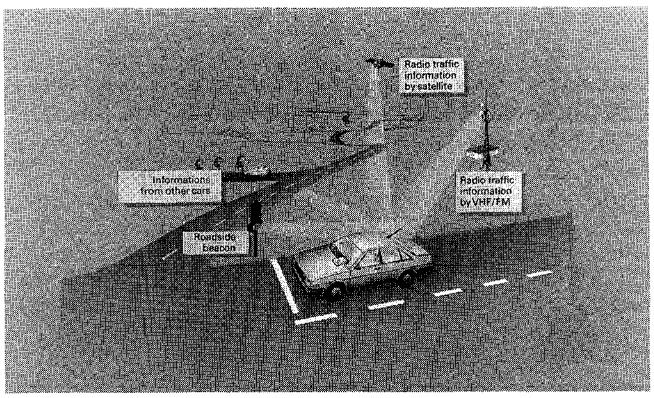
Engine and Transmission:
- ○ 1,8 l 4-cylinder engine supercharged (Digijet)
- ○ M5-A gear box: automatically shift between 4th and 5th gear
- ○ 0 - 100 km/h: 7.4 sec
- ○ Top-speed: 212 km/h (132 mph)

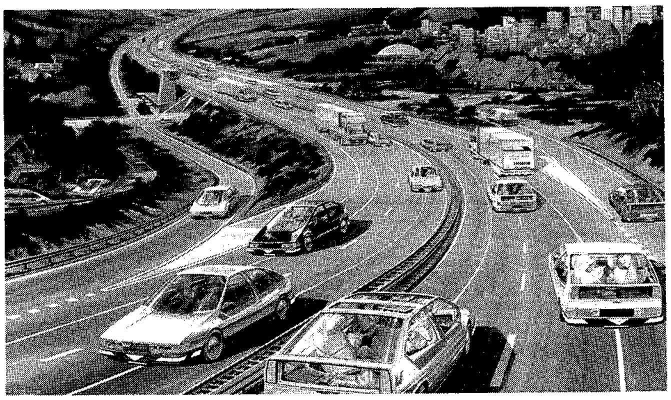
Emissions and Fuel Consumption:
- ○ Automatic radiator grille control
- ○ Complies with ECE 15/04 emission limits

VOLKSWAGEN Research









The Original Goals and History of the Nissan ESV Program

Kenichi Goto,
Japan Automobile Research Institute, Inc.
Japan

The theme I have been asked to speak on is the original goals and history of the ESV Program in Japan. I would like to address this topic by reviewing the course of our research and development work on the ESV.

In 1970, the U.S. Department of Transportation proposed that an international ESV Program be initiated for the purpose of improving vehicle safety. We were in full agreement with the aims of this program and decided to give it our complete support and cooperation. At that time in Japan, too, traffic accidents were rising sharply along with the increasing number of vehicles on the road. Therefore, we felt that it was important for countries to cooperate in carrying out research to enhance vehicle safety.

The international ESV Program can be broadly divided into the different phases. Nissan's R&D programs have progressed from our ESV, to the Nissan Safety Vehicle, to the Nissan Research Vehicle-II. At each stage we have incorporated our own ideas into the programs and have focused on the establishment of technologies for building safer vehicles. Let us take a closer look at the aims of each of the projects, the results achieved and some of the problems that were encountered.

DOT called upon countries around the world to participate in the ESV Project. In 1970, Japan became an official participant in the 2,000-pound class. The Japanese government and the Japan Automobile Manufacturers' Association worked out the specifications for Japan's ESV in reference to those of the U.S. vehicle. R&D programs were then launched by Nissan, Toyota and Honda.

Our main objective was to focus on improving safety technology for small cars in contrast to the U.S. project. Even before the first oil crisis, the Japanese automobile industry was working hard to improve the safety of small cars. At Nissan, we decided on a four-passenger ESV, weighing 2,500 pounds.

The Nissan ESV had the same fundamental aims as its U.S. counterpart, although we also incorporated our own specifications in several aspects. One of these was the condition for rear collisions. Since the moving barrier weighs 4,000 pounds, the collision speed was lowered to 40 miles-per-hour. That yielded an energy value equivalent to a collision at 50 miles-per-hour between the Nissan ESV and another vehicle of the same weight. We set the allowable safety limit for compartment intrusion at a maximum of 125 millimeters. It was felt that this would provide adequate space for occupant survival, in line with the occupant injury criteria already specified. The condition for visibility was selected in consideration of practical driving requirements.

The Nissan ESV yielded many basic safety technologies. The main features of the occupant protection system included an airbag in conjunction with three-point seat belt for the driver and an airbag in conjunction with lap belt for passengers. The periscope improved rear visibility and the urethane-covered front bumper provided better protection for pedestrians.

Our ESV project contributed greatly to the establishment of many basic safety technologies. These included simulation techniques for analyzing handling and stability properties. The mechanism through which the body construction absorbs energy was also clarified. However, the high collision target speed of 50 miles-per-hour had been achieved only by sacrificing other areas of performance. One problem, for instance, concerned the reliability of the new mechanisms developed for the airbags and preloaded seatbelt. Another problem was that the utility of the rear seat was greatly compromised by the feeling of oppression caused by the airbags. These and other drawbacks raised the question of whether an ESV vehicle was actually feasible in terms of cost effectiveness.

Upon completion of our ESV project, we began searching for the next direction to take in safety technology. That was around the time of the first oil crisis in 1973. The resulting requirements for energy and resource savings greatly increased the demand for compact cars. Subsequently, they also had a major impact on international ESV research. This led to the Research Safety Vehicle (RSV) Project, which reflected the idea of safety improvement harmonized with the three well-known E-factors—energy, economy and environmental protection—as well as aggressiveness and compatibility.

The RSV Project involved the 3,000-pound vehicle class, with a target speed of 50 miles-per-hour for frontal collisions. Nissan did not participate in this project directly, as we developed our Nissan Safety Vehicle in line with our own collision conditions, though we did refer to the RSV Project specifications.

We aimed to develop a four-passenger subcompact, weighing 2,200 pounds, which was intended for use in the 1980s. The objective of this experimental vehicle was to determine what levels of S3E performance could be achieved. A collision speed target of 40 miles-per-hour was set for the NSV. This speed was chosen in view of the cost effectiveness question which had been raised in our ESV project. Studies of accident data indicated that this speed would cover 95% of all fatalities and injuries sustained in frontal collisions in Japan and 75% of all fatalities in the United States.

One result of the NSV project was that it identified the issues involved in assuring occupant safety in subcompact cars in collisions at 40 miles-per-hour. It also yielded new insights and technical advances for reducing aggressiveness and improving other safety aspects.

The development of the first two vehicles clarified a number of issues regarding occupant protection technologies. Consequently, the focus of attention now shifted to the development of subsystems.

At the same time, the field of active safety, including the human-machine interface, came to be regarded as a key factor in building safer vehicles. In addition, the second energy crisis in 1978 saw oil prices double, causing near panic in some areas of the world. This situation made it necessary to achieve even higher levels of fuel economy. In view of these new requirements, we developed the NRV-II. In developing this 1,800 pound (850-kilogram) class vehicle our aims were to improve accident avoidance capabilities, utilize alternative fuels and achieve weight reductions, while maintaining the economy and utility of a subcompact car.

To attain the goals set for this vehicle, we made extensive use of the remarkable advances that were being achieved in electronics and composite materials at the time.

A "drive information system" and other techniques were developed to reduce the driver's workload. Research into drowsiness resulted in a drowsiness warning system. And a turbocharged methanol engine was developed to take advantage of substitute sources of energy.

Following the NRV-II, we have continued to push ahead with various programs aimed at achieving higher levels of vehicle safety. For example, our concept car, CUE-X, is a four-wheel drive vehicle, which incorporates more advanced electronic technologies, especially in the area of the human-machine interface.

Some of its technical highlights are a laser radar sysem, an electronically controlled four-wheel anti-skid system, a high-capacity, actively controlled suspension system, called HICAS, one result of four-wheel steering technology, and a satellite drive information system.

I have given you a brief outline of the aims, results and problems encountered at each stage of our ESV program. I would now like to sum up again the aims of the different projects.

First, the Nissan ESV project. This work was carried out in conjunction with the original DOT proposal. In this project we examined the technological possibilities for improving the safety of compact cars. The major objective here was to enhance occupant safety in collisions at 50 miles-per-hour.

Next, the NSV project. This project reflected the economic and social environment at the time of the first energy crisis. In this project, greater attention was given to the development of a subcompact car that would provide a practical balance of economy and utility, in addition to safety.

Then, in the 1980s, Nissan's NRV-II has been developed to meet the stronger needs for greater energy and resource savings following the second oil crisis. Electronic devices and new materials have been used extensively in this vehicle to improve safety technologies, focusing in particular on the human-machine interface. The CUE-X represents a further refinement of accident avoidance capabilities and the human-machine interface through expanded application of technological innovation.

In the process of carrying out these projects, many new advances and further refinements were achieved in safety technology. In addition, by incorporating those new developments into the experimental vehicles for evaluation, we were able to identify many of the issues involved in achieving good harmony between safety, utility and economy in cars. Solutions to those issues were then sought by shifting the focus of our work to subsystem development. A number of the new technologies that were developed in our ESV program have already been incorporated in our production vehicles, such as energy-absorbing vehicle structures, urethane bumpers, a four-wheel anti-skid braking system, and many others.

Safety issues have to be treated comprehensively in terms of three aspects: the vehicle, the driver and the environment. During this century, the automobile has become one of the most useful and convenient tools of modern society. On the negative side, however, we have the fatalities and injuries that occur in traffic accidents. It is our responsibility as 20th century citizens to minimize this negative aspect, so that we can pass on to the 21st century a more refined transportation system.

Another positive result of an integrated approach to traffic safety, including the ESV program, has been a significant reduction of traffic fatalities since 1970. In view of this achievement, I believe that the international ESV conference should be continued as a forum where representatives of government and industry from around the world can meet and exchange their experience and knowledge about automobiles and traffic systems.

Panel Two: *ESV/RSV Accomplishments*

What Was Accomplished in ESV/RSV?

Kenichi Goto,
Japan Automobile Research Institute, Inc.
Japan

I have been asked to discuss the achievements we have made in the ESV and RSV Projects. This is very difficult to answer the question. If we say that we made a great deal of accomplishments, we will be criticized unfavorably for not applying the accomplishments to new vehicles. If we say that we did not make many accomplishments, people will doubt our competence as automobile engineers and say that we wasted the taxpayers' money.

In these two projects, we aimed at ideal safety cars. We could have completed an ideal safety car in the form of a prototype vehicle, but extensive evaluations on production method and cost would have to have been made before they could be applied to a mass-production car. People will say that is why we are so slow, and we would have to answer that we are doing our best steadily.

After all, we may have to answer formally that we have made a considerable accomplishment. To my personal view, it might be too much to say that merits and demerits of the projects are just about offsetting. We have made significant accomplishments but it also is true that there were many points which should have been made in other ways.

To end the introductory remarks, I would like to first talk about the ESV and RSV Projects. According to the former DOT secretary, John A. Volpe, the experimental safety vehicle in mind is a vehicle which is filled, from front bumper to rear bumper, with maximum safety concepts such as superior driveability, better view, fire-proofness and a less-pollution engine in addition to offering passenger protection against collisions at 50 miles per hour and turnovers at 70 miles per hour. As for the ESV Project, the DOT showed specifications for each of the five items of (1) accident avoidance, (2) alleviation of injury at collisions, (3) safety after collision, (4) safety of pedestrians and (5) safety at stopping for the purpose of pursuing the ultimate limits of safety technologies. These targets were very high for that time, and I felt that it would not be easy to realize these objectives.

I examined the specifications carefully, and found that some specifications were unrealistic, insufficient, or obscure, and I thought that it could be better if they were a little more harmonious targets as a whole. Nevertheless, these targets were of great significance in the sense that they showed main directions for development.

As for ESVs in Japan, specifications proposed by DOT were partly modified considering that Japanese ESVs are small-sized vehicles. But tests on Toyota's and Nissan's ESVs revealed that both ESVs satisfied the specifications in all test items.

The RSV Project, on the other hand, aims at developing safety vehicles which meet consumers' trends, including environmental measures and effective utilization of resources and energy in addition to the performances targeted in the ESV Project with a view that they could be put into mass production in the middle of the 1980s.

It is true that these targets advanced those of the ESV Project one step forward because the targets of the ESV Project aimed only at safety causing disadvantages in the area of practicality. It is regrettable, however, that among the five companies that participated in Phase I, most of them kept paying efforts for measures against collisions and only two of them employed measures for avoiding accidents.

Two companies participated in Phase IV. When actually testing such cars, I found that they were yet to meet the targets in many points, and I felt that their level of completion was rather low. Some cars were good in individual performances, but lacked the balance in overall performance as cars for mass-production.

I am not quite sure about ESVs of other countries because I have not seen them personally, but I must say that the RSV cars' level of completion as commercial vehicles was low compared, at least, with Japanese ESVs. It would be fully worthwhile examining why they became this way while they were developed also with marketability in mind. One of the reasons I can think of is that they were also developed with much emphasis on measures against collisions, although they had a wide range of targets mentioned earlier.

Now, what were the accomplishments of the ESV-RSV Projects?

The foremost accomplishment of these projects can be that they changed the concept of car body design. When I looked at ESV specifications for the first time, I thought cars might look like tanks. I did not know the philosophy of car body design prevailing at that time in the United States and Europe. In Japan, the main emphasis in car body design was on durability. Road improvements were slow in Japan even at that time, and demands were high for vehicles which were equipped to run on bad roads. As roads improved gradually, more and more users demanded higher performance, and it was just about the time when the design of cars for lighter weight began. Lighter cars were examined, however, only for improving their durability.

The ESV Project was made public just about this time, and various discussions were made on how we could manage the rigidity of car body to satisfy the specifications. The answer was the development of a car body which had a structure for absorbing impact energies. It is a great accomplishment of ESV that the concept of collision safety has been implemented in the car body design.

Another field of accomplishment is the development of simulation technology. I said earlier that there was too much emphasis on measures against collisions, but it was not too bad for simulation technology because the development of simulation technology owes much to studies on collisions. Studies by simulation were carried out not only on car body rigidity but also on behaviors of passengers.

The survival space of small-sized cars, like Japanese cars, is small from the outset, and the permissible ranges for arrangements of the dash board, steering wheel and seats and others are limited, and their examination by simulation was very useful. The simulation technology is utilized widely also for design of current production vehicles.

As for the air bag, there were many difficulties in its development, but it was impressive to see that air bags for RSV were much better in reliability than those for ESV. As a result, air bags are being applied to production cars in other countries. In Japan, air bags were not usable because of the Explosives Control Act. The law has been revised since then and it is now possible to use air bags for cars from last year. I think that there will be production cars with air bags soon in Japan.

Seat-belts have been developed to meet the comfortability requirement of RSV, and seat-belts with ELR were developed. The use of seat-belts with ELR from 1985 was stipulated by law also in Japan, and this is the only law which implemented the accomplishments of ESV.

The development of anti-skid brakes was going on before the ESV project started, but it can be thought that the anti-skid brake is one of those of which development was accelerated by the ESV-RSV Projects. The number of cases where anti-skid brakes are used in production cars is increasing.

I have described those which appear like accomplishments, but Japanese automobile manufacturers are developing experimental safety cars by themselves even after the RSV Project. Toyota ESV III and Nissan NRV II, for example, were exhibited at the ESV International Conference held in Kyoto.

There is a trend of continued effort for safety measures, such as Project 2000 of the West German government, for developing prototype safety vehicles, and such a trend can be said to be one of the accomplishments of the ESV Conference.

Before ending my talk, I would like to add a few words. That is, it is 17 years now since the ESV project began; seven years have passed since the evaluation of RSV ended.

This may not be the time to discuss the old past problem of what the ESV-RSV was. I think that the discussions to be made in Part 3 following our session are far more important. When we have this conference next time, we should have ample time for discussion on what we should do in the future for safety problems of motor vehicles.

The discussions we had this time could be meaningful still as a review on ESV-RSV to conclude the age of ESV-RSV.

Panel Three: *Future Directions in Advancing the State of the Art in Motor Vehicle Safety*

Statement by

Bertil Aldman,
Chalmers University of Technology,
Sweden

It is indeed a privilege for an old academic to be invited to participate in this panel which turns to the future. The two other panels in this morning session have evaluated the past and summarized the present situation. Before I begin to discuss what I think could be an operative program for the future, I would like to make some short remarks about the value of what has been achieved so far.

There is no doubt in my mind that the ESV/RSV program, as it has developed over these years, has been very successful in at least two respects:

- first, it has brought together people from government, industry and research and made them work together towards a common goal: to save lives. This in itself is an achievement of great importance.
- second, it has influenced the design and construction of production cars, which are now clearly safer to the motoring public than fifteen years ago or before this program started.

When we now turn to the future we have therefore a sound basis from which to start and a great challenge to go further from these higher levels of knowledge and performance.

One problem which immediately comes to mind is whether the total vehicle development concept is viable in today's research activities. I believe it is, and would like to explain why I have come to this conclusion.

What has been learnt in several studies during these years is that the kinematics of the entire car and its occupant as well as several separate car structures influence the injury producing process. A restraint system is therefore not only an airbag, a three-point belt, or a head rest. The function of these components is greatly influenced by the construction of the seat, the floor, the steering assembly, and other car structures as well.

In our efforts to reduce the severity of accidental injuries the total vehicle development concept will have to be retained because of the complexity of the situations in which the injuries occur.

In this context I would also like to comment shortly on the place of the full-scale vehicle development in a safety program of this kind. For mainly the same reasons as I mentioned earlier I believe that there is a need for this. The fact that most automobile manufacturers present their own concept cars at intervals seems to point in the same direction. But perhaps it will not be necessary or even desirable to duplicate in the ESV research efforts what is already being done by the car manufacturers. However, some kind of coordinated effort may be needed to enhance the safety of all road users.

It is of course necessary to continue the transfer of technical knowledge into safer means of road transport, which if not started was anyhow catalyzed by the ESV/RSV program.

However, there is one link in this transfer, in which almost all of us here have some experience, but which has not really been scientificly studied or developed in this particular context. I am thinking of the process of transforming wishes, desires or formally expressed plans into standard requirements and rulemaking.

Normally, a group of people will undertake to look at one particular component or one particular crash mode and try to come up with the best possible standard and test procedure for this purpose. Such groups of well qualified people meet in different parts of the world in different organizations and make a marvelous job of producing perfect test methods, standards, and rules for their particular area of interest.

This is the way in which we have managed to create the conditions necessary for producing better and safer motor vehicles. But at the same time, this process is such that it does not automatically lead to an optimal solution in a total vehicle concept. I think nobody will deny that the imperfections of this system have also created problems. When we consider cars being produced for a world market, it seems unnecessarily difficult and costly that these vehicles should have to comply with a large number of different and sometimes conflicting rules and standard requirements.

It has become popular recently to raise one's voice and cry for international harmonization of standards. But, while harmony is a word with a nice ring to it, harmonization to almost everyone means a kind of bargaining which results in giving up something he feels important to his product, his country, or whatever. The reason for this is partly that a lot of work has been done to produce these standards in the first place and partly because most people think of this as such a complex problem that one can only deal with one standard or rule at the time. I am not going to join the group that cries for harmonization of the present standards on an international level now, but would like to propose a different approach to this problem.

Imagine as a first step a research program aiming at the creation of an optimal combination of rules and requirements in which all the different accident situations would simultaneously be taken into consideration.

In a second step the results from this endeavor could be compared with the present set of standard, rules and test procedures. The aim would be to assess the total, combined effects of the differences between this and the existing individual requirements, which constitutes the problems we have to cope with today.

A third step could then be to discuss the feasibility of globally substituting the new set of standard requirements for the present set of national rules.

In a program of this kind it would probably be necessary first to define a set of models, which eventually could be combined to simulate the entire system: the motor vehicle and its occupants, different driving conditions and traffic environments as well as other road users. The models would have to be designed to accept data about all kinds of accident modes, even some odd ones; their respective frequencies, and severities would be used with an appropriate weighting factor to simulate conditions in different parts of the world.

Much of this exists of course already but would probably have to be capable of being combined and used for this particular purpose in a systems approach to this problem. Theoretical studies in this field would probably also have to be complemented by practical tests using mechanical and biological models and eventually full scale dummies and cars.

An approach of this kind could be seen as a logical continuation of the present ESV/RSV program as it would address some of the problems which have surfaced in the process and would demonstrate what degree of safety, under all conditions and to all road users, it would be possible to build into production vehicles on the basis of the current state of knowledge.

Statement by

Georges Dobias,
Institut National de Recherche sur les Transports et leur Sécurité (INRETS), France

It is a difficult task to look to the long term future. First, I would like to make some short remarks on the work already done since 17 years.

A lot of excellent work has been done and the technical results made by car manufacturers are pretty good; with the improvement of the drivers' behaviour, it explains the evolution of safety data.

Some researches, in course, have to be completed, that is protection for side impact, protection for pedestrians, protection for small children. The most evident protections are behind us, except for heavy duty vehicles. The number of these vehicles is growing rapidly, together with their speed, their weights and dimensions. It is a worrying problem and new solutions have to be found to reduce the unsafe effects.

I must remind you that road safety works like a complex system; the car is only a part of this system between the driver, the other vehicles and the infrastructures. It is not sufficient to cope with the cars to improve the whole system.

I see three main changes for the future:

- First, the most evident problems have already scientific solutions, even if these solutions are not yet adopted in normal production.
- Secondly, the large introduction of electronics and we may expect, from it, an improvement of the safety. The PROMETHEUS Project, of the European car manufacturers, can improve the driver's behaviour—by aids—, the safety maintenance of the car, the driving on infrastructures—in bad weather, for example, and the mixing of the car in the general traffic—for example, to appreciate the safety distance.
  The normal effect may be a reduction of the number of collisions and, also, the speed of occurrence of the crashes. This change in primary safety should also affect the types of crashes and change also the types and priorities of secondary safety.
- Thirdly, the construction of cars will take more composite materials, which may decrease the aggressivity of the cars; the project CARMAT 2000, initiated by Peugeot SA, will precise the effects of the new components on safety.

As a consequence of the changes, it seems to be necessary to develop cost-benefits analysis to determine the new kinds of types of accidents and set up new priorities.

New tools will be developed, by mathematical simulation, first in biomechanics as said by Mr. Aldman, secondly in car stability as already said by Mr. Goto and Mr. Frig, but also in ergonomics to insure that the electronic aids given to the drivers will be used by them in the sense of safety.

But, I must also express my worries about the increase of the speeds measured on the roads and the speed limits of the new cars produced. If this progression is going too fast, the gains in safety may be less than expected.

All this work needs a closer international cooperation and the ESV meetings will have a more important role to play in the future.

Toyota ESV and Safety Development

Yutaka Kondoh,
Toyota Motor Corporation,
Japan

We developed Toyota ESV-1 under contract with the Japanese Government. I would like to review our R&D results regarding the car from the viewpoint of today, 13 years after its development.

Toyota ESV-1 was designed as a 2,000 lb.-class compact 2-seater touring sedan. It was not a modified version of a production vehicle, but was of a totally new design. In designing this model, the latest technology at that time was adopted, apart from a conventional design concept. Both front and rear windows were designed for full front and rear views. Large, isolator-type energy-absorption bumpers were installed on the front and back. The wide tread and low gravity center balance this car securely. Large rear combination lamps contribute to better visual perception.

Toyota ESV-1, designated as an experimental vehicle in quest of an even broader technological feasibility, was developed to meet the Japanese specifications.

Overall Length: 4300mm
Overall Width: 1800mm
Loaded Height: 1360mm
Wheelbase: 2300mm
Tread (Fr.,Rr.): 1500mm
Curb Weight: 1290kg
Capacity: 2 Persons
Engine: 1588cc, 102HP

Figure 1. Dimensions of TOYOTA ESV-I

The ESV-1 was developed in about 3 years, starting from 1971.

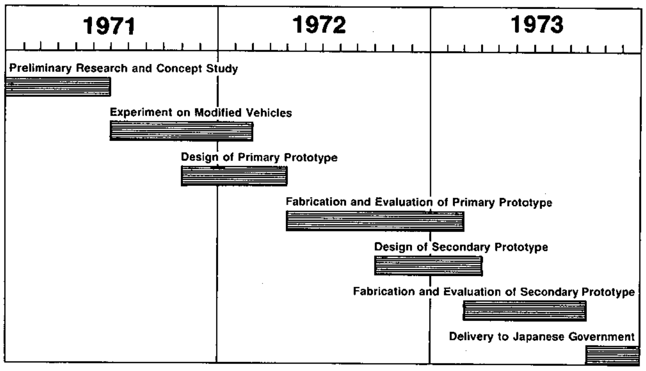


Figure 2. Time schedule for Toyota ESV-I

The new technologies adopted in ESV-1 can be seen in Figures 3-5.

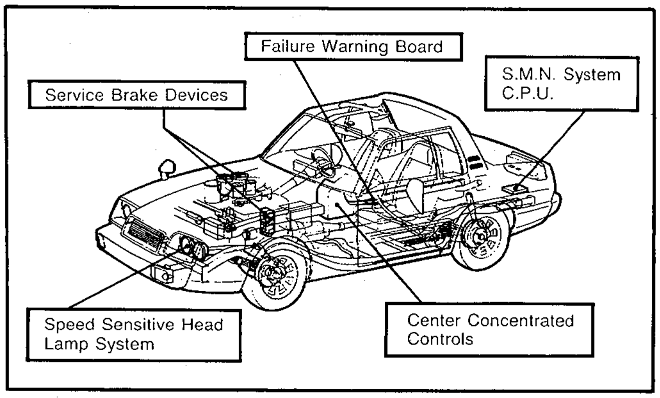


Figure 3. TOYOTA ESV-I main safety devices (accident prevention)

Accident-prevention innovations include service brake devices, a speed-sensitive headlamp system, a failure warning board, a single-wire multiplex network system, and center-concentrated controls.

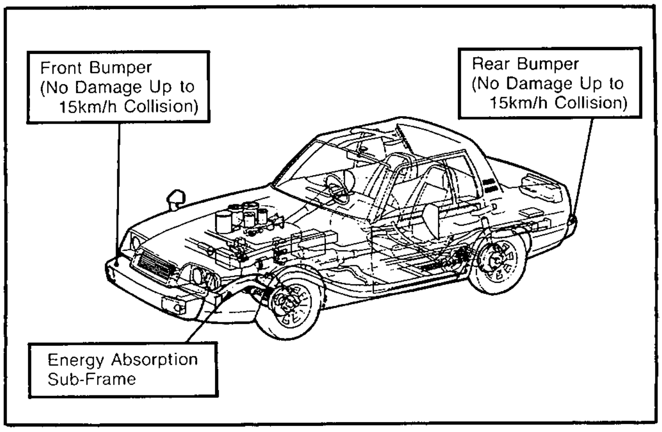


Figure 4. TOYOTA ESV-I main safety devices (crashworthiness)

Among the new technologies related to impact alleviations are the energy-absorption subframe and large, isolator-type energy absorption bumpers. These front and rear bumpers were designed to completely prevent damage in a 15 km/h fixed-barrier collision.

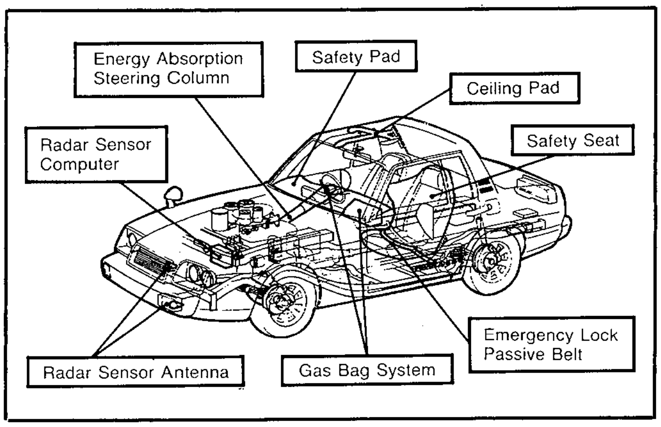


Figure 5. TOYOTA ESV-I main safety devices (occupant protection)

Major new technologies related to occupant protection are a radad sensore gas bag system, passive lap belts, and a new safety seat.

To implement the ESV-1 development project, a team of 40 engineers was organized to stay with the project from beginning to end. During the course of the project, more than 200 engineers were used. Related parts manufacturers also extended generous cooperation.

- Formation of the Project Team
- More Than 200 Persons at Peak Activity Including 40 Fully-Engaged Engineers
- Cooperation with Several Parts Manufacturers

Figure 6. Organization of TOYOTA ESV-I development

Development costs were very high, reaching 2 billion yen, or 7.5 million dollars, calculated at the currency rate at that time. One hundred experimental models were manufactured, of which 10 were delivered to the Japanese Government; some of these 10 were also evaluated by the U.S. Government.

Although the ESV-1 attained almost all of the performance targets created for it, the vehicle was excessively heavy. This meant that its aggressiveness would increase, it could not be produced efficiently, it would be very expensive to buy, even if mass produced. The ambiguity of its performance evaluation for new technology also posed a serious problem.

In restrospect, I must admit that very little safety technology from the ESV-1 was introduced into our production vehicles, in spite of the tremendous amount of resources, such as manpower, money and facilities spent in this project. This result is considered to have been due to the strenuous pursuit of technological feasibility without paying sufficient attention to public acceptance. Without fully considering cost/benefit, harmony with society and primary purpose of automobiles, we placed too much emphasis on the development of a vehicle whose purpose was occupant safety in high-velocity collisions.

- Tremendous Amount of Resources Such as Manpower, Money and Facilities Were Spent in This Project
- Very Little Technology Was Introduced into Production Vehicles due to Lack of Public Acceptance

Figure 7. Conclusion of TOYOTA ESV-I development

Toyota believes that accident prevention and occupant protection technology should be developed in view of the vehicle's role in these three factors: Human, Environmental, and Vehicle. Under this safety policy, Toyota has been committed to improving automotive safety.

Accident Prevention and Occupant Protection Technology Were Developed Steadily in View of the Vehicle's Role in These 3 Factors:

Human

Environmental

Vehicle

Figure 8. TOYOTA's efforts on vehicle safety

Let me here introduce Toyota's efforts in making production vehicles ever safer.

First, improvement of vehicle performance for accident prevention. This includes efforts toward better basic performance, higher reliability, fewer exterior projections, higher controllability and driver fatigue reduction.

Improved Vehicle Performance for Accident Prevention(1)

- Basic Performance (Braking, Handling, Acceleration)
- Reliability
- Removal of Exterior Projections
- Driver Controllability
- Reduction of Driver's Fatigue

Figure 9. Results of TOYOTA's efforts on vehicle safety

Also in this category are prevention of misactuation, vehicle performance compatible with environment, easier access to information and better presentation of information.

Improved Vehicle Performance for Accident Prevention(2)

- Prevention of Misactuation
- Vehicle Performance Compatible with Environment
- Easy Access to Information
- Presentation of Information

Figure 10. Results of TOYOTA's efforts on vehicle safety

Approaches to better vehicle performance for occupant protection include integrity of the passenger compartment in collisions, as well as collision impact alleviation, removal of interior projections, prevention of occupant ejection at collision, prevention of fuel spillage and other secondary damage, and easy rescue and evacuation.

Improved Vehicle Performance for Occupant Protection

- Integrity of Passenger Compartment
- Impact Alleviation
- Removal of Interior Projections
- Prevention of Occupant Ejection
- Prevention of Secondary Damage
- Easy Rescue and Evacuation

Figure 11. Results of TOYOTA's efforts on vehicle safety

Toyota has been doing its utmost toward achieving well-balanced improvements of these aspects step-by-step with careful consideration to developing every new production model. Of course we have fabricated special experimental vehicles as needed for use in assessing new technologies.

These efforts have already been realized in the Toyota vehicles now readily available. For example, you can easily see the 4-wheel Anti-Lock Brake System in the Toyota Supra, the Electronically Controlled Transmission in the Cressida, Camry and so on, the Electronic Instrument Panel in the Supra, Cressida and Camry and the world's first electrically motorized automatic belt system in the Cressida and Camry for the U.S. market.

Toyota has been and will continue to make steady vehicle safety improvement efforts as our responsibility to society. We should continue to make such R&D effort without compromising the primary purpose of the vehicles, apart from an ESV project. We should obtain more information about human tolerance and accidents. This is because of the still insufficient knowledge as to the human tolerance needed to fully evaluate where the largest improvement in vehicle safety can be achieved. Regarding accident data for statistical analysis, it is our desire that automakers should be given more opportunities to participate in discussions. These points were included in the NHTSA conclusions at the 5th ESV Conference. We must review what is going on regarding these points.

We should

- Make Such R&D Efforts Without Compromising the Primary Purpose of the Vehicle.
- Obtain More Information about Human Tolerance and Accident Data.
- Promote Safe Driving and Safety Seat Belt Usage - The Most Effective Measures to Save Lives.

Figure 12. Summary

Furthermore, we should promote safe driving and safety seat belt usage to make the best of the potential which the current production vehicles have to ensure occupant safety - those are the most effective measures to save lives.

Section 4
Technical Sessions

Technical Session One
Occupant Protection for Side Impact

Chairman: Ian D. Neilson, United Kingdom

Injury Pattern and Parameters to Assess Severity for Occupants Involved in Car-to-Car Lateral Impacts

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Laboratory of Physiology and Biomechanics Associated With Peugeot SA/Renault;
Alain Patel, and **Claude Got**,
Orthopaedic Research Institute,
France

Abstract

The description of real-world lateral impacts presented here gives a distribution of injury severity by position occupied in relation to the crush area for 1189 occupants involved in car-to-car collisions.

The levels of risk incurred by front-seat occupants are examined in relation with the wearing of seatbelts and the ΔV of impacted vehicles.

We restate the method for evaluation of the "occupant-wall" velocity following detailed examination of real-world accidents.

This method was utilized on a sub-sample of 134 near-side occupants exposed to direct penetration of the impacted lateral wall.

We analyse the characteristics of the deformations observed on the one hand, and the relationship between the parameters for impact severity and injury severity on the other.

An experiment carried out confirms the hypothesis that the residual deformation capacity of certain current production doors at the moment of contact with the occupant cannot be ignored in the injury figures recorded in real-world lateral impacts.

Introduction

Important research work into lateral impact such as the production of impact dummies, mobile deformable barriers and full system and subsystem tests is being carried out in various laboratories.

This work is aimed at defining reliable reproduceable tests which give a significant improvement in the simulation of real-world lateral impacts with the best "cost-benefit" ratio possible.

Whilst the aim is the same, it seems that the ways of achieving it are different.

Grishwold(1)* draws his arguments from the wide variety of lateral impacts and causes of injury to advocate sufficiently simple reproduceable "components tests" based on the principal injury mechanisms observed in real-world situations.

Other authors(2,3) feel that, in the absence of a "car-to-car" test, a full system impact of the "Mobile Deformable Barrier-car" type gives a guarantee of unquestionable realism. The complexity and lateness in the car's production process of this type of test explains the interest in alternative procedures, called "subsystem" procedures.

These two apparently divergent approaches are both concerned with the analysis of real-world lateral impacts. A knowledge of the conditions experienced by lateral impact victims is still necessary, but this is difficult to achieve because so many parameters are involved.

The principal factors are:

- the position of the occupant in relation with the impacted area,
- the type of object contacted,

* Numbers in parentheses designate references at end of paper.

- the angle of collision and the direction of the occupant,
- the violence of the impact,

but other factors also have an appreciable influence on the potential injury severity of the occupant, such as:

- ejection,
- the wearing of a safety belt,
- the age of the occupant,
- the presence of an adjacent passenger,
- the relative structures of the elements present,
- the relative rigidities of the struck vehicle and the object.

In the analysis which follows:

- The first part is concerned with the classification of the different types of lateral impact configurations between cars and their respective levels of severity. This approach allows us in particular to evaluate the relative importance of "nearside" occupants involved who directly experience the effects of the penetration of the side wall.
- The second part provides a more detailed analysis of the above sub-sample. As well as the classical parameters for lateral impact evaluation which are "closing speed," ΔV of impacted vehicle, and maximum crush, use is made of the results obtained from utilizing the method of estimating the contact velocity between wall and occupant. This "occupant-wall" velocity, which is highly applicable to this lateral impact configuration, is the most pertinent violence parameter which can be calculated from post-impact analysis of cars. The basic parameters of this method and its value in experimental situations are briefly restated.

The risks of thoracic, abdominal or pelvic injury are studied in relation with the "occupant-wall" velocity in particular.

Description of "Car-to-Car" Lateral Impacts

The database was obtained from the multi-disciplinary research network of the "engineer-doctor" type carried out in the west of Paris by the Laboratory of Physiology and Biomechanics. It concerns at the moment 11,510 side occupants, 1747 of which have an M.AIS $\geq$ 3, in all impact situations. This survey is overrepresented by severe and fatal collisions with respect to the French average.

On the total of obstacle situations, lateral impacts represents 16% of the body accident total, but its level of severity is so high that it is responsible for 21% of the severely injured and 29% of fatalities.

Front-to-side collisions between cars are the cause of two-thirds of occupants involved in lateral impact. Front-seat occupants constitute 83.5% and 90% of all occupants involved in M.AIS $\geq$ 3 respectively (table 1) (4).

Table 1. Distribution of occupants in Car-to-Car lateral impact by degree of severity according to seat occupied.

	Occupants		
-	Front	Rear	Total
M.AIS 0-1-2	826 83%	177 90%	1,003
M.AIS 3-4-5	105 11%	11 6%	116
Fatalities	62 6%	8 4%	70
Total	993 100%	196 100%	1,189

The lower gravity coefficient (M.AIS $\geq$ 3/occupants involved) observed for rear-seat occupants (0.10) compared to front-seat occupants (0.17) is explained on the one hand by the lower frequency of ejection in rear-seat occupants (among unbelted occupants, 7% against 11% in the front seats) because of the nature of "2-door" cars (15% of cases) and on the other hand because of a bias in the statistics due to the lower occupancy levels in rear seats.

A severe lateral impact on the driver thus systematically produces a severely injured victim in the front seat, whilst in the rear seats the injury severity level is sometimes lower.

Inversely, a severe lateral impact on a line with the rear seats does not systematically produce severely injured victims because these seats are often unoccupied.

Location of occupant in relation with intrusion

It is not enough to differentiate, as often happens, the nearside occupants from the farside occupants... The degree of injury severity varies significantly between nearside occupants according to whether they are exposed to penetration of the wall on the level of the pelvis or not.

Unless this distinction is made, many studies are difficult to interpret and rarely comparable. The distribution of levels of injury severity according to the location of the occupant in relation with the impact area is given in table 2 for front-seat occupants alone, to avoid possible bias.

Overall, the number of "nearside" and "farside" occupants (504 and 489 respectively) is similar. However, severity differs significantly between nearside occupants according to whether or not they are exposed to direct penetration of the wall. The levels of severe injury (M.AIS $\geq$ 3/involved) and fatality (fatalities/involved) of occupants "with intrusion" are two and four times higher respectively than those observed for occupants "without nearside intrusion." This result fully justifies the differentiation of these two lateral impact situations.

Ejection

11% of unbelted front-seat occupants are ejected. This accounts for 25% and 29% of M.AIS $\geq$ 3 and fatalities respectively.

Ejection occurs most frequently (14%) when unbelted occupants are located on the nearside without intrusion. The opening of the door is a result of a strong force towards the exterior under the thrust of the occupant generally associated with deformation of the whole body panel.

Ejection also occurs when unbelted occupants are located on the nearside but this time with intrusion (11%).

The opening of the door is in this case the result of the bursting of the lock or hinge following the impact of the striking car.

Ejection occurs during spinning or rollover after the lateral impact.

The strength of the lock, hinges and door-opening mechanisms plays an important part in preventing ejection of unbelted occupants in lateral impacts.

The wearing of a seatbelt is clearly the most effective protection against this serious risk.

The safety belt

The rate of seatbelt wearing in front seats was 41% in our sample. It does not differ according to the location of the occupant in relation with the crush area.

The severity rate (M.AIS $\geq$ 3) for belted and unbelted occupants are 0.11 and 0.21 respectively. The effectiveness of the reduction in injury severity is 51%.

The effectiveness differs according to the location of the occupant. It is only 35% for nearside occupants with intrusion, 44% for nearside occupants without intrusion, and rises to 65% for farside occupants.

The seatbelt's main effectiveness comes in part from its preventing ejection.

If we compare the levels of injury severity observed with belted occupants and unejected unbelted occupants (figure 1), the overall effectiveness of the seatbelt still reaches 41%, due to its protection against the risk of impact with walls.

The lowest level of effectiveness was recorded for unejected nearside occupants.

Lastly, we can see that one-fifth of belted occupants are nearside occupants with intrusion, but they represent half the fatalities and severely injured belted occupants because of the severity of this impact configuration.

Risks of M.AIS $\geq$ 3 by class of $\triangle$V for impacted vehicle

The risk of severe or fatal injury reaches 0.50 above the threshold of 30 km/h ($\triangle$V) for the impacted car for unejected nearside occupants directly exposed to intrusion (figure 2).

This level of risk is only reached between 40 and 45 km/h ($\triangle$V) for other unejected occupants.

This considerable difference in level of risk fully justifies priority being given to nearside occupants exposed directly to penetration of the wall in lateral impacts.

Table 2. Distribution of level of injury severity for front-seat occupants in Car-to-Car lateral impact by location in relation with impact and intrusion.

	Location of occupant			
	nearside with intrusion	nearside without intrusion	farside	Total
M.AIS 0-1-2	155 68%	245 88%	426 87%	826
M.AIS 3-4-5	42 19%	26 9%	37 8%	105
Fatalities	28 13%	8 3%	26 5%	62
Total	225 100%	279 100%	489 100%	993

Sample of Occupants Exposed to Intrusion by Impact Severity

Characteristics of sample utilized

The classification criteria utilized in figure 3 takes account of the validity limits of the method for estimating the "occupant-wall" velocity expounded later in the report.

The sample chosen comprises 134 occupants amongst 50 severely and fatality injured (M.AIS $\geq$ 3).

The distribution of body panel impact area by injury severity (figure 4) shows that the A-pillar (considered stiff) is involved in almost half the cases (66/134). In 54 cases, the deformation is limited to the compartment and does not concern the A- and C-pillars. The low level of occupancy of the rear-seats explains why the C-pillar is only impacted in 14 cases in the sample.

The average of closing speed in this severe sample is 52 km/h for the overall occupants and 57 km/h for the M.AIS 3-4-5. For fatalities, 13 out of 20 are exposed to closing speed higher than 75 km/h(5).

The distribution of levels of injury severity by body area is given in table 3.

On 100 AIS $\geq$ 3 injuries, a predominance of injuries to the abdomen (34%) and thorax (32%) over the pelvis (16%) and the head (13%) was noted (figure 5).

The head is the only body area where severe injuries are not attributable to the penetration of the side wall, AIS $\geq$ 3 injuries to the head are the consequence of impacts against the hood of the striking vehicle in two-thirds of cases and against the window-frame in the remaining third of cases.

It can be seen that the seatbelt plays an important protection role since the percent of AIS $\geq$ 3 to the head is less by half for belted occupants than for unbelted occupants (4.7% and 9% respectively).

In conclusion, the most severely injured body areas (thorax, abdomen and pelvis) are a result of penetra-

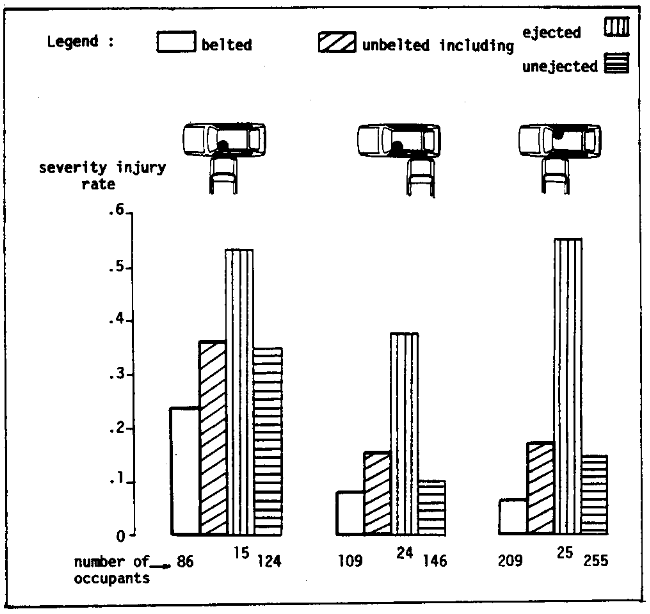


Figure 1. Severity rate (M.AIS $\geq$ 3) of front seat occupants in lateral car-to-car collisions according to belt wearing, ejection and seat location

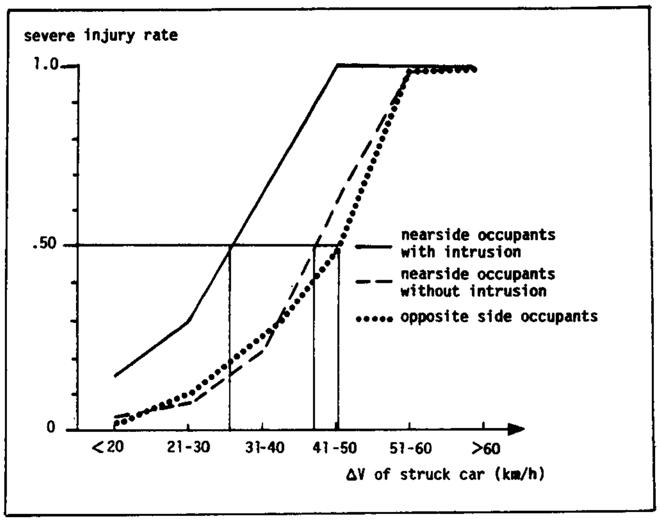


Figure 2. Severe injury rate (M.AIS $\geq$ 3) by class of $\triangle$V— Unejected occupants—Car-to-Car collisions—

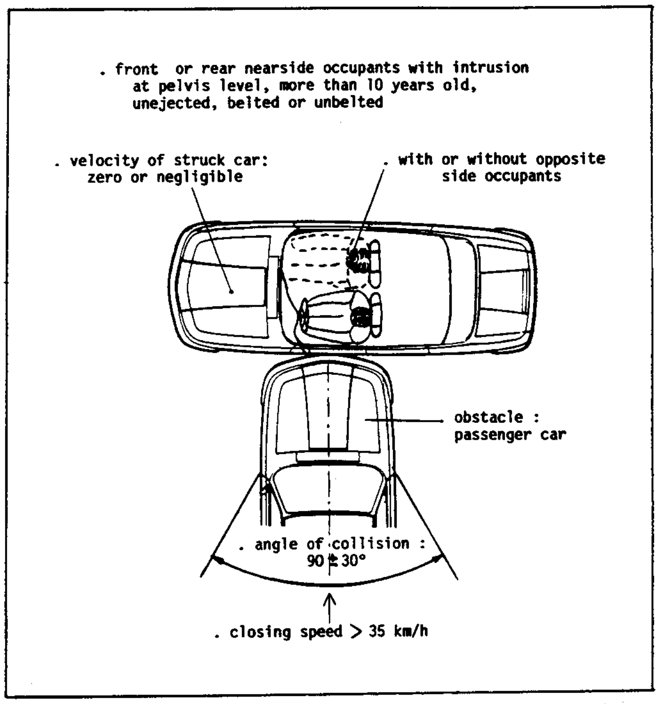


Figure 3. Description of criteria used to select the sample of nearside occupants with intrusion

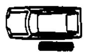
	AREAS OF DAMAGE			TOTAL
				
M.AIS 0	5	7	1	13
M.AIS 1-2	31	33	7	71
M.AIS 3-4-5	12	13	5	30
Killed	6	13	1	20
TOTAL	54	66	14	134

Figure 4. Severity of injuries (M.AIS) according to areas of damage (sample of nearside occupants with intrusion)

tion of the wall, whose velocity at the moment of contact with the occupant can be calculated.

Parameters of impact severity for body areas exposed to intrusion

The analysis of lateral impacts on the basis of experimental impacts with dummies has allowed the two main parameters which characterize the level of injury severity for occupants sustaining penetration of the wall to be defined.

1. The "occupant-wall" impact velocity for occupants sustaining a strong intrusion.
2. The rigidity of the wall impacted by the occupant.

At the present moment, only the parameter giving the "occupant-wall" velocity is calculable from the study of real-world accidents, utilizing the mathematical model conceived for this purpose by B. Loyat (6), in application since 1980 at the Laboratory of Physiology and Biomechanics(7). This model, quoted in annex 1, brings into play the masses of the vehicles present, the closing speed, their degree of deformation and their elastic restitution coefficient.

Table 3. AIS by body areas for occupants exposed to intrusion in car-to-car lateral impacts (N = 122*).

	AIS 0	AIS 1	AIS 2	AIS 3	AIS 4-5
Head	41	37	35	3	6
Neck	113	7	1	1	0
Thorax	65	24	11	6	16
Upper Members	87	26	8	1	0
Dorso-Lumbar Column	107	12	3	0	0
Pelvis	74	17	20	11	0
Abdomen	92	5	2	8	15
Lower Members	94	21	6	1	0

(* 12 unautopsied fatalities are omitted from this table)

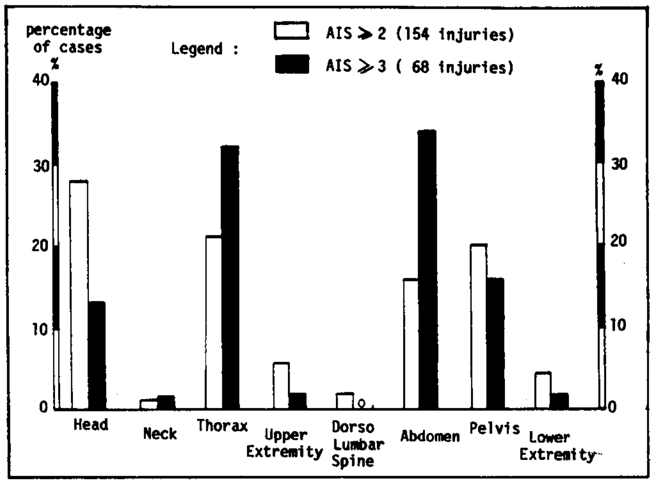


Figure 5. Distribution by body areas of AIS ≥ 2 and AIS ≥ 3 injuries for 122 nearside occupants with intrusion

The assumptions made consist mainly in supposing that the distribution of crush between striking and struck cars remains constant in time.

The Thorax

- *Injuries:*

57 of 122 occupants incurred thoracic injuries of all levels of severity. The majority of the AIS ≥ 2 injuries (27/33) are to be found on the nearside. The systematic presence of an adjacent occupant can be seen when the thoracic injuries are bi-lateral (5 cases) or on the farside to the impact (1 case) (figure 6).

The distribution of the 191 noted fractured ribs located on the nearside shows that the 5th rib is the most frequently injured (figure 7).

This median point in the height of the thoracic segment generally corresponds to the upper part of the door panel in European cars.

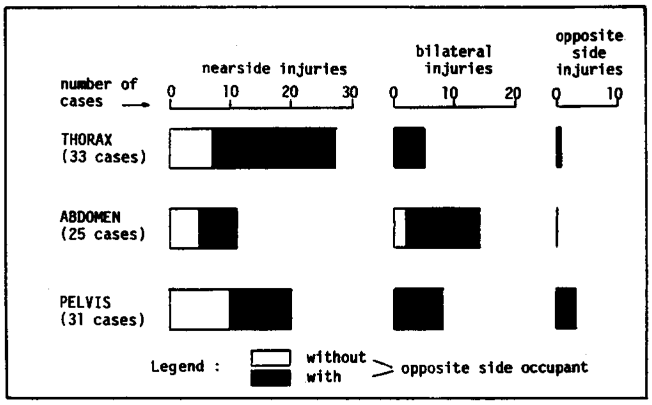


Figure 6. Location of AIS ≥ 2 injuries of torso for nearside occupants with intrusion according to presence or absence of opposite side occupant

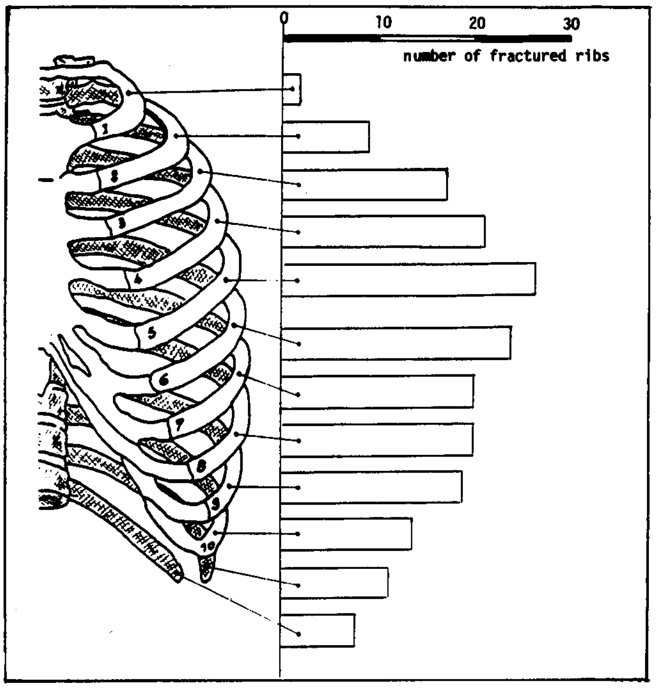


Figure 7. Location of the 191 nearside fractured ribs (39 occupants)

The assumption that there is a shearing effect between the unimpacted part of the thorax and the lower part exposed to penetration of the wall is a plausible one.

Fractures of the lower (10th to 12th) ribs are only associated with severe abdominal injuries in less than one-third of cases (9/23).

The distribution of victims with at least an AIS $\geq$ 3 value for the torso (thorax and/or abdomen and/or pelvis) shows that thoracic injuries are present in nearly two-thirds of cases (associated with other areas in 38.2%, in isolation in 26.5%) and are absent in the remaining third of cases (see figure 8).

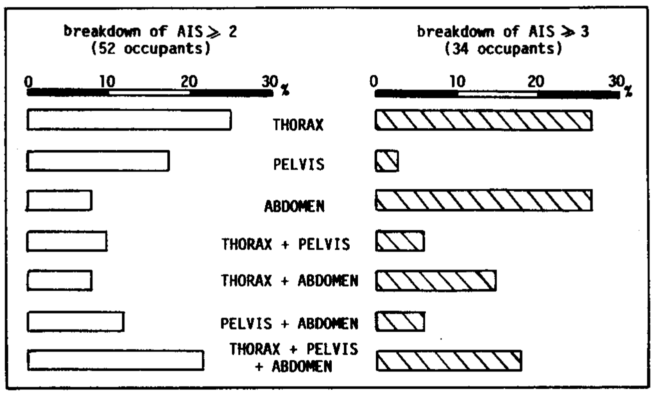


Figure 8. Breakdown (in %) of AIS $\geq$ 2 and AIS $\geq$ 3 injuries to thorax, abdomen and pelvis for nearside occupants with intrusion

It can be noted that the association of severe thoracic-abdominal-pelvic injuries is only observed in 17.6% of severe cases of injury to the torso.

On the 22 AIS Thorax $\geq$ 3, 19 cases of injuries to the bone with internal injuries such as hemothorax or pneumothorax were noted against only 2 cases of bone injuries on their own and 1 case of internal injury without rib fracture.

- *Impact severity and injury severity*

The relationship between thoracic injury severity and "thorax-wall" velocity is only significant in impacts during the course of which there was direct contact of the front of the striking vehicle on the level of the occupant's thorax (in two-thirds of cases).

In the remaining third of cases (42/122), the crushing of the upper door is attributable to dragging of the panel consecutive to the crushing of the lower part alone. In this sub-sample, the only "moderate" thoracic injury observed has an AIS 2 value, whilst the maximum values for vehicle ΔV and static crush at thorax level are 39 km/h and 500 mm respectively (see figure 9). As well as the absence of external forces at thorax level, the door panel has three very favourable attributes for thoracic protection.

1. The upper part of the door is quite thick and provides a capacity for energy dissipation on contact with the thorax.
2. The upper part of the wall is deformable towards the exterior.
3. The slope of the door panel (whose lower part is more recessed) allows the rotation and gliding of the thorax against the panel area.

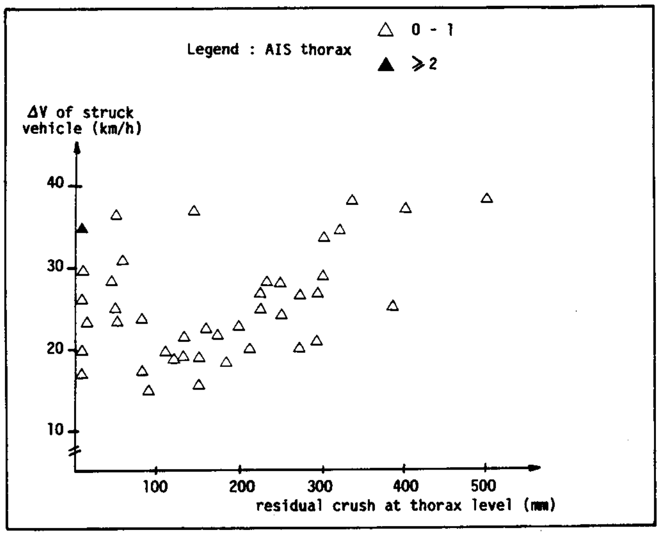


Figure 9. Severity of thoracic injuries according to ΔV of struck vehicle and crush at thorax level for the 42 occupants without direct impact at thorax level

In conclusion, the risk of thoracic injury is only observed for the 80 occupants sustaining direct contact with the striking vehicle at thorax level.

The accumulated frequency of AIS Thorax $\geq$ 2 values as a function of "thorax-wall" velocity is:

25%	33 km/h
50%	38 km/h
75%	44 km/h

The average values of closing speed and vehicle ($\triangle$V) for this sub-sample are 53 km/h and 26 km/h respectively.

The correlation between "thorax-wall" velocity and AIS-thorax is 0.58 (figure 10).

This low correlation is in virtue of the fact that the "wall-thorax" velocity alone is not enough to fully explain the risk of thoracic injury.

Other parameters such as the stiffness of the deformed wall or the difference in individual thoracic tolerance are complementary explaining factors. A knowledge of the degree of rigidity of the deformed wall is not available from analysis of real-world accidents.

These factors do however have an influence on thoracic risk in the wall velocity range of between 31-45 km/h. This segment contains 47 on the 80 involved occupants and 25 of the 32 AIS $\geq$ 2.

The parameters governing the various significant factors observed are given in table 4.

- The risk of injury is doubled when intrusion exceeds 300 mm (figure 11).
- A multiplication of the risk of AIS thorax $\geq$ 2 by 1.7 is noted when the residual thickness of the door after impact is less than or equal to 30 mm. Overall, this thickness varies in an inverse manner to the crush distance at thorax level.

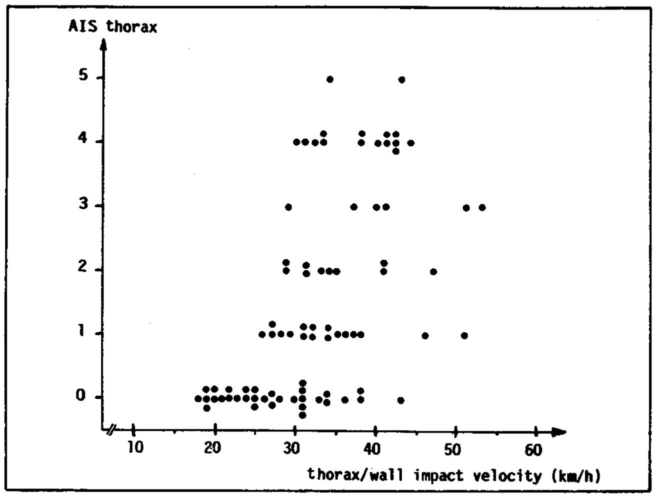


Figure 10. AIS thorax according to occupant/wall impact velocity for the 80 occupants sustaining direct thoracic impact

Table 4. Differences in risk of thoracic injury in the "wall-thorax" velocity range between 31 and 45 km/h according to static crush, residual wall thickness and age.

		Severity rate (AIS thorax ⩾ 2/ occupants)	$\chi^2_{.10}$
Static crush (thorax level)	< 300 mm	0.29 (5/17)	HS
	⩾ 300 mm	0.66 (20/30)	(6.04)
Residual thickness of wall (thorax level)	⩽ 30 mm	0.68 (15/22)	S (3.73)
	> 30 mm	0.40 (10/25)	
Age	< 40	0.39 (11/28)	HS
	⩾ 40	0.79 (15/19)	(7.20)

- Lastly, the role of age in thoracic injury risk is confirmed in the above sample. The risk of AIS thorax $\geq$ 2 is doubled when the occupants are 40 years old and over.

The Abdomen

- *Injuries*

Depending on the side of the impact, the spleen or the liver are the most frequently injured organs (14 cases of AIS $\geq$ 3).

Other injured organs on the nearside such as the diaphragm, the small intestine or the kidneys show an AIS $\geq$ 3 value in 18 cases in total. These injuries are associated with hepatic or spleen injuries in 9 cases and are in isolation in 9 cases.

No case of serious abdominal injury on the farside to the impact was observed. Conversely, the presence of an adjacent occupant would explain the frequency of bi-lateral injuries (see figure 6).

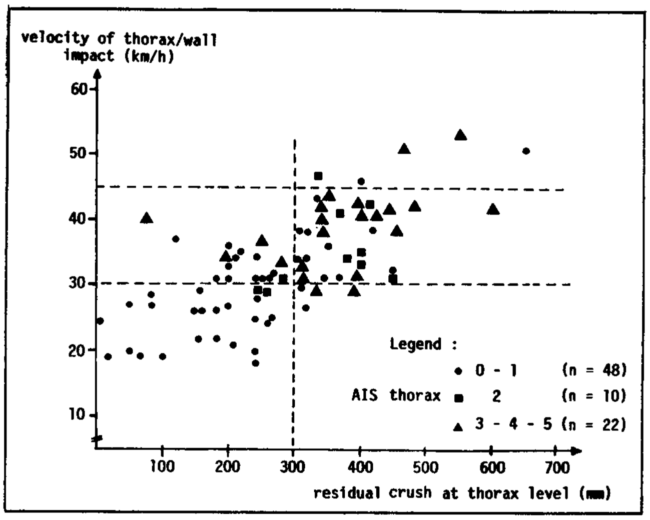


Figure 11. AIS thorax according to occupant/wall impact velocity and residual crush at thorax level

The majority of abdominal injuries (18/23) are associated with severe nearside injuries to pelvis (5 cases), to thorax (4 cases) and to thorax + pelvis (9 cases).

- *Impact severity and injury severity*

The cumulative frequency of AIS $\geq$ 3 abdominal injuries as a function of wall velocity calculated at pelvis is:

- 25% 42 km/h
- 50% 48 km/h
- 75% 54 km/h

In this sample of severe abdominal injuries, the average closing speed and vehicle ($\triangle$V) are 64 km/h and 32 km/h respectively.

These velocity levels are significantly higher than those shown for the risk of AIS thorax $\geq$ 2 in direct impacts.

21 of the 23 cases of AIS abdomen $\geq$ 3 occur when the "wall-pelvis" velocity is $\geq$ 40 km/h and the static crush at pelvis level $\geq$ 350 mm (figure 12). The risk of abdominal injury is very different on either side of these limits (2/81 and 21/41 respectively).

The influence of the risk linked to crush at the same "wall-pelvis" velocity is clear in the 40-50 km/h range where we note the absence of abdominal injury in the 8 occupants for whom the static crush at pelvis level is less than 350 mm.

An impact against armrest is highly probably for 17 out of 23 occupants sustaining severe abdominal injuries, but there is no evidence that such an impact is the only cause of injuries. All the more reason for accidentologists not to be able to correlate armrest characteristics (height, shape, stiffness) with injury severity.

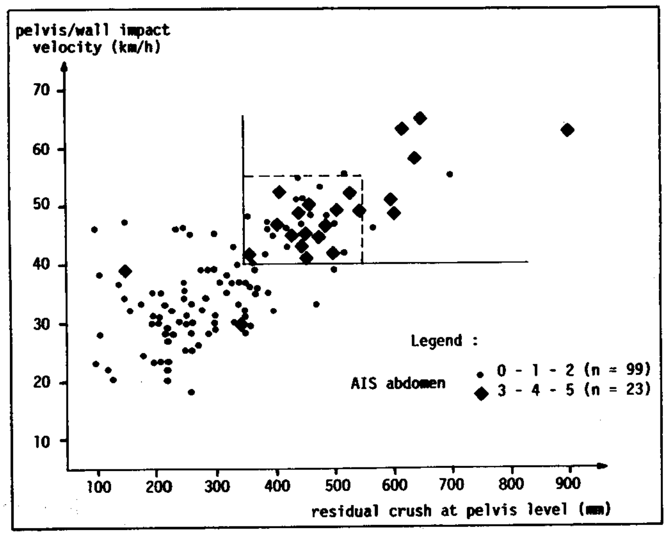


Figure 12. AIS abdomen according to occupant/wall impact velocity and residual crush at pelvis level

The role played by armrest on abdominal injuries is discussed by F. Brun-Cassan(8) referring to experimental work.

15 of the 32 occupants who sustained the same wall velocity of between 40 and 55 km/h and the same static crush of between 350 and 550 mm display AIS 3 and over values at the abdomen.

In this sub-sample the significant influence of the following parameters is noted (table 5).

This risk is doubled when the crush distances measured at thorax and pelvis level show little divergence (difference of $\leq$ 100 mm).

In these cases, the door panels are only slightly sloped towards the exterior. The possibilities for the occupant to rotate and glide against the wall are diminished compared with the case where the door is significantly more recessed in its lower part than in its upper part.

The risk is more than doubled when the pelvis is fractured.

The Pelvis

- *Injuries*

In 30 of the 31 AIS pelvis $\geq$ 2, the injuries displayed are ilium or ischium fractures. A fracture of the acetabulum in isolation is observed in the remaining case. Other types of pelvic fractures are associated in 8 cases.

- *Impact severity and injury severity*

The cumulative frequency of AIS pelvis $\geq$ 2 as a function of "wall-pelvis" velocity is:

- 25% 41 km/h
- 50% 46 km/h
- 75% 55 km/h

The average of closing speed and vehicle ($\triangle$V) are 57 and 32 km/h respectively for this sample of pelvis injuries.

These levels of impact severity are very close to those observed for the abdomen and much higher

Table 5. Difference in the abdominal injury risks according to crush distance differences measured at pelvis and thorax levels and the presence of associated pelvic injuries (sub-sample of occupants sustaining a "wall-pelvis" velocity of 40 to 55 km/h and a static crush of between 350 and 550 mm).

		Severity rate (AIS abdomen $\geq$ 3/ involved occupants)	$\chi^2_{0.10}$
Differences between "pelvis" crush and "thorax" crush (in mm)	$\leq$ 100	0.64 (9/14)	S
	> 100	0.33 (6/18)	3.03
Presence of associated pelvic fractures	yes	0.75 (9/12)	HS
	no	0.30 (6/20)	6.09

than those shown for thoracic injuries in a direct impact.

The risk of pelvic fracture grows overall in line with the "wall-pelvis" velocity. The first case is observed at 29 km/h. All the occupants injured at "wall-pelvis" velocities of over 55 km/h display a fracture to the pelvis.

The "wall-pelvis velocity" parameter is necessary but is not enough to fully explain the presence or absence of fracture. We note in particular that women show a higher level of risk than men (figure 13). The other available parameters do not give an acceptable explanation of the scatter observed.

In almost all the cases, the residual thickness of the wall is very small. This parameter was consequently unusable on a statistical level. A knowledge of the influence of the thickness of the deformed wall on the prevention of pelvic fractures cannot be usefully achieved by the study of real-world accidents.

Discussion

The analysis of thoracic, abdominal or pelvic injuries in nearside occupants sustaining direct intrusion shows that the "wall-occupant" velocity is necessary, but is not enough to fully explain the injury risks incurred.

The "stiffness of deformed wall" parameter cannot be taken into account in the analysis of real-world lateral impacts. It could be supposed that the inadequacy of shock-absorbing material in current production doors results in the total absence of energy absorption capacity during impact against occupant.

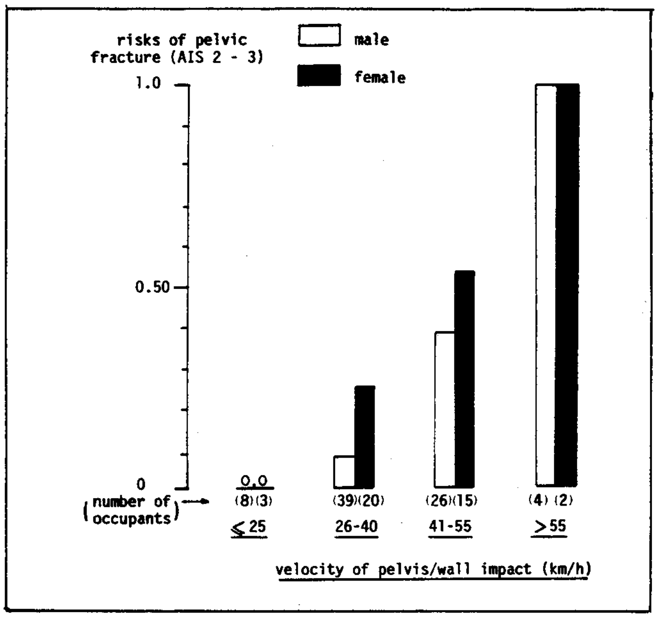


Figure 13. Pelvis fracture risks according to pelvis/wall impact velocity and sex (sex unknown for 5 M.AIS< 2)

The experimental work carried out shows that this hypothesis is incorrect.

Due to the occupant-wall impact, the door panel structure has a not negligible residual deformation capacity.

This role of shock-absorber varies in importance according to the structural design of the door panel. Figures 14 and 15 give a diagrammatic representation of the speed variation versus time of the striking mobile barrier, the interior wall of the door panel and of the dummy pelvis (APROD dummy) for lateral impacts with a CEVE-type mobile deformable barrier launched at 50 km/h and at 90° against Renault A (4-door) and Renault B (2-door). On the basis of these figures, the moment of "wall-pelvis" contact is defined by the instant t = 1, when the pelvis begins to be accelerated.

At this same moment, the velocity of the interior wall is close to that of the striking mobile barrier, or 13 m/s for vehicle A and 12,5 m/s for vehicle B.

It can be seen that although the two vehicles have a different mass and design (2- and 4-door), the "wall-pelvis" velocity is similar. After impact, we observe from the figures a variable but important decrease in the "wall $\triangle$V" of the inside surface of the door panel between the instant t = 1 and t = 2. At t = 3, the energy of the impact against the occupant is dissipated and the occupant's velocity is equal to the velocity of the interior wall. The "wall-pelvis" impact is finished. We can supposed that during the whole period t = 1 to t = 3, the velocity of the external wall of the door panel is equal to the velocity of the striking barrier.

The area formed by t = 1 and t = 3 (figures 14 and 15) gives the relative closeness between the interior wall and the exterior wall of the door panel due to pelvis intrusion into the interior wall. Although

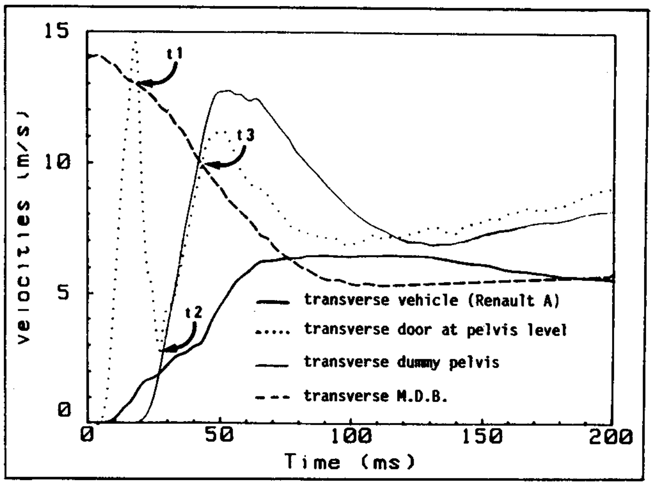


Figure 14. Velocity diagram—Mobile Deformable Barrier (M.D.B.) against the Renault A in lateral collision

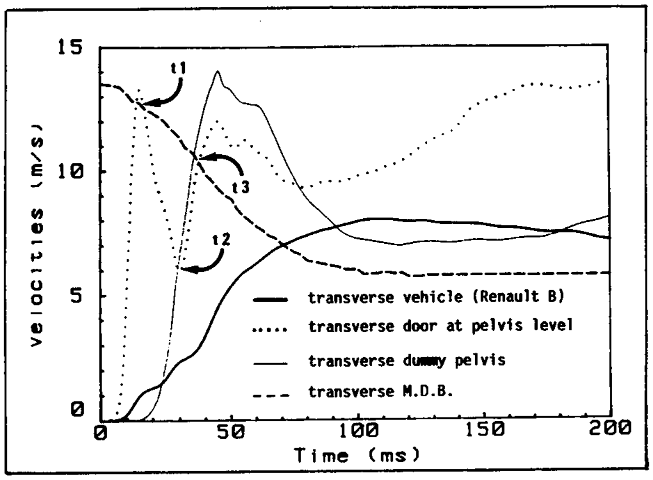


Figure 15. Velocity diagram—Mobile Deformable Barrier (M.D.B.) against the Renault B in lateral collision

during the course of the impact, the sensitive axis of the door sensor may undergo a rotation, the information obtained allows the visualization of the tendency but does not allow rigorous quantification of the phenomenon.

Figure 16 is an illustration of the different residual energy absorption capacities of these two types of vehicle obtained by superimposition of areas t = 1, t = 2 and t = 3.

This figure shows that Renault A's door panel has a more protective role for the pelvis than that of Renault B. This explains why, at a slightly higher contact velocity, the variation in transverse velocity sustained by the pelvis of the dummy located in Renault A (46 km/h) is significantly lower than sustained by the pelvis of the dummy located in Renault B (51 km/h).

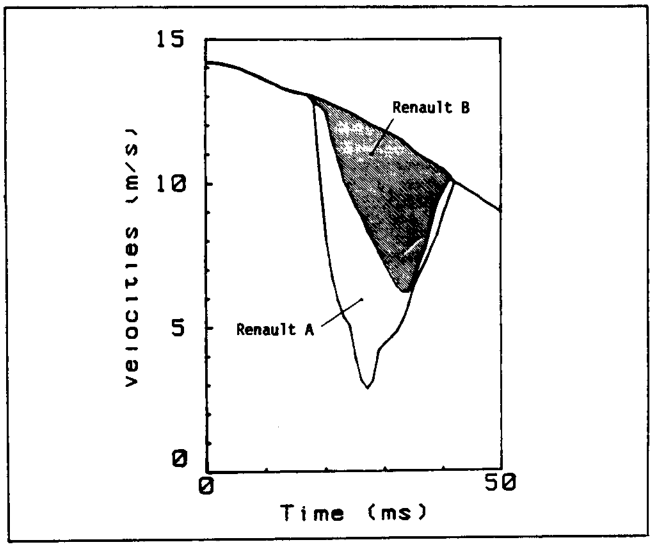


Figure 16. Illustration of energy-absorbing capabilities differences between two types of door structures as a result of dummy pelvis impact

This experiment shows that the wall velocity is not enough to account for the severity of the impact sustained by the occupant. A consideration of the impacted wall's residual energy absorption capacities is essential to the understanding of the injury risk. In this wall, the part played by the shock-absorbing material made up of the interior padding itself is hard to define.

The Laboratory of Physiology and Biomechanics was aware of the complex interaction between intrusion, wall velocity, and the wall's residual deformation volume at its interface with the occupant since starting from 1980, the method utilized for estimating the wall velocity necessitated the measurement of the door panel's residual thickness after impact (see annex 1). It would however be advantageous to complete this measurement of the wall's residual thickness by an estimation of the wall's residual stiffness at its interface with the thorax, the abdomen and the pelvis.

To be able to make further progress, we need to have a light tool available which allows the static or dynamic measurement of the force as a function of the crush. This tool could be utilized in the analysis of both real-world accidents and of experimental impacts.

Conclusion

This study of real-world "car-to-car" lateral impacts concerns the statistical importance of the configurations met and the parameters for explaining the injuries observed for nearside occupants sustaining direct intrusion.

The statistical analysis shows that

- The risk incurred by nearside occupants who do not sustain direct intrusion is slight. They represent 28% of total occupants involved in collisions between cars, but only 12% of those killed in this configuration.
- More than three-quarters of occupants involved in lateral impacts do not sustain direct lateral intrusion. They account for half the severely injured victims.
- In the case where the occupant does not sustain penetration of the wall, the seatbelt shows an effectiveness of 59% in reducing injury severity by preventing ejection and limiting the risk of impact against the walls.
- A 0.50 risk of M.AIS $\geq$ 3 is observed above 30 km/h ($\triangle$V of impacting car) for unejected nearside occupants who sustain intrusion, against a $\triangle$V of 40 to 45 km/h for other unejected occupants.

Analysis of a sample of nearside occupants sustaining direct intrusion of the front end of a car shows that:

- On 100 AIS $\geq$ 3 injuries, we observe a predominance of injuries to the abdomen (34%) and the thorax (32%) over the pelvis (16%) and the head (13%); (the remaining 5% are distributed between other body areas). Association of AIS $\geq$ 2 or $\geq$ 3 at the thorax, the abdomen and the pelvis occurs in less than a quarter of cases.
- The application of the "occupant-wall" method of calculating velocity to 122 occupants allowed the calculation of the risk of severe injuries to the thorax, the abdomen and the pelvis. This method is only applicable to the thorax in the case of direct impact of the upper part of the front end of the striking car. This is not the case in particular when the front of the vehicle has a "V-type" profile. No instance of AIS thorax $\geq$ 3 was observed in the absence of direct impact on this level by the front of the striking car.
- The "occupant/wall" velocities corresponding to the 50th-percentile of AIS $\geq$ 2 to the thorax, the abdomen and the pelvis are 38, 48 and 46 km/h respectively. Complementary parameters such as the level of the intrusion, the residual thickness of the door and age for the thorax, the slope of the door panel and the association of pelvic fracture for the abdomen, as well as the sex of the occupant for the pelvis help to explain partly the scatter observed.
- The relationship between occupant/wall velocity and severity of injury is established for cars whose interior padding is sometimes inadequate. The present condition of the door is not however prejudicial to the occupant in the absence of direct impact on the door. The scatter observed in the opposite case shows that the residual local stiffness of current door panels has an influence on the risk of injury. The experimental work carried out shows that in the present state of current upholstery of cars, the door panel may provide an energy absorbing capacity in the occupant/wall impact.
- The agressivity of front ends differs according to car model. It has an influence on thoracic risk in particular. Neglecting to test the upper part of the front end obliges one to opt for tests limited to the side parts only. The high cost of this option can only grow if the agressivity of car front ends increases with the growth in frontal impact needs. A compromise between the structural design and the stiffness of front and side ends is the only way to obtain a higher "cost-benefit" ratio.

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Appendix

Calculating the "Occupant-Wall Impact Velocity" in Real-World Car-to-Car Lateral Collisions

Assumption

Figure A is a diagrammatic representation of the development of vehicle velocities in the general case where the initial point of impact is not in a line with the occupant level. After contact between the two vehicles and crushing of the door, the wall velocity becomes steady and increases slowly to converge with the struck vehicle's $\triangle V$ at the end of the impact. This wall velocity is equal to the velocity Vi of the interface between the two vehicles, which depends on the rigidity of the two vehicles present.

If R1 and R2 are the rigidities, Vi the interface velocity, V1 and V2 the velocities of the striking and struck vehicles:

$$\frac{Vi - V2}{V1 - Vi} = \frac{R1}{R2}$$

In practise, the development of R1 and R2 in time are not known, but experience shows that we can obtain an excellent approximation by assuming them to be constant.

Assumption 1: R1 = constant, R2 = constant

In these conditions, if e1 and e2 are the crushes:

$$\frac{e1}{e2} = \frac{R1}{R2} = \text{Constant}$$

Knowing the residual crushes e1r and e2r as well as restitution coefficients E1 and E2 for each car, it is possible to calculate the interface velocity from the velocity of the vehicles.

Assumption 2: the velocity of the vehicles is linear.

Velocity of Centers of Gravity

By comparing the crashed cars with reference tests, we can estimate:

- the approach velocity Vr,
- the transverse approach velocity VT1 = Vr $\cos\alpha$, where α is the angle of incidence,
- the variation in velocity of the impacted and impacting vehicles

$$\triangle V1 = \frac{M2}{M1 + M2} .VT1 \qquad \triangle V2 = \frac{M1}{M1 + M2} .VT1$$

where M1 and M2 are the masses of the impacting and impacted vehicles.

In order to reconstruct the diagram of vehicle velocities, the time length of the impact remains to be determined.

With E1r and E2r being the crush distances at point of impact I (figure B), the maximum dynamic crush is: Ed = E1d$\cos\alpha$ + E2d = E1r (1 + E1)$\cos\alpha$ + E2r (1 + E2) where E1 and E2 are the coefficients of elastic restitution.

The time length of the impact (figure C) is:

$$T = \frac{2.Ed}{VT1}$$

Velocity of Occupant/Wall Impact

Wall velocity is equal to that of the interface between the two cars after crushing of the door. It is thus possible to calculate the velocity of occupant/wall impact by determining only this interface velocity at occupant level.

With e1r and e2r being the residual crush distances at occupant level (figure B), the dynamic crush is: e1d = e1r (1 + E1) and e2d = e2r (1 + E2)

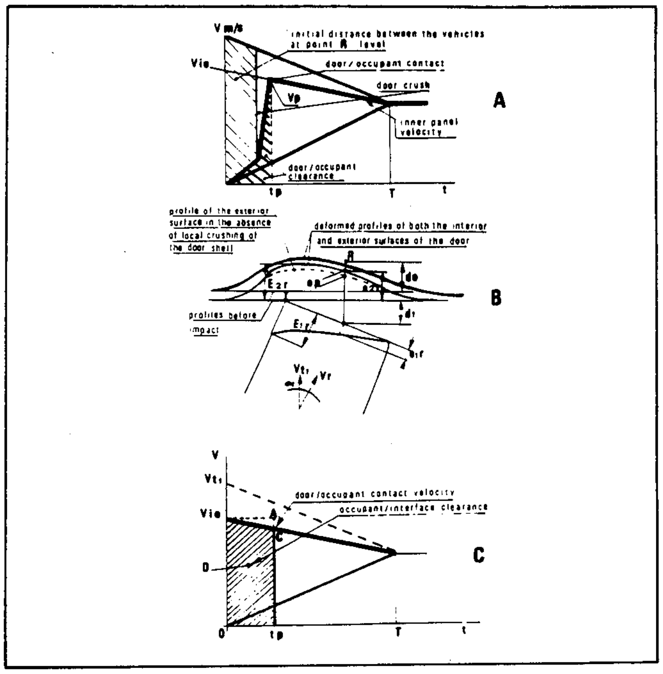
The interface velocity Vi is such that:

$$\frac{V1 - Vi}{Vi - V2} = \frac{e1d\cos\alpha}{e2d}$$

At the initial instant, this velocity is Vio:

$$Vio = e2d \frac{VT1}{e1d\cos\alpha + e2d}$$

In general, the vehicles are not in contact at occupant level at the onset of collision (staggered oblique impacts). The interface between the two cars



Figures A - B - C: A - Diagram of wall impact velocity
B - Notations
C - Initial distance (d) between interface and occupant, and wall/occupant contact velocity

is therefore not defined. So as not to differentiate the calculations according to the accident conditions we introduced a "virtual" interface between the two vehicles. The initial velocity of this virtual interface is an extrapolation of that of the real-world interface.

At the onset of impact this interface is in a clearly defined position in relation with the impacted vehicle. It depends on the rigidities present.

This initial position eio is the following:

$$eio = d1 \frac{e2d}{eldcos\alpha + e2d}$$

where d1 is the initial distance between the vehicles at occupant level (figure B).

Between the onset of impact and the occupant/wall impact (figure C), the real or virtual interface will have covered a distance D equal to the sum of:
eio: initial position of the virtual interface
ep: crushing of the door
do: initial occupant/wall distance

$$D = eio + ep + do$$

The instant tp of the occupant/wall contact is defined by:

$$\int_{o}^{tp} Vidt = D$$

With all the calculations made, it is the result of the following quadratic equation:

$$\frac{\Delta V2 - Vio}{2T} tp^2 + Viotp - D = 0$$

The velocity of the "occupant/wall" impact Vp is given by:

$$Vp1 = \frac{\Delta V2 - Vio}{T} tp + Vio$$

which must be increased to take account of the elastic restitution of the wall:

$$Vp = Vp1 (1 + R)$$

Feasibility and Validation of the Method

The coefficients used are based on the average values measured in experimental collisions:

- The static crush values E1r, E2r, e1r and e2r are increased by 20% to allow for elastic restitution.
- Wall velocity Vp1 must be increased at the end of the calculation by a factor R to take account of elastic restitution. The elastic restitution coefficients R are the averages obtained during experimental collisions. They are 5% at the thorax and 25% at the pelvis respectively.
- This method was applied to 40 cars tested in experimental impacts operating in the same conditions as real-world accidents.

The correlations obtained between occupant/wall impact velocities calculated by this method and the ΔV of dummies measured by the integration of the acceleration curves at thorax and pelvis are 0.78 for the thorax and 0.80 for the pelvis (figures D and E).

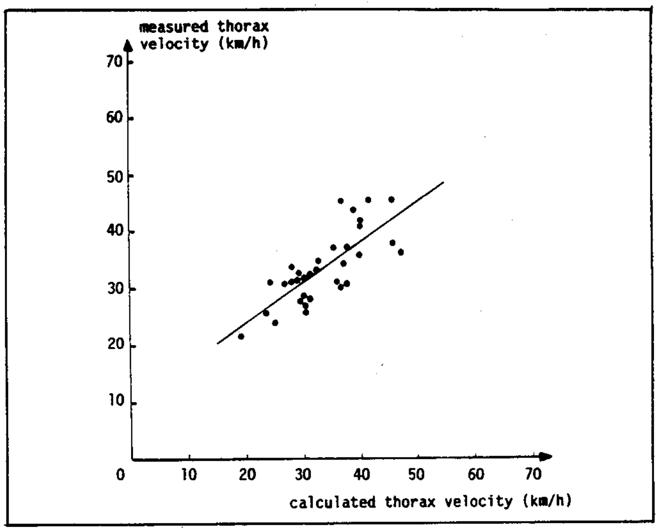


Figure D. Correspondences between calculated values and measured values for the dummy (thorax level) in 33 crash tests (result unknown for 7 cases)

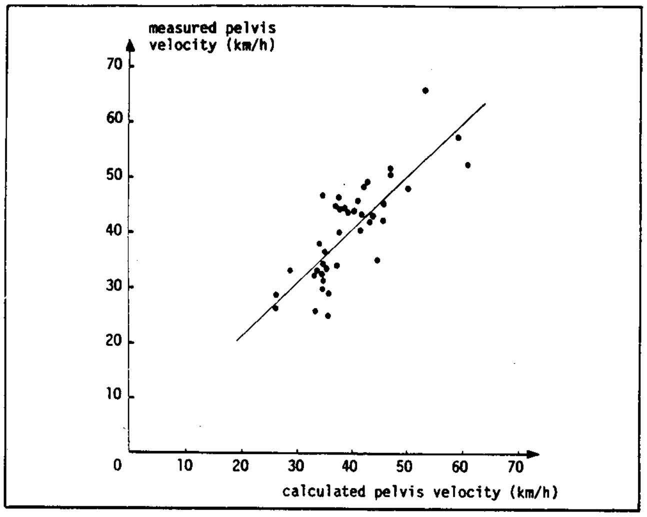


Figure E. Correspondences between calculated values and measured values for the dummy (pelvis level) in 40 crash tests

Analytical Simulation of the Effects of Structural Parameters on Occupant Responses in Side Impacts of Passenger Cars

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Abstract

Prior side impact modelling studies concentrated on characterizing the structural and occupant responses in perpendicular (90 degrees) and oblique (60 degrees) impacts where the striking vehicle's bumper engaged the door of the struck vehicle. In this model, the two colliding vehicles were modelled with three masses (representing the striking vehicle, and the struck vehicle's door and passenger compartment) and the struck car occupant was modelled with four masses (representing the pelvis, rib, spine and head body segments) from which the lateral motions of the vehicle and the occupant were obtained. While this model worked satisfactorily, only a limited number of variables could be parametrically assessed to determine their influence on occupant responses. It became apparent that a number of refinements were necessary to thoroughly investigate the effects of structural improvements and other injury mitigation concepts on occupant thoracic responses.

A new model was, therefore, developed with fifteen masses and thirty-one connecting energy absorbing elements which more closely simulated the interaction of the side door and adjoining structure of passenger cars and the interaction of the door with the occupant in side impacts. The most important additions were: (a) inclusion of rate sensitive factors in the energy absorbing elements, (b) inclusion of inner and outer door masses to account for the penetration distance between the exterior and interior door surfaces, (c) inclusion of upper and lower load paths for the door and the A/B pillars, (d) inclusion of door inner core energy absorbing elements to account for the internal stiffness and the stiffness of the interior door surface, and (e) inclusion of coupling stiffness between the pelvis and spine body segments which was assumed to have a negligible effect on occupant responses in the earlier model.

Simulation studies were performed with the new model to compare the responses obtained in side impact tests of various models of production cars conducted by the National Highway Traffic Safety Administration (NHTSA). Specifically, a VW-Rabbit, Nissan-Sentra, Honda-Civic, Chevrolet-Celebrity and Chevrolet-Spectrum were simulated using the new model. Additionally, side impact tests of Ford-LTD's conducted by Motor Vehicle Manufacturers Association (MVMA) were also simulated.

With the aid of this model, various vehicle characteristics and their effects on the thoracic response of an unrestrained occupant were investigated. These included: striking vehicle stiffness and its geometry, door internal structural characteristics of struck vehicles and internal geometry, stiffness characteristics of A/B pillars, and occupant's lateral seating positions. From the above parametric changes, the interactions of the occupant and the door structure were evaluated in an attempt to explain the occupant response differences seen in production car tests.

Introduction

Side impact collisions are the second leading cause of fatalities and injuries to motor vehicle occupants. In 1985, approximately 7,500 passenger car occupants were killed and about 25,000 were seriously injured in side impact accidents. An analysis of the available accident data files indicate that approximately 40 percent of the serious to fatal injuries in side impacts occur due to contacts of the occupants' "hard" thorax to the side interior components such as door panels, arm rests, and other interior hardware. The NHTSA has conducted a considerable amount of research on side impact protection during the last few years. It has developed various injury mitigation concepts in passenger vehicle design which would maximize occupant protection at a reasonable cost. Side impact modelling efforts have become an integral part of such side impact research. Under this effort, parametric effects of various structural modifications and other design changes on occupant responses were investigated systematically. This paper describes such an investigation utilizing an improved side impact model suitable for these studies.

The basis of our side impact modelling effort originated from the work presented in Ref(1). Although a simple representation was used to model the occupant, the concept of utilizing crash test data to characterize the deforming vehicle structures, and a single mass model of the occupant proved a first step in modelling real car collisions. Follow-on efforts improved on this model by including the other biomechanical elements such as the rib, spine and the head.

In our previous studies(2,3), a simple one-dimensional lumped parameter non-linear spring-mass

(1) Numbers in parenthesis indicate references at the end of the paper.

model was used to describe the occupant response due to the lateral interaction between a striking vehicle and a struck stationary vehicle and its normally seated occupant during side impacts. Three major parts were modelled, the impactor, the struck car and the occupant. In our previous model, three masses represented the impactor, the struck vehicle door, and the struck vehicle compartment, respectively. Two interconnecting non-linear springs represented the interaction between the striking vehicle's structure and the struck vehicle's compartment with the door. The structural characteristics (force versus deflection properties) were developed from the acceleration time histories obtained from dynamic crash tests. A Side Impact driver Dummy (SID), was modelled with four masses (rib, pelvis, spine and head), and four non-linear interconnecting springs with a damper in the thorax area. The rib and pelvis-to-door contacts were modelled with two non-linear springs. Parametric studies were carried out investigating the influence of striking vehicle stiffness, struck vehicle side stiffness, impact angle, impact velocity and door padding thickness and strength on driver responses. Sensitivity analysis provided acceleration response trends and probability of thoracic injury in collisions between a mobile deformable barrier (MDB) and a two-door VW-Rabbit at the MDB lateral striking velocities of 26, 30 and 35 mph. The parametric studies on these collisions concluded that the simulated GTR-door padding material (a General Tire and Rubber Company pad used in car crash tests)(11) was very effective in reducing the occupant thoracic acceleration levels. The analysis also showed that stiffening the side structure of the VW-Rabbit showed some benefit in reducing the peak thoracic accelerations, but stiffness alone could not achieve the large reductions in occupant accelerations offered by door padding alone. Finally, it was shown that reducing the stiffness of the MDB did not appreciably affect the acceleration levels in the thoracic area.

Numerous crash tests(4) have been performed by NHTSA on a variety of production vehicles in attempts to characterize their safety performance in 30/15 mph, 90-degree side impacts. These tests indicated the potential for a wide spectrum of thoracic injury levels. Most importantly, two small vehicles of approximately the same size and curb weight, a MY 82 Nissan-Sentra, and a MY 82 Honda-Civic showed under repeated side impact testing, notable differences in their potential for thoracic injuries. The Honda-Civic vehicle showed consistently lower driver acceleration levels. Review of crash test data and subsequent inspection of these vehicles' dynamically crushed doors was revealing. Contrasting features were observed in the thoracic acceleration patterns. The Nissan-Sentra rib responses were sharp and narrow whereas the Honda-Civic rib responses were low in magnitude and wider, thus, suggesting differences in the energy absorbing characteristics of the two doors. A comparison of rib velocity responses from these tests also supported this finding. From examination of the crushed Nissan-Sentra collapsed side structure, the evidence strongly indicated that the space between the exterior and interior door panels was consumed (both door panels were fully crushed together), thus causing the occupant in the crash to come in contact with a pancaked "hard" interior door surface which adversely affected the thoracic acceleration response levels. (A hard contact was evidenced since the struck occupant did not leave any indications of an impression on the crushed interior door surface.) The space between the exterior and interior surface appeared to be preserved in the Honda-Civic vehicle and, since an obvious contour of the occupants body contact was evidenced on the interior door surface, this suggested that the occupant contacted a compliant "cushioned" door surface during the impact. It was concluded from these data therefore, that a door cross-section which remains relatively undisturbed (stable) during the initial time period of occupant contact and acts as an energy dissipation element obviously provided improved occupant protection. Besides this, crash tests involving MDB profile changes(5) and component testing where a rigid thorax was propelled into an uncrushed door(6), showed further evidence that a door whose inner-to-outer panel thickness remains basically undisturbed in the early phase of a crash reduced the response levels in the thorax area. Based on these observations, it was desired to examine the effect of the interaction of the door structure with pillars on struck occupant responses. Since this could not be accomplished with our previous model because the door and surrounding structure were idealized as one mass, a new model was developed which included outer and inner masses for the door at the pelvis and thorax locations, and partitioned masses for A- and B-pillars to capture the dynamics of the door and the pillars. Studies were thus undertaken with the new model to evaluate the effectiveness of potential countermeasures in reducing thoracic injury to occupants, and to verify the observed differences noted in various crash tests.

This paper presents comparisons of simulation and test results of a number of side impacts. SID acceleration trends and a sensitivity data from parametric studies with the new model are discussed in detail. The unrestrained occupant modelled in this study was a 50th percentile male. For sensitivity studies, the occupant was seated four inches from the struck door. The new model is one-dimensional and considers the lateral interaction between a moving impactor and the left door of a stationary vehicle. The test configuration investigated the structural responses of a stationary target vehicle that was struck by a crabbed MDB simulating a two car collision involving vehicles moving perpendicular to each other at the striking vehicle velocity of 30 mph and struck vehicle velocity of 15 mph, and the occupant responses using the SID. The influence of the following parameters on driver responses are discussed.

- impact velocity
- door padding material
- struck vehicle side structure
- struck vehicle side structure plus door padding material
- door interior surface-to-pillar stiffness
- door inner core stiffness
- door interior panel stiffness
- door thickness (door gap)
- MDB stiffness
- MDB weight

These parametric variations were evaluated on three different vehicle designs: A VW-Rabbit, a Nissan-Sentra and a Honda-Civic in 90-degree collisions at the 30/15 mph impact. The struck vehicle design parameters were extensively varied for the Nissan-Sentra collision.

The struck occupant responses include the rib, pelvis and spine peak G-levels. Some safety performance comparisons using the NHTSA Thoracic Trauma Index (TTI)(7) were also made. TTI(0 age) is used for this purpose. TTI(0 age) is defined as the average thoracic acceleration unadjusted for age which is computed from the peak G-levels of the rib and lower spine responses. As an additional consideration, the relationships between the thorax peak G-levels, the crush between the rib and spine body segments and the peak value of the instantaneous product of the relative velocity and crush between the rib and spine body segments known as the viscous criteria (V*C)(8) are discussed for the simulated GTR-door padding material of varying thicknesses.

It is emphasized that the results presented in this study are for specific vehicle designs only, and should not be interpreted as being representative of all vehicle designs. In addition, the sensitivities resulting from the limited number of parametric changes serve here as indication of trends in the pelvic and thoracic response levels. Further efforts must be pursued to determine the influence of various parametric changes and their interactions in minimizing the potential for injury to occupants in all vehicle designs.

The Crash Model and Its Characteristics

As shown in Figures 1 and 1A (pictorial and top views of the model), the struck vehicle is characterized by 11-masses and 22 non-linear energy absorbing elements and the MDB is characterized by one mass and three non-linear energy absorbing elements including the MDB-to-sill element. Also see Figure 2 and Table 1. This model includes greater details of the characteristics of components of the struck vehicle which are encountered by the MDB on impact. The passenger compartment was modelled as a single mass (M10) to which the pillar sections and floor sill are connected. The door was subdivided into four interacting masses (M2, M3, M4, M5) to simulate the crush behavior between the exterior and interior door surfaces as well as the lower and upper segments at the driver pelvis and thorax contact locations. The A- and B-pillars were subdivided into lower (M6, M8) and upper masses (M7, M9) with interconnecting non-linear energy absorbing elements. A number of energy absorbing elements were also arranged among the different parts of the door structure and interfaced with the upper and lower pillar sections. This arrangement permitted the impact loads to be trans-

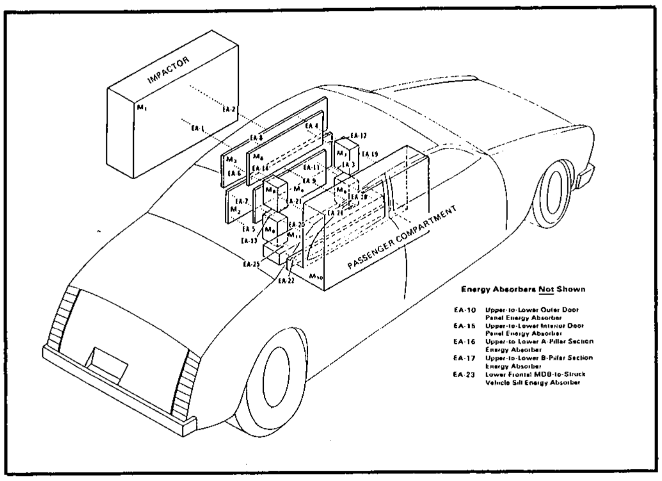


Figure 1. Lumped spring/mass model for struck vehicle

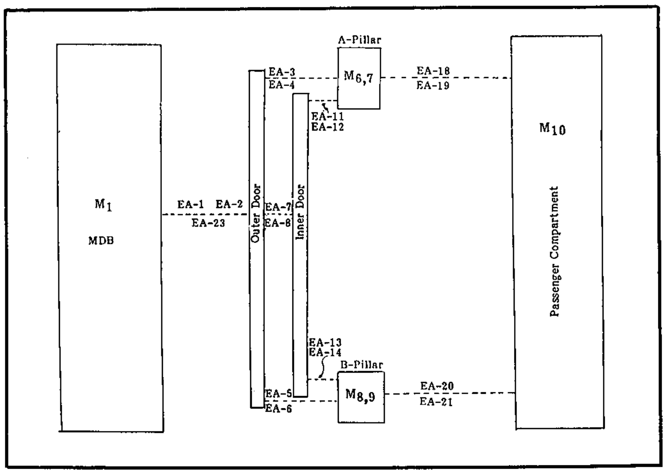


Figure 1A. Top view of the model

ferred to the compartment mass via two critical paths: (a) from the exterior door structure to the pillars and to the compartment, (b) from the combined exterior-interior door structure to the pillars and to the compartment. An energy absorbing element was also positioned between the lower inner door panel mass (M4) and sill mass (M11) to account for doors which are clipped to the floor sill. The striking vehicle was modelled by a single mass (M1) and two non-linear energy absorbing elements (EA-1, -2) to provide for the impact response of its upper and lower front-end structure for studies on the effects of MDB-stiffness and profile changes on occupant responses. The model also featured two types of engagements; the first where the MDB contacted the door of the struck vehicle and during this process rode over the floor sill, and the second where the MDB's bumper initially contacted the floor sill of the struck vehicle. Structural energy dissipation was accounted for by including the crush rate effects through the non-linear energy absorbing spring elements. The driver was modelled according to the model used in our previous studies(2,3), Figure 2.

The model masses were chosen to relate as closely as possible to the physical components of the vehicles. The weights were obtained from weight tear down data and from the weight measurements from crash tests. The structural data for the vehicle energy absorbers were: (a) extracted non-linear force-deflection characteristics from crash test acceleration data (b) measured force-deflection characteristics from component testing, (c) approximated force-deflection characteristics, and (d) engineering judgement where no data were available. The non-linear force-deflection traces from the crash tests were rate corrected for dynamic effects, thus each energy absorber was characterized by a static force deflection trace and a rate at which the energy absorber was deforming. The force-deflection properties for the MDB's upper and lower energy absorbing elements (EA-1,-2) were derived from acceleration traces obtained from MDB-to-stationary barrier crash tests. The energy absorbing elements characterizing the struck vehicle's door exterior surface-to-pillar (EA-3,-4,-5,-6) and pillar-to-compartment (EA-18,-19,-20,-21) upper and lower load paths were based on load apportioned force-deflection properties. Sensitivity analyses of various levels of apportionment were used to verify the structural and occupant responses obtained and the selection of the final values for use in the model. Experimental accelerometer time histories of the inner surface of the door at the mid-location were used for this purpose. Since the experimentally derived MDB-to-inner door surface force-deflection trace contained the influence of the MDB, the structural characteristics of the MDB were subtracted and the remaining force-deflection trace was then load apportioned among the upper and lower exterior door-to-pillar energy absorbers. The upper and lower energy absorbers (EA-7,-8) which interface the exterior and interior door panels were composed of two non-linear energy absorbers connected in parallel. The first energy

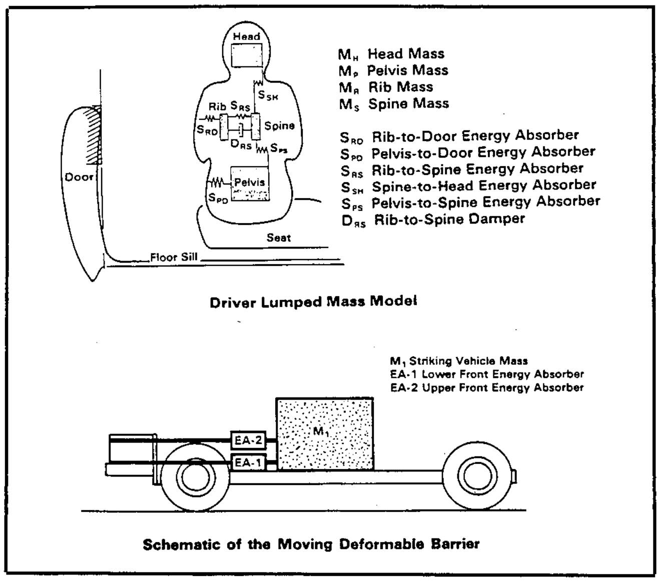


Figure 2. Lumped spring/mass model for MDB and driver in struck vehicle

Table 1. Model energy absorbers.

EA-1	MDB Lower Front
EA-2	MDB Upper Front
EA-3	Lower Outer Door Panel-to-Lower A-Pillar Section
EA-4	Upper Outer Door Panel-to-Upper A-Pillar Section
EA-5	Lower Outer Door Panel-to-Lower B-Pillar Section
EA-6	Upper Outer Door Panel-to-Upper B-Pillar Section
EA-7	Lower Outer Door Panel-to-Lower Inner Door Panel
EA-8	Upper Outer Door Panel-to-Upper Inner Door Panel
EA-9	Lower Outer Door Panel-to-Sill
EA-10	Upper-to-Lower Outer Door Panel
EA-11	Lower Inner Door Panel-to-Lower A-Pillar Section
EA-12	Upper Inner Door Panel-to-Upper A-Pillar Section
EA-13	Lower Inner Door Panel-to-Lower B-Pillar Section
EA-14	Upper Inner Door Panel-to-Upper B-Pillar Section
EA-15	Upper-to-Lower Interior Door Panel
EA-16	Upper-to-Lower A-Pillar Section
EA-17	Upper-to-Lower B-Pillar Section
EA-18	Lower A-Pillar Section-to-Passenger Compartment
EA-19	Upper A-Pillar Section-to-Passenger Compartment
EA-20	Lower B-Pillar Section-to-Passenger Compartment
EA-21	Upper B-Pillar Section-to-Passenger Compartment
EA-22	Sill-to-Passenger Compartment
EA-23	Lower Frontal MDB-to-Sill
EA-24	Lower A-Pillar-to-Sill
EA-25	Lower B-Pillar-to-Sill
S_{RD}	Rib-to-Door Energy Absorber
S_{PD}	Pelvis-to-Door Energy Absorber
S_{RS}	Rib-to-Spine Energy Absorber
S_{SH}	Spine-to-Head Energy Absorber
S_{PS}	Pelvis-to-Spine Energy Absorber
D_{RS}	Rib-to-Spine Damper

absorber is characterized as a non-linear spring which has no force buildup until the entire door gap is consumed and thereafter the force-deflection is modelled as a bottomed out spring (assumed as interior panel). Force-deflection traces from rigid thorax-to-unbacked door interior test(6) were assumed for the second energy absorber in the parallel branch (assumed as inner core). The energy absorber (EA-10) connecting the exterior upper and lower door panels was not considered for modelling, since this energy absorber supports practically little loads before yielding. On the other hand, the energy absorber (EA-15) connecting the interior door upper and lower panels usually supports higher loads before yielding, even though these panels are composed of open areas. For our purposes, this energy absorber's force-deflection trace was approximated from scaled rigid thorax unbacked door tests with the assumption that the shape of the energy absorbers force-deflection curve is similar to the shape of the rigid thorax unbacked door force-deflection curve. Force-deflection properties assumed for the door interior surface-to-pillar energy absorbers (EA-11,-12,-13,-14) were linear approximations so that stiffness values coud be assigned to simulate a multitude of door conditions: that is, where either the upper or lower interior door cavity, or both cavities, are preserved during the collision, or where either the upper or lower inner door section, or both sections, collapse with the A- and B-pillars, thus causing the door's exterior surface to come together with the interior door surface during collision. Force-deflection properties for the energy absorbers (EA-16,-17) which interface the upper and lower pillar sections were based on the force-deflection traces from rigid surrogate head-to-pillar tests(6). The characteristics described above constitute baseline characteristics for the model. These characteristics were further refined where the model responses were calibrated with test data.

Model Verification with Characterization Data

Mass and force-deflection characteristics were developed for six vehicle collisions in the 90-degree impacts:

- 1984 Chevrolet-Celebrity, 4 door
- 1985 Chevrolet-Spectrum, 2 door
- 1985 Ford-LTD, 4 door
- 1982 Honda-Civic, 4 door
- 1982 Nissan-Sentra, 2 door
- 1981 VW-Rabbit, 2 door

2 Special tests where dynamic impacts are performed on door interiors using a rigid surrogate thorax

Experimental response data from over twenty crash tests were used to verify the responses simulated by the new model for a variety of impact conditions and striking and struck vehicle characteristics. During this verification process, the model's energy absorbing elements were adjusted to match, as closely as possible, the occupant and vehicle responses from the crash test measurements. Response data from six crash tests were used to verify the VW-Rabbit structural characteristics as a function of impact speed, and to assess the sensitivity of the model to predict occupant seating position changes, the effect of door padding material thickness changes, and the effect of alterations to the striking vehicle front end characteristics on struck occupant responses. Ford-LTD model characteristics were based on five crash tests as a function of occupant seating position and struck vehicle side structure strength.

Figure 3 shows comparisons between the peak rib, pelvis and spine acceleration responses simulated by the model and measured from 90-degree crash tests for collisions involving the VW-Rabbit at three impact speeds. These results are based on 300Hz filtered traces. The experimental rib and spine accelerations are an average of two acceleration measurements in the upper thorax area. A reasonable agreement between model predictions and experimental results can be observed, although the predictions at the higher impact speeds show slightly lower G-levels for the rib and higher G-levels in the pelvic area than were measured. A sensitivity study showed this difference to be due to the relative phasing between pelvis and rib contacts with the door. Predicted and measured rib, spine, pelvis and head peak G-levels are also shown in Figure 4 for struck occupants in the Honda-Civic, Nissan-Sentra, Chevrolet-Spectrum and

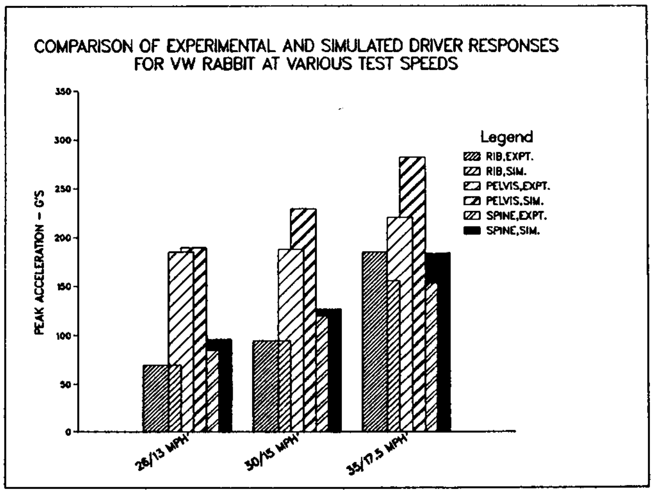


Figure 3. Comparison of simulated and experimental occupant peak G-levels for the VW-Rabbit vehicle

Chevrolet-Celebrity vehicles involved in 30/15 mph collisions. The measured Nissan-Sentra and Honda Civic peak G-levels here were from the first series of crash tests. Model predictions for the thorax and head appear to be quite reasonable. The predicted Nissan-Sentra pelvic peak G-level is also in good agreement with the test. The higher predicted Honda-Civic and Chevrolet-Spectrum occupant pelvic peak G-levels are due mainly to the higher stiffness assumed for the EA-7 energy absorber characterizing the door interior panel, which was the same as in the Nissan-Sentra and VW-Rabbit models. Certainly, a lower stiffness would allow for a softer pelvic contact, and thus result in lower G-levels in the pelvic area. Additional verification efforts were also carried out on the Ford-LTD, at the test conditions specified earlier. Similar agreement between model and experimental accelerations was achieved in this case also. For example, the average rib and spine G-levels from the two crash tests were 112.8G (rib), 113.1G (spine) for the passenger seated next to the door and 83.7G (rib), 82.3G (spine) for the passenger seated 5-inches from the door. The simulated values were 114.7G (rib), 123.0G (spine) and 85.1G (rib), 111.2G (spine) respectively, for the two seating positions.

In addition to the aforementioned simulations, the VW-Rabbit model was further exercised to simulate the dynamic interaction of two altered MDB's (a "low profile" front and a reduced crush strength front) crash tested at the 35/17.5 mph, 90-degree configuration in the NHTSA's "Side Impact Aggressiveness Attributes" program(5). Again, quite good agreement between model predictions and experimental results was obtained which verified the reliability of the model to simulate the effect of structural design changes on struck occupant peak G-levels. In the 'low profile' MDB test, the predicted thoracic peak G-levels were 73G for the rib and 92G for the spine compared to the measured peak G-levels of 90G for the rib and 80G for the spine, resulting in occupant TTI(0 age) values of 79.25G (calculated) and 77.5G (measured). The predicted rib and spine responses for the collision involving the reduced crush strength front MDB were 105G and 135G, compared to measured levels of 125G and 126G, respectively. Notice that the "low profile" MDB struck occupant thorax peak G-levels were much lower in value than the occupant's thorax peak G-levels observed in the standard MDB (see Figure 3). Moreover, compared to the baseline VW Rabbit, the "low profile" MDB indicated that the measured thorax peak G-level reductions were more than those obtained in the modified structure and padding case. Nevertheless, a review of the model responses indicated that the thoracic peak G-levels for the "low profile" MDB were lowered compared to the baseline because in the

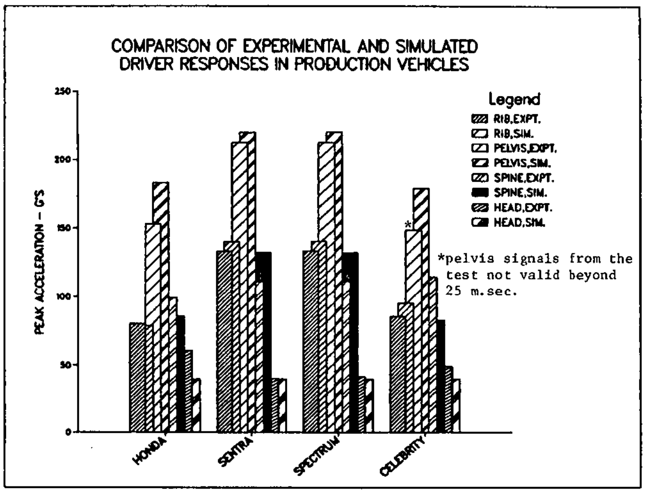


Figure 4. Comparison of simulated and experimental driver peak G-levels for four crash tested vehicles (30/15 mph, 90 deg. impact)

lower MDB profile simulation, thoracic contact was made with the door at a much later time than in the collision involving the standard MDB. Also the upper portion of the door was minimally crushed prior to the thorax contact when impacted by the "low profile" impactor.

In all these calibrations, the force deflection properties from crash tests were apportioned among the EA-3,-4,-5,-6 and EA-18,-19,-20,-21 parallel energy absorbers by the following force multiplication factors; 0.6 for the energy absorbers at the lower section of the struck vehicle and 0.4 for the energy absorbers at the upper section of the struck vehicle. In addition, the two energy absorbers at the B-pillar location were further force apportioned 75% lower in value than the two energy absorbers at the A-pillar location. Both cubic and linear force-deflection forms were used to model the bottomed out spring features observed in rigid thorax backed door force-deflection traces(6). These forms were then assigned for the bottomed out springs, characterizing the stiffnesses of the lower (a cubic form) and upper (a linear form) door interior surfaces, respectively. The following door interior surface-to-pillar stiffness values were found to be suitable in simulating the door responses of production vehicles. Stiffness values selected for the upper door interior surface-to-pillar energy absorbers were: 82 Klbs/in for the VW-Rabbit, Chevrolet-Celebrity and Nissan-Sentra, 8.2 Klbs/in for the Ford-LTD, 4.2 Klbs/in for the Chevrolet-Spectrum, and 0.82 Klbs/in for the Honda-Civic. Stiffness values selected for the lower door interior surface-to-pillar energy absorbers were: 82 Klbs/in for the VW-Rabbit, Chevrolet-Celebrity and Nissan-Sentra, 41.2 Klbs/in for the Ford-LTD and 8.2 Klbs/in for the Chevrolet-Spectrum and Honda-Civic. It is worthwhile to mention that these values were based on a heuristic approach utilizing the waveform properties of the crash test signatures together with simulations to arrive at suitable stiffnesses.

The structural weights for the VW-Rabbit vehicle, a two door model, were based on weight tear down data(9) and the total measured vehicle weight prior to testing. One-half of the vehicle's front side panel weight was evenly distributed between the upper and the lower sections of the A-pillar and one third of the combined weight of the vehicle's rear member and wheel housing was evenly distributed between the upper and lower B-pillar. The weight of the impactor was 3000 lbs. The VW-Rabbit tear down door and pillar weights were used for the other vehicles also since tear down data was not readily available on these vehicles or others of comparable size. Therefore, the mass selections for the simulations are considered appropriate for all the vehicles included in this investigation. A sensitivity study also showed that the pelvic accelerations were unaffected by door weight changes of up to 25% of the value estimates from the tear down data. Changes in spinal accelerations were also found to be minimal, on an average of less than 2.0G per doubling of door weight and rib accelerations changed anywhere between 10.0G to 20.0G per doubling of door weight. The passenger compartment weights were assumed equal to the measured vehicle weights in crash tests minus the door and pillar weights and the weight of the struck occupant. Table 2 summarizes the door and pillar weights used for the six vehicles, and Table 3 provides the compartment weights and the weights assumed for the occupant body segments.

Finally, the structural energy dissipation was based on the "power-like" expression in Ref(10) for structures subjected to steady and dynamic loads. In our model, the form F(dynamic) = F(static)*(1. + 0.203DV**0.2) was assumed where F is the force in kilo-pounds and DV is the energy absorber's deformation rate in inches/second. This resulted in a dynamic magnification (F dynamic/F static) of 1.7 at the 35 mph impact. This form was assumed to be adequate based on the sensitivity of this expression to simulate the occupant responses, and from independent least square fittings of multiple speed frontal and side crash data which suggested dynamic magnifications ranging from 1.25 to 2.1 at the 35 mph impact.

Parametric Studies

Based on the new model, parametric investigations were carried out for the side impact condition at 90-degrees and 30/15 mph with the MDB's bumper engaging the lower portion of the door. The occupant in the struck vehicle was a normally seated SID, four inches for the door interior surface. The two most commonly mentioned countermeasures, door padding thickness and struck vehicle side stiffness including their combined effect, were examined initially for crashes involving the Nissan-Sentra, Honda-Civic and VW-Rabbit vehicles. (Note that A- and B-pillar stiffnesses are referred to in this study as the struck vehicles side stiffness). Follow-on investigations included parametric characterizations with the Nissan-Sentra vehicle which dealt with the following side structural parameters, as potential countermeasures:

(1) door interior surface-to-pillar stiffness (EA-11,12,13,14)
(2) outer-to-inner door stiffness (EA-7,8)
(3) door thickness.

The following striking MDB parameters were included:

weight (M1), and stiffness (EA-1,2)

The striking MDB and struck vehicle stiffnesses were varied by multiplying the nominal loads in the energy absorber force-deflection diagrams by factors corresponding to the percentage change desired from the baseline design. Door padding force-deflection characteristics were based on a nonlinear expression developed from measured force-deflection characteristics of a General Tire and Rubber (GTR) pad used in the side impact car crash tests. All simulations were carried out with an integration time step of 0.0625 msec, which was determined to be the most suitable to use for such simulations.

Note that the sensitivity results which follow are not necessarily representative of the passenger car fleet because only input variables from the three vehicles were employed. Also, the interaction between various

Table 2. Weight data for door and pillars.

M_2	DOOR (OUTER LOWER SECTION)	15.3 Lbs
M_3	DOOR (OUTER UPPER SECTION)	6.34 Lbs
M_4	DOOR (INNER LOWER SECTION)	5.13 Lbs
M_5	DOOR (INNER UPPER SECTION)	8.08 Lbs
$M_{6,7}$	A-PILLAR (UPPER AND LOWER)	5.13 LBS
$M_{8,9}$	B-PILLAR (UPPER AND LOWER)	9.41 Lbs.

Table 3. Model weight data for passenger compartment and driver.

M_{10}		DRIVER	
VW-RABBIT	2280.0 lbs	M_{RIB}	20.0 lbs
NISSAN-SENTRA	2135.0 lbs	M_{SPINE}	32.0 lbs
HONDA-CIVIC	2283.0 lbs	M_{PELVIS}	79.0 lbs
CHEVROLET-SPECTRUM	2160.0 lbs	M_{HEAD}	10.0 lbs
CHEVROLET-CELEBRITY	3169.0 lbs		
FORD-LTD	3042.0 lbs		

parameters considered in this study were not taken into account in this analysis.

In general, for the range of variables considered, the parametric analyses showed that:

Driver (rib, pelvis and spine) peak G-levels decreased when:

(1) the thickness of the simulated GTR door padding material was increased,
(2) the side stiffness of the struck vehicle was increased,
(3) the interior door bottomed out stiffness was decreased, and
(4) door thickness was increased.

Driver (rib, pelvis and spine) peak G-levels increased when:

(1) the speed and weight of the striking MDB increased, and
(2) the frontal stiffness of the MDB was increased.

The accelerations of the rib and spine reach a minimum level for an optimum stiffness of the inner core energy absorber.

Door Padding Thickness and Struck Vehicle Side Stiffness Effects

Table 4 summarizes the occupant (rib, pelvis and spine) peak G-levels as a function of door padding thickness changes and struck vehicle side stiffness changes. These results show that the simulated GTR-door padding material (a pad of approximately constant stiffness properties) was a very effective energy dissipater in reducing the driver peak G-levels in the thoracic and pelvic areas for the 30/15 mph impact. Increased side structure stiffness provided some reductions in driver peak G-levels based on the vehicles analyzed, however, it was assumed that the same reductions could be achieved by the GTR-door padding alone.

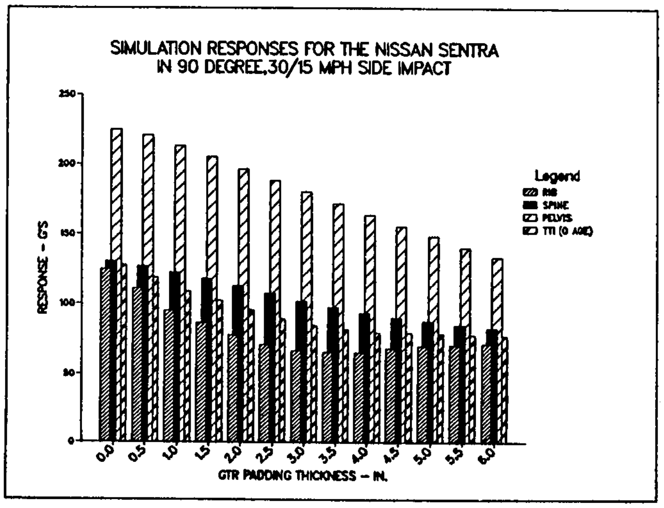


Figure 5. Driver thoracic responses for varying pad thicknesses

Door Padding Thickness Effects

Figure 5 illustrates, referring to the Nissan-Sentra collision as an example, the manner in which the struck driver peak G-levels varied for various thicknesses of the simulated GTR-padding material added to the door. Typically, the G-levels for the rib and spine were reduced effectively with about 3 inches of padding at the 30/15 mph impact. Further increase in the thickness of padding up to about 4 inches appeared to minimally reduce the rib peak G-levels from the levels already achieved with the 3 inch thick sample. Beyond the thickness of 4 inches, the rib responses indicated some change in the peak G-levels, where a slight increase was observed up to the 6 inch in thickness value. Pelvis and spine responses decreased linearly with padding thickness. Also, see Table 4 which indicates that the thoracic responses appear to be effectively reduced with a simulated GTR-padding thickness between 2 inches and 4 inches

Table 4. Door padding thickness and side stiffness (Peak G-levels).

		NISSAN-SENTRA			VW-RABBIT			HONDA-CIVIC		
		RIB	PELVIS	SPINE	RIB	PELVIS	SPINE	RIB	PELVIS	SPINE
BASELINE VEHICLE		124.6	225.0	130.6	119.7	231.3	122.4	102.3	186.3	78.6
GTR DOOR PADDING	2 inch	77.8	196.4	113.1	83.1	202.6	106.3	74.8	164.5	67.4
	4 inch	65.4	163.6	93.4	76.3	161.3	87.4	62.2	133.6	59.3
INCREASED SIDE STIFFNESS	1.2 Nom	128.2	213.0	129.2	90.4	218.8	118.6	97.9	171.3	75.7
	1.4 Nom	132.1	199.4	127.6	84.1	206.7	119.0	95.5	155.5	73.7
	2.0 Nom	96.3	145.7	105.8	66.9	174.6	96.8	101.0	102.6	68.5
INCREASED SIDE + STIFFNESS (1.2 Nom)	2 in GTR Pad	78.4	184.2	108.7	69.6	190.6	98.8	72.6	149.9	64.1
	4 in GTR Pad	60.8	153.6	90.2	68.7	183.0	95.8	58.3	122.8	56.1
INCREASED SIDE + STIFFNESS (1.4 Nom)	2 in GTR Pad	78.5	167.3	104.1	68.7	182.7	95.8	69.6	131.0	62.1
	4 in GTR Pad	61.7	140.2	86.4	74.9	141.3	75.4	55.1	107.2	54.8
INCREASED SIDE + STIFFNESS (2.0 Nom)	2 in GTR Pad	64.1	126.9	83.7	69.6	134.2	70.8	66.3	87.7	59.2
	4 in GTR Pad	59.4	104.6	76.5	69.7	102.2	66.6	50.6	75.1	52.6

for the other vehicles simulated in this study. It is evident that larger padding thicknesses are required to reduce the pelvic responses.

Most importantly, simulated GTR-padding performance evaluated on a basis of G-level reduction indicates sensitivity to vehicle design. As an example, the Nissan-Sentra driver's rib was reduced from 124.6G to 77.8G and 65.4 (46.8G and 59.2G reductions) and spine was reduced from 130.6G to 113.1G and 93.4G (17.5G and 37.2G reductions) with a 2 inch and a 4 inch GTR-paid, respectively. The VW-Rabbit driver's rib was reduced from 119.7G to 83.1G and 76.3G (36.6G and 43.4G reductions) and spine was reduced from 122.4G to 106.3G and 87.4G (16.1G and 35.0G reductions) with a 2 inch and a 4 inch GTR-pad, respectively, and the Honda-Civic driver's rib was reduced from 102.3G to 74.8G and 62.2G (27.5G and 40.0G reductions) and spine was reduced from 78.6G to 67.4G and 59.3G (11.2G and 19.3G reductions) with a 2 inch and a 4 inch pad, respectively. These reductions for the 2 inch simulated GTR-pad translate into an effectiveness of 37% for the Nissan-Sentra, 30% for the VW-Rabbit and 26% for the Honda-Civic for the rib and 13.3% for the Nissan-Sentra, 13% for the VW-Rabbit and 14.2% for the Honda-Civic for the spine. For the 4 inch simulated GTR-pad, the percentage rib reductions were respectively 47.5% for the Nissan-Sentra, 36% for the VW-Rabbit and 39% for the Honda-Civic, and the spine reductions were 28.4% for the Nissan-Sentra, 28.5% for the VW-Rabbit and 24.0% for the Honda-Civic. The simulated GTR-padding material effectiveness for the pelvis were similar to those observed for the spine.

Struck Vehicle Side Stiffness Effects

The struck driver peak G-levels (rib, pelvis and spine) are summarized in Table 4 for crashes involving the Nissan-Sentra, VW-Rabbit and Honda-Civic vehicles for side stiffness (pillar sections) increases of 20%, 40% and 100% greater than the nominal level. These results show basically that the benefits of added side stiffness, reduce the momentum transferred to the driver, thereby reducing the thoracic accelerations. This is clearly illustrated where doubling the stiffnesses for simulation purposes in the Nissan-Sentra and VW-Rabbit side structure compared to the baseline reduced the driver's thoracic peak G-levels, respectively, by 28.3G and 52.8G for the rib and 24.8G and 25.6G for the spine. Doubling the Honda-Civic's side stiffness indicated little-to-no advantage in reducing the thoracic peak G-levels. This indicates that the effect of the increase in side stiffness may have been overridden by other design characteristics which were responsible for the reduced thoracic G-levels which have already been achieved. It is observed in Table 4 that a two-fold increase in side stiffness projected much larger reductions in the pelvic area for all the three vehicles analyzed: 79.3G-level reduction for the Nissan-Sentra, 57.6G-level reduction for the VW-Rabbit and 83.7G-level reduction for the Honda-Civic.

Combined Door Padding and Struck Vehicle Side Stiffness Effects

The results in Table 4 indicate, for the Nissan-Sentra and VW-Rabbit vehicles with the same door padding thickness at the rib and pelvis locations, that stiffening the side structure by 20%, 40%, and 100% appears to provide little benefit in reducing the rib responses compared to the reductions achieved by door padding alone, since the simulated GTR-pad behaved as an effective dissipater of energy independent of the amount of increase in the stiffness as long as there was a mismatch between the padding and side structure stiffnesses, with the padding being less stiff than the structure. This is illustrated where a 4 inch thick door padding in the Nissan-Sentra and VW-Rabbit vehicles reduced the rib peak G-levels by 59.2G and 36.6G, respectively, whereas a twofold increase in side stiffness plus a 4 inch thick pad reduced the rib peak G-levels only by 65.2G and 50.1G, respectively. The results also indicated that the rib G-levels at the lower 1.2 nominal side stiffness can be reduced to levels comparable to those achieved by a smaller pad (2 inch thick pad) at the higher 2.0 nominal side stiffness by adding more padding. In spite of the fact that the spinal and pelvic responses are bio-mechanically coupled, and even though the spinal mass is heavier than the rib mass by a factor of 1.6, nearly similar peak G-level reduction trends were indicated for the spine and the rib due to the padding and stiffness changes. For the spine, a twofold increase in the Nissan-Sentra and VW-Rabbit side stiffness combined with a 4 inch thick simulated GTR-pad yielded reductions of 54.1G and 55.6G, respectively, whereas a 4 inch thick simulated GTR-pad alone yielded reductions of 37.2G and 35G, respectively, in these two vehicles. For the Honda-Civic vehicle, door padding and the combination of door padding and side stiffness indicated the same benefits in reducing the rib and spine peak G-levels (a 4 inch door pad alone reduced the rib and spine peak G-levels 40.1G and 19.3G, respectively, compared to the rib and spine peak G-level reductions of 51.6G and 21.0G, respectively, when the 4 inch thick simulated GTR-door pad was combined with the stiffer, 2.0 nominal side structure). Finally it is apparent that both door padding greater than 4 inches in thickness and doubling of the side stiffness was required to reduce the response levels of the heavier pelvic body segment below 100 G's in the VW-Rabbit and Nissan-Sentra.

Table 5. Door padding thickness and side stiffness (TTI-86).

		NISSAN-SENTRA	VW-RABBIT	HONDA-CIVIC
BASELINE VEHICLE		127.6	121.1	90.5
GTR DOOR PADDING	2 inch	95.5	94.7	71.1
	4 inch	79.4	81.9	60.8
INCREASED SIDE STIFFNESS	1.2 Nom	128.7	104.5	86.8
	1.4 Nom	129.9	101.6	84.6
	2.0 Nom	101.5	81.9	84.8
INCREASED SIDE STIFFNESS (1.2 Nom) +	2 in GTR Pad	93.6	84.2	68.4
	4 in GTR Pad	75.5	82.3	57.2
INCREASED SIDE STIFFNESS (1.4 Nom) +	2 in GTR Pad	91.3	82.3	65.9
	4 in GTR Pad	74.1	75.2	55.0
INCREASED SIDE STIFFNESS (2.0 Nom) +	2 in GTR Pad	73.9	70.2	62.8
	4 in GTR Pad	68.0	68.2	51.6

For measure of injury, Table 5 presents the TTI's (zero age) for the GTR-door padding thickness and side stiffness variations listed in Table 4.

Door and Door/Pillar Structural Effects

Table 6 summarizes the effect of door interior surface-to-pillar stiffness changes (Items B and C) and door interior panel and door inner core stiffness changes, (Items D and E) on occupant (rib, pelvis and spine) peak G-levels. These changes were evaluated on the Nissan-Sentra vehicle, for the 30/15 mph, 90-degree impact. It can be observed that the reductions predicted by the combined effect of increased door inner core stiffness, reduced door interior panel stiffness (EA-7,-8), and a reduced door interior surface-to-pillar stiffness (EA-11,-12,-13,-14) were the most significant characteristic in reducing the thoracic acceleration. Furthermore, the acceleration reductions due to these changes appeared to be of the same magnitude as was observed from the door padding thickness results discussed in the previous section. This is because these changes act to stabilize the door

Table 6. Nossan-Sentra door related stiffness variations (Peak G-levels).

ITEM				RIB	PELVIS	SPINE
A.	BASELINE			124.6	225.0	130.6
B.	DOOR INTERIOR PANEL-TO-PILLAR STIFFNESS (EA-11, -12, -13, -14)	0.2	Nom	135.0	225.5	132.2
		0.1	Nom	145.5	224.3	134.2
		0.01	Nom	155.7	221.7	136.9
C.	DOOR INTERIOR PANEL-TO-PILLAR STIFFNESS AT THORAX LOCATION, ONLY (EA-12, -14)	0.2	Nom	118.7	223.4	128.8
		0.1	Nom	120.0	228.8	129.3
		0.01	Nom	92.2	222.9	124.9
D.	DOOR INTERIOR PANEL STIFFNESS AT THORAX LOCATION, ONLY (EA-8)	1.0	(Base)	124.6	225.0	130.6
		0.5	Nom	114.5	225.0	128.7
		0.1	Nom	78.2	225.0	120.3
E.	DOOR INNER CORE STIFFNESS AT THORAX LOCATION, ONLY (EA-8)	0.5	Nom	172.4	220.6	141.4
		1.0	(Base)	124.6	225.0	130.6
		2.0	Nom	94.3	240.7	112.1
		3.0	Nom	150.2	242.8	132.7
F.	ITEM B (0.1 Nom) + DOOR INNER PANEL STIFFNESS (EA-7,-8; 0.5 Nom) + ITEM E	0.5	Nom	165.1	219.2	139.2
		1.0	Nom	143.9	217.7	133.6
		2.0	Nom	72.3	228.8	118.0
		3.0	Nom	115.6	231.3	112.1

interior panel with respect to the door outer panel such that its energy dissipative capability is always available at thorax contact.

Door Interior Surface-to-Pillar Stiffness Effects

The door interior surface-to-pillar energy absorbers (EA-11,12,13,14) are attachments which couple the motions of the door interior surface with the motions of the pillars, directly. Higher stiffness values for these energy absorbers in the model denote that both motions are strongly coupled, which may result in a situation where the door's exterior panel bottoms out with the door interior panel in the crash. When moderate stiffness is used, the opposite situation is encountered where the interior panel moves more-or-less in a manner where a gap is maintained between the exterior and interior panels when thorax contact is made with the door interior surface.

From Table 6, it can be seen that the pelvis, rib, and spine peak G-levels were not seriously affected by the door interior surface-to-pillar stiffness investigated here for the Nissan-Sentra vehicle. When all of the energy absorbers' (EA-11,-12,-13,-14) stiffnesses were lowered 100-fold (which in effect eliminates all of the above EA's), the rib responses increased by 25%, the spine responses increased by 5% and the pelvis responses decreased by 1%. This same trend was also evident when the two upper door interior surface-to-pillar stiffnesses were held fixed at their nominal values in the model and the two lower door interior surface-to-pillar stiffnesses were incrementally reduced to lower values. However, when just the upper door interior surface-to-pillar stiffnesses were lowered 100-fold, the rib responses decreased by 26%, the spine responses decreased by 4% and the pelvis responses decreased by 1%. It should be mentioned that a similar exercise on the VW-Rabbit showed the same trends and, most importantly, further verified the rather low sensitivity exhibited by the model when the door interior surface-to-pillar stiffnesses were varied, parametrically.

Door Interior Panel Stiffness Effects

It was of interest to observe the extent that the struck driver peak G-levels were affected by parametrically reducing the stiffnesses characterizing the bottomed out springs of the upper and lower EA-7,-8 energy absorbers (referred in the text as the door interior panel stiffnesses). The results are indicated in Table 6, Item D, where the nominal bottomed out spring stiffness at the thorax contact location was reduced by 50% and 90%, respectively, and Figure 6 where the rib, spine and pelvis peak G-level trends are presented in carpet plots for bottomed-out spring stiffnesses varied at both pelvis and thorax contact locations between 0.2 and 1.0 nominal stiffness. Basically all three body segment peak G-levels were reduced by lowering the door interior panel stiffness at both the pelvis and thorax contact locations. As expected, a decrease in the door interior panel stiffness at the thorax levels reduced the thoracic peak G-levels, and minimally affected the pelvic peak G-levels, whereas a decrease in the door interior panel stiffness at the pelvis contact level, while effectively reducing the pelvic peak G-levels, adversely affected the thoracic peak G-levels because of the diminished role of the pelvis body segment in reducing the momentum of the door before rib contact.

As illustrated in Table 6, the driver's rib seemed to be the most sensitive of the three body segments when this stiffness was reduced 90% from its nominal value. In general, lowering the door upper interior panel stiffnesses by 50% and 90% indicated rib peak G-levels reduced by 8% and 37%, respectively, and spine peak G-levels by 1.5% and 9%, respectively. However, the pelvic responses remained the same because the interior door panel characteristics were changed at the thoracic location only.

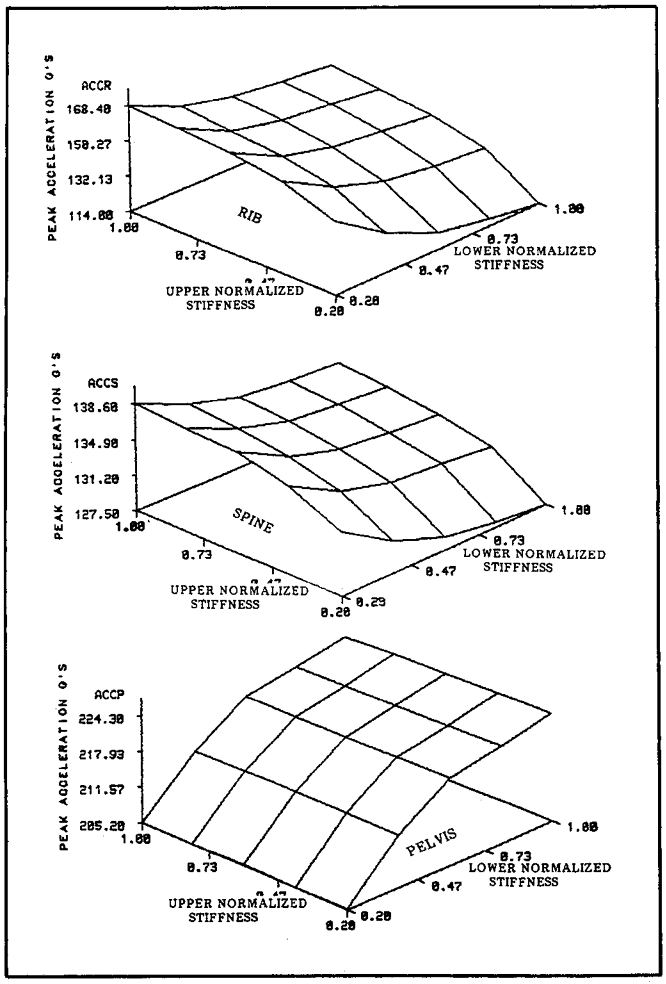


Figure 6. Effect of door interior panel stiffness

Door Inner Core Stiffness Effects

The door inner core non-linear spring represents, for our modelling purposes, the structural internal interface between the exterior and interior door panels and constitutes the second non-linear spring in the parallel spring arrangement characterizing the force-deflection properties of the EA-7,-8 energy absorbers. Item E in Table 6 illustrates the effectiveness of added stiffness between the exterior and interior door panels at the thorax contact location in reducing the acceleration responses in the thorax area, and Item F illustrates the effectiveness of this countermeasure in reducing the thoracic acceleration responses when it was considered together with door interior panel-to-pillar and door interior panel stiffness reductions. For example, a two-fold increase in the door inner core stiffness at the thorax contact location reduced the rib G-levels by 30.3G (24% reduction) and spine G-levels by 18.5G (14% reduction), thus resulting in an occupant TTI(0 age) reduction of 19%. On the other hand, a two-fold increase in the door core stiffness at the thorax contact location, in addition to a 90% reduction in the door interior panel-to-pillar stiffness and a 50% reduction in the door interior panel stiffness, resulted in rib and spine peak G-level reductions of 52.3G (42%) and 12.6G (10%), respectively, and an overall reduction of 25% for the occupant TTI(0 age).

Figure 7 shows the occupant responses for a combination of stiffness changes in the door interior panel to pillar stiffness, inner panel stiffness and door inner core stiffness (Item F, Table 6). These structural stiffness variations, excluding the variations previously discussed on door padding thickness changes, showed the greatest change in the thoracic peak G-levels. Most importantly, these variations bear out the significance of the design of the inner door structure in reducing the thoracic responses. When stiffness was added between the inner and outer upper surfaces where originally there was minimal stiffness between these surfaces, thoracic G-levels reduced to a minimum G-level, and as further stiffness was introduced between these two surfaces, the thoracic responses increased. The increase in the thoracic responses is to be expected, since as more stiffness was included between the inner and outer door sections, the driver responded more-or-less to a "hard" rather than a "soft" surface contact. The pelvic peak G-levels exhibited minimums also when the door internal stiffnesses at the upper and lower section took on the values in the range of 1.0 to 1.5 nominal. Note that the pelvic peak G-levels were influenced the least by the inner core stiffness changes.

The combined effect of door interior panel stiffness and door inner core stiffness changes on occupant responses were not parametrically examined in this

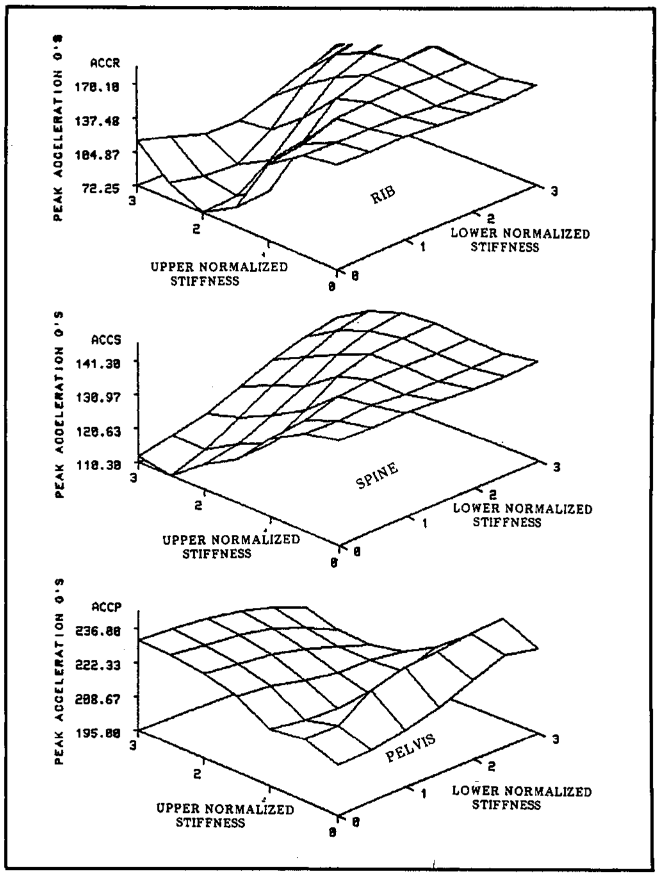


Figure 7. Effect of door inner core stiffness

study. The effect of such a design alteration can, however, be determined by mapping out parametrically the optimum thoracic peak G-levels as a function of the door inner core stiffness and door interior panel stiffness.

MDB Weight, Stiffness and Striking Velocity Effects

A least squares linear, global fit was performed on the parametrically derived driver response data which was obtained for the VW-Rabbit for various collision conditions of impact speed, the normalized MDB stiffness, and the MDB weight. The baseline conditions used were: impact speed = 30/15 mph, normalized MDB stiffness = 1.0 and MDB weight = 3.0 Klbs. The parametrically changed variables ranged in this analysis from: 30/15 mph to 45/22.5 mph for the impact speed, 0.2 to 1.6 for the normalized MDB stiffness, and 2.0 to 3.5 Klbs for the MDB weight. The sensitivity coefficients (C's: note change of peak G-levels per unit change of the parametric variable) resulting from this fit are shown in Table 7. In this example, special note should be made that these coefficients are valid only within a small region about the baseline. In addition, the values for some of these

Table 7. Least squares derived sensitivity coefficients for VW-Rabbit MDB, speed, stiffness and weight effects.

BODY SEGMENT	C_{SP}* (G/MPH)	G_{NS}* (G/10% INCREASE IN STIFFNESS	C_W* (G/KLB)
RIB	7.4	10.8	13.1
PELVIS	8.1	24.2	7.1
SPINE	2.9	6.5	6.4

*SP denotes impact speed, NS denotes normalized MDB stiffness and W denotes MDB weight.

coefficients, for instance those related to MDB-stiffness, are expected to differ from vehicle to vehicle.

Parameter Assessment on TTI

Figure 8 shows an example of the effectiveness of the door and door/pillar variables, discussed in the previous section, on the potential TTI reductions for the Nissan-Sentra vehicle in a 30/15 mph, 90-degree impact. It is important to note that the values assumed for the door and door/pillar property changes were estimates, since the purpose here was to illustrate a possible improvement which may be obtained through their combined influence. Also, because of their mutual interaction, the order in which these variables were changed in this example was completely arbitrary. This example shows that a 3% reduction in the TTI(0 age) was predicted when the two upper door interior panel-to-pillar energy absorbers' stiffnesses were reduced by 60%. The TTI(0 age) was further reduced 15.5% from the baseline value by increasing the stiffness of the door inner core energy absorber at the thorax contact location by 50%. This, combined with a 90% decrease in the upper door interior surface stiffness, commensurate with the previous change to allow for a softer thorax contact, resulted in lowering the TTI(0 age) further, to 27.5% of the baseline value. Finally, by widening the door by one-half inch, the TTI(0 age) was lowered to 85.2G, a 33% overall cumulative reduction from the baseline Nissan-Sentra design. The effect of the combined door property changes on the relative displacement of the outer and inner door panel is illustrated in Figure 9. It is observed that the door property changes combined with the interaction of the pillars retarded the door's cross-section from being crushed from the outside to inside and improved on the energy absorbing characteristics of the door at the time of thorax contact. In this figure, the outer door panel of the baseline vehicle accelerated more quickly towards the inner door panel than the modified vehicle initially because of the lower door inner core stiffness of the baseline vehicle. Once sufficient momentum was transferred to the interior panel, the entire door cross section moved inward as a rigid body. Most interestingly, less outer to inner door crush was produced in the modified design in the time period before the thorax engaged the intruding door. As expected, once thorax contact was made, the door thickness decreased. Note that a sharp increase in the relative displacement between the outer and inner panels after the thorax contact indicates a more energy absorbing door structure with a softer door interior panel stiffness.

Injury Assessment

Based on cadaver testing, NHTSA has proposed the TTI which is an acceleration based measure of the

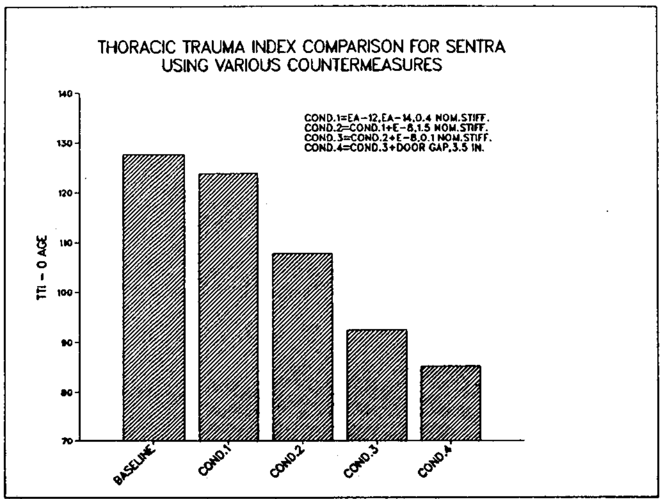


Figure 8. Thoracic Trauma Index for various countermeasures (Nissan-Sentra, 30/15 mph, 90 deg. impact)

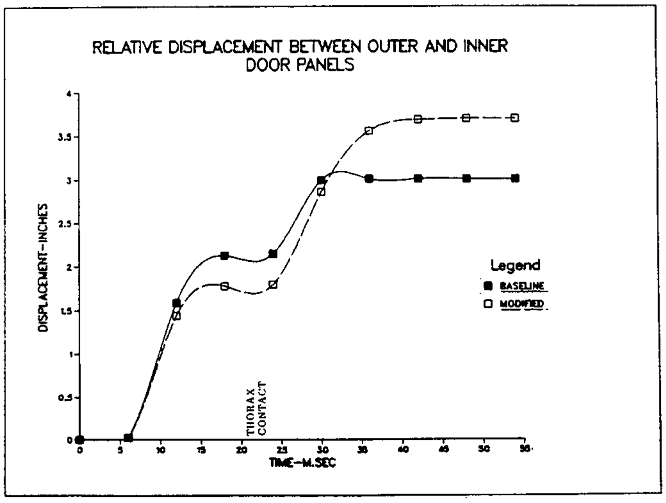


Figure 9. Relative displacement between outer and inner door panels as a function of time for the baseline and modified Nissan-Sentra vehicle

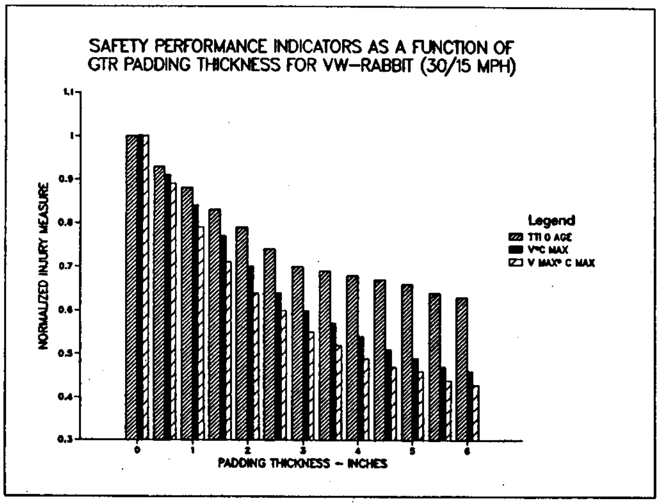
thoracic injury potential. This by far is the most elaborate research on injury criteria involving cadaver testing in side impacts. General Motors (GM) has proposed an alternate measure of thoracic injury known as the Viscous Injury Criteria (V*C)(8) which has its origin in frontal impact and is yet to be fully developed for side impacts. However, GM is in the process of extending its research for eventual application of V*C to side impacts. The peak value of V*C which is purported to be an indication of the thoracic injury potential, is the product of the instantaneous velocity of chest deformation and the chest deformation itself. A variation of the V*C is the use of the product of the maximum V and maximum C referred to as V(max)*C(max). Another criteria which has been proposed by researchers is the chest deflection which has been found to have serious limitations because in most of the full scale testing, the ribs reach their maximum mechanically allowed displacements.

In our analysis, the V*C and V(max)*C(max) have been obtained from the model and compared to the TTI in Figure 10, for an MDB to VW-Rabbit collision at 30/15 mph in which the VW-Rabbit was equipped with a GTR type pad. It should be recalled that the occupant from which these data were obtained is a simulation of the NHTSA SID.

The effect of increased pad thickness is expected to reduce the thoracic injury level as shown in Figure 10. However, the recent GM crash model results by Deng(12) show the effect of increasing pad thickness to increase the V*C while the corresponding TTI level decreased, as expected. Although this contrasting trend is not fully explained in Reference(12), it is suspected that it may be due to the characteristic features of the GM model attributed to the use of a solid unsegmented door, the structural properties of side structure, and the properties used to simulate the dummy. The V*C(max) and TTI(0 age) were also computed for various GTR-pad thicknesses at other impact velocities as well as other door stiffness characteristics. These included:

- a rigid door (a very large stiffness value was assumed for the EA-8 energy absorber).
- a rigid door plus GTR-pad placed between the driver and door while a four-inch distance was maintained between the driver and the unpadded door surface.

These variations also showed a similar trend between V*C(max.) and TTI as was indicated in Figure 10. Simulation with the VW-Rabbit, Nissan-Sentra and Honda-Civic vehicles on a variety of MDB stiffnesses, as well as profiles, further substantiated these findings.



* Normalized to the baseline response

Figure 10. Comparison of three safety performance measures as a function of door GTR-padding thickness

Summary and Conclusions

A new lumped-mass model was developed for the assessment of countermeasures in reducing thoracic injuries in side impacts. This model included, outer and inner masses for the door at the pelvis and thorax locations, and partitioned masses for A- and B-pillars to simulate the dynamics of the door and the pillars. It is ideally suited for in-depth investigations of door property changes, MDB and side structural design changes for occupant response trends and sensitivity studies, which could not be accomplished with the earlier lumped mass model(2). Baseline calibration studies focused on six vehicles of different curb weights and sizes. Experimental response data from over twenty crash tests were used to verify the responses simulated by the new model for a variety of impact conditions and striking and struck vehicle characteristics. Acceptable agreement between model predictions and experimental results were observed. The reliability of the model to simulate a multitude of structural design changes and their influence on occupant responses was also verified.

Parametric studies carried out on three small production vehicles, a VW-Rabbit, Nissan-Sentra and a Honda-Civic in 30/15mph, 90-degree impacts are reported here. In general, the unrestrained occupant in each of these vehicles displayed different sensitivities to the striking and struck vehicle changes parametrically varied in this study. In addition, the effectiveness of the countermeasures examined in this paper was dependent on vehicle make and the design characteristics of the door and the side structure of these vehicles. The most beneficial countermeasures in reducing the thorax peak G-levels were: (a) simulated GTR-door padding material, and (b) a stabilized door structure with adequate energy absorption which retains its original cross-sectional thickness at the thoracic contact location in the crash. It was also observed that the latter concept, depending on the vehicle and the severity of the crash, can provide the same reduction as was obtained with the simulated GTR-door padding material. Increase in side structure stiffness showed for two out of three vehicles in this study some benefits in reducing the thoracic responses, however, stiffness alone could not achieve the large acceleration reductions offered by the simulated GTR-door padding material or by utilizing a stable door structure which acts as an effective energy absorbing element. (Critical experiments in 1982(13) also reached similar conclusions where it was found that an effective energy absorbing door structure was most effective and practical in reducing occupant G's and that adding stiffness to the exterior structure was found to be a much less effective countermeasure.) Finally the effect of MDB weight changes on occupant acceleration responses was found to be small.

In the evaluation of three injury severity measures, TTI(0 age), V*C(max.) and V(max.)*C(max.) the model indicated that for the 50th percentile dummy, all the three injury measures showed similar trends for all the parameters varied in the model. It must be pointed out that in contrast to what was reported by Deng(12), the TTI and V*C(max.) showed the same trends for varying pad thicknesses. In the case of the TTI, the effectiveness of the padding becomes negligible above a thickness of four inches. However, the V*C(max.) as well as V(max)*C(max.) indicates that the padding will continue to be effective beyond a thickness of four inches. Based on the model calculations of the energy dissipated in the padding, it was observed that there was very little additional energy dissipation obtained by the simulated GTR padding material which had a thickness greater than 4 inches. Therefore, what is seen as an apparent reduction in injury potential in the V*C criteria may be an artifact of the definition of the criteria itself. Realistically, once the impact energy is dissipated by an optimum level of padding, any further addition of padding will not allow additional gain in effectiveness, as indicated by the TTI, in Figure 10. The effect of striking vehicle stiffness and profile changes simulated for the MDB also showed that their effects on the TTI and V*C(max.) were similar.

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Side Impact Simulation and Thoracic Injury Assessment

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Abstract

A car-to-car side impact model was developed using the Cal3D occupant simulation program and validated against test data for the vehicle and the occupant kinematics. Compared to the existing side impact vehicle/occupant models in the literature, this model has a more detailed vehicle side structure representation and a two-segment thorax featuring the rib cage and the spine. The former gives a more realistic impact response for the door and the latter provides the necessary kinematic information for evaluating both the acceleration-based thoracic injury criteria, such as Thoracic Trauma Index (TTI), and deformation-based criteria, such as Viscous Criterion (VC). The parameter variation study indicated that for the specific vehicle modeled, a stiffer side structure and more spacing between the occupant and the door would reduce injury severity indices. The use of padding material between the occupant and the door inner panel resulted in a reduction in the occupant acceleration response but, because of increased duration of occupant contact, an increase in the deformation response. These predictions were confirmed by a recent side impact test series conducted by the MVMA. The simulation results also indicated that the particular selection of the padding material is important in achieving the lowest possible occupant acceleration response.

Introduction

Car-to-car side impact is a frequent type of traffic accident and often leads to serious occupant injury[1]. Most restraint systems, like seat belts and airbags, are designed to avoid occupant contact with the vehicle interior by limiting his motion after impact. However, they are not effective means for side impact protection of the near side occupants, in which the injury usually resulted from occupant contact with the intruding side structure. There has been significant effort in the automotive industries as well as the government safety agencies in seeking other protective methods, for example, stiffer structure and padding on the door inner panel[2,3]. Federal Motor Vehicle Safety Standard 214, specifying the static strength requirement for side doors, is intended to maintain structural integrity when side impact occurs. It has been reported[4] that this requirement has effectively reduced the fatality risk and hospitalization in side impacts.

Recently, the National Highway Traffic Safety Administration (NHTSA) has been conducting an extensive research program on the improvement of side impact protection[5,6]. The results include: the development of a Side Impact Dummy (SID) and the moving side impact barrier, establishing the thoracic injury criterion (TTI), and the evaluation of the use of padding material on the door inner panel. Their test results and the TTI have led them to conclude that the padding of the door inner surface would improve occupant protection in side impact. However, there is a considerable dispute within the biomechanics community upon the biofidelity of the TTI criterion. Other criteria have been proposed for evaluating the thoracic injury, for example, the Viscous Criterion(VC)[7] developed by the Biomedical Science Department of the General Motors Research Laboratories. This criterion has been successfully utilized for assessing the thoracic injury due to frontal impact.

This study will examine the side impact mechanics with an occupant simulation program Cal3D[8] to gain more insight and understanding of the mechanism involved. For analyzing complex physical systems, such as those involved in side impact, mathematical models are effective tools for the following reasons: (1). they do not suffer repeatability problems as often encountered in test programs and (2). parameter studies can easily be conducted. Side impact simulation with Cal3D program has been conducted by Padgaonkar and Prasad[9], and Segal[10] on various aspects of the subject. This investigation adopts a different modeling approach than these previous studies and will consider a number of thoracic injury indices which have appeared in recent literature. The intention was to achieve more physical correlation in the model and to broaden the scope of injury assessment. Then a study was carried out to evaluate the effect of several design parameters on occupant response.

Model Description

This model is based on a test conducted at the GM Proving Ground, in which a moving NHTSA barrier was used to strike the passenger side of a stationary compact car at 42 km/h. A NHTSA SID dummy was located at the right front passenger position with shoulder and lap belts. As shown in Figure 1, the struck car and the barrier front are represented by a number of contact planes. The side structure of the struck vehicle is modeled by a series of segments connected with pin joints. The first segment of this linkage is connected to the car body and the mass center of the last segment is constrained to slide along a longitudinal line fixed on the car. This arrangement would allow the side structure to intrude into the passenger compartment as the impact occurs. Forces are transmitted along the joints of this linkage to the two end points to move the struck vehicle in the lateral direction.

The parameters in the model were first estimated from available structure component test data and then fine-tuned to match the vehicle and door kinematics. Utilizing the available modeling features of the Cal3D program, the joints of the side structure linkage have a rotational spring constant of 5 kN•m/deg with an energy dissipation factor 0.1, and a viscous coefficient of 50 N•m•s/deg. The contact between the barrier front and the side structure is characterized by a force-deflection factor of 5.55 kN/cm, an energy absorption factor 0.1, a permanent deformation factor 0.9, and a coefficient of friction 0.5.

The Side Impact Dummy (SID) is a modification of the Part 572 dummy for more human-like side impact response in the thoracic area[11]. It is modeled by a number of rigid segments representing the head, neck, chest, pelvis and the lower limbs. Since thoracic injury is the major consideration in side impact, the chest is modeled in a more detailed fashion with two segments representing the thoracic spine and the rib cage. The thoracic spine is connected to the neck and the lumbar spine with joints and the rib cage is coupled to the spine with eight spring and damper sets placed at some strategic locations as shown in Figure 2. This arrangement was designed to provide the appropriate compliance in translation and rotation between the two segments. The contact between the door panel and the chest is characterized by a force-deflection factor of 1.5 kN/cm, an energy absorption factor of 0.2, a permanent deflection factor of 0.8, and a friction coefficient of 0.5. These numbers, adjusted from the chest kinematics comparison, would produce a dynamic force-deflection curve similar to the SID chest pendulum test response as will be seen in the following section. The inertial properties of the vehicle/occupant model is summarized in Table 1.

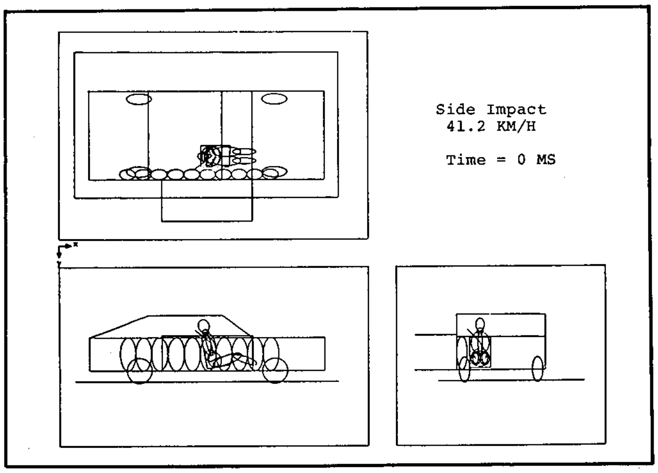

Figure 1. The side impact model

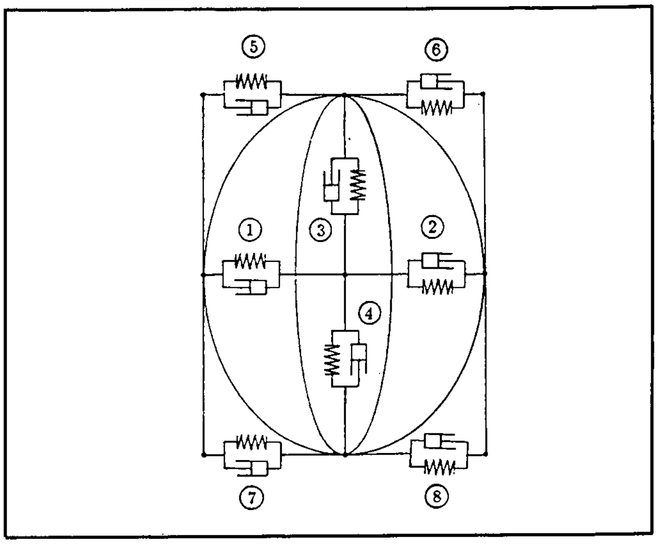

Figure 2. The dummy thorax model

Table 1. The inertial properties of the side impact model.

Segment	Mass kg	Moments of Inertia kg•m² X	Y	Z
Struck Car	1028.	2960.	742.	2804.
Door Segment	30.	.98	.98	.49
Barrier	1349.	2960.	742.	2804.
Head	4.39	.0248	.0289	.0185
Neck	0.82	.00133	.00133	.000569
Thoracic Spine	8.5	.118	.0883	.0785
Rib Cage	8.5	.118	.0883	.0785
Lumbar	1.36	.0024	.0024	.0098
Pelvis	13.17	.224	.156	.167
Upper Leg	9.52	.0873	.0873	.0131
Lower Leg	3.18	.0673	.0673	.00363

Model Validation

Figures 3-6 depict the vehicle/occupant configuration during the impact event at 20 ms intervals. It can be observed from the 80 ms configuration that the head of the occupant is striking the top of the impacting barrier. This would cause a late spike in the head acceleration time history. The kinematics comparison between the test data and the model is presented in Figures 7-8 for the struck vehicle and the door. The velocity response in the test is the result of integrating the acceleration measurement. The correlation between the two is considered to be satisfactory except that some oscillations were found in the vehicle acceleration test data. This measurement was taken from an accelerometer mounted on the driver side floorpan, which may contain some complex interaction among the various vehicle structure components. Nonetheless, the average acceleration of this oscillatory response seems to be in reasonable agreement with the simulation result. The door acceleration measurement shown in Figure 8 was obtained from an accelerometer mounted on the door inner panel at a point near the area that struck the occupant chest. The counterpart in the model is the response of the 5th segment from the front of the side structure linkage. In addition to the door acceleration and velocity correspondence between the model and the test, the amount of door intrusion in the model is 31 cm which also closely reflects the measured door crush near the occupant position after impact.

The occupant response correlation between the test and the simulation is illustrated in Figures 9 and 10 for a number of dummy segment accelerations: the pelvis, the lower and the upper spine, the head and the right upper rib. The comparison is considered to be acceptable. It should be pointed out that there is a slight difference in the response timing between the test and the model prediction. This difference may result from approximating the contact surfaces in the physical system with ellipsoids in the model. For the sake of comparing the response characteristics, the simulation response curves have been shifted as indicated on these figures. As noted earlier, the head experienced an acceleration spike at about 70 ms due to its contact with the top of the incoming barrier.

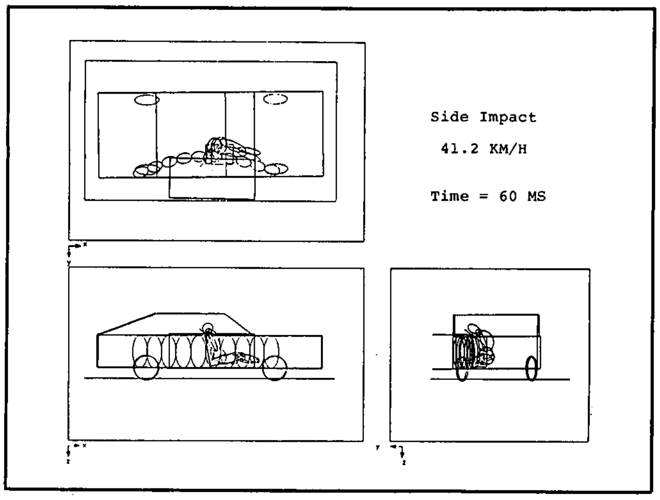

Figure 4. The vehicle/occupant configuration at 40 MS

Figure 11 depicts the velocity profiles of the striking and the struck vehicles, as well as the door area near the occupant chest area. After impact, the door quickly picks up the striking vehicle velocity with an overshoot. This overshoot may be due to the fact that the mass of the door is much smaller than that of the striking vehicle. During the contact phase, the door is following the striking vehicle with some fluctuations. The peak door velocity occurs at about 18 ms, which is about the time the door impacts the occupant chest.

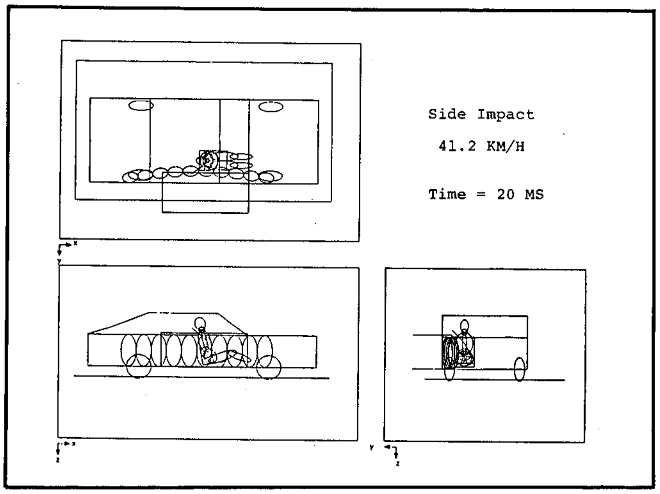

Figure 3. The vehicle/occupant configuration at 20 MS

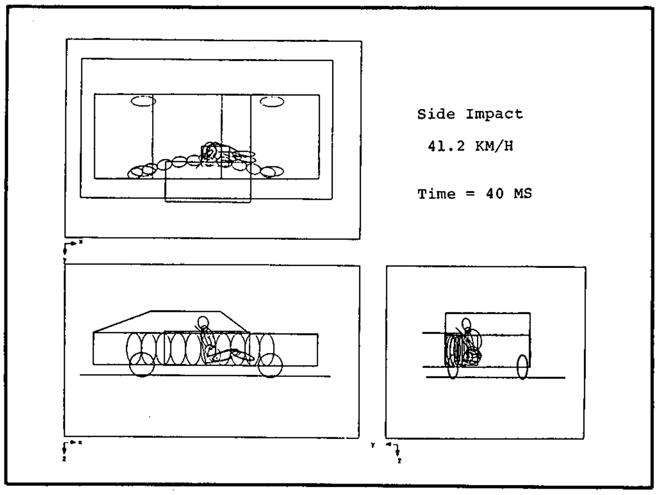

Figure 5. The vehicle/occupant configuration at 60 MS

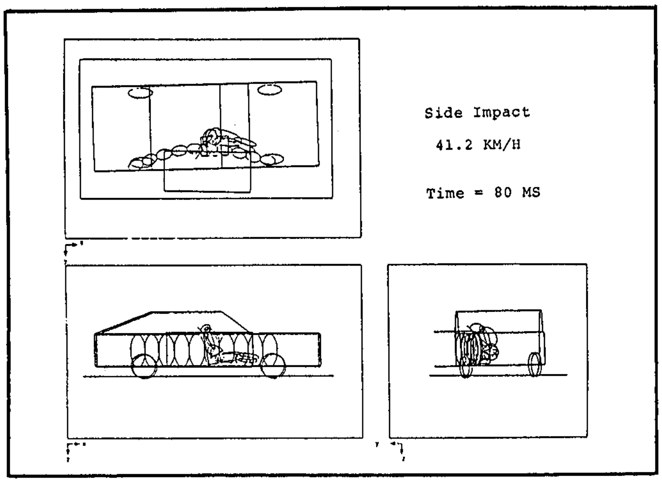

Figure 6. The vehicle/occupant configuration at 80 MS

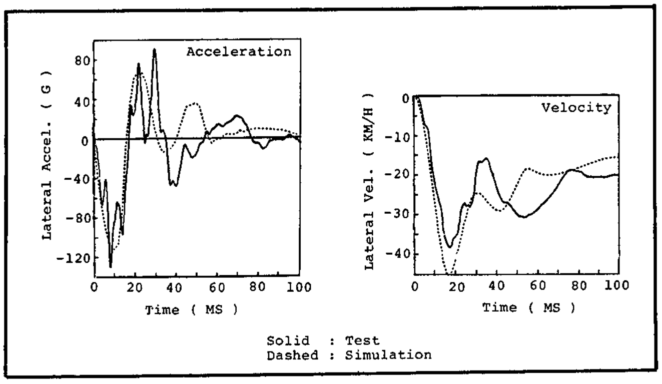

Figure 8. The acceleration and the velocity of the door

The fact that there is a substantial reduction in door velocity immediately after the peak value point suggests that a delay of door/occupant contact would be beneficial. The common velocity between the two vehicles was reached at about 58 ms. After this point, the two vehicles start to separate.

Another observation can be made regarding the side impact response of the Side Impact Dummy chest. Figure 12 shows the dynamic force-deflection characteristics of the chest contact from the simulation results. It can be seen that the simulation curve has a similar response pattern as that of a lower speed thorax pendulum test except the magnitude difference. This agreement provides further support for relating the thorax model in this simulation to the real physical structure.

Injury Criteria

For injury assessment purposes, it is necessary to have some criteria to quantify the injury severity. Several mechanisms have been proposed in the literature for characterizing the thoracic side impact trauma. In general, they can be divided into acceleration-based criteria and deformation-based criteria. This investigation will consider two from each category in the parameter variation study. The first index is the peak acceleration at the mass center of the rib cage segment. This index is related to the average rib acceleration and can be viewed as some measure of chest acceleration. The second acceleration-based criterion is the Thoracic Trauma Index (TTI) proposed by the NHTSA[5] based on statistically matching cadaver test data. The TTI takes the following expression:

$$\text{TTI} = 1.4 \text{ Age} + 0.5 \text{ (T12Y)} + \text{LURY)} * \frac{\text{Mass}}{165}$$

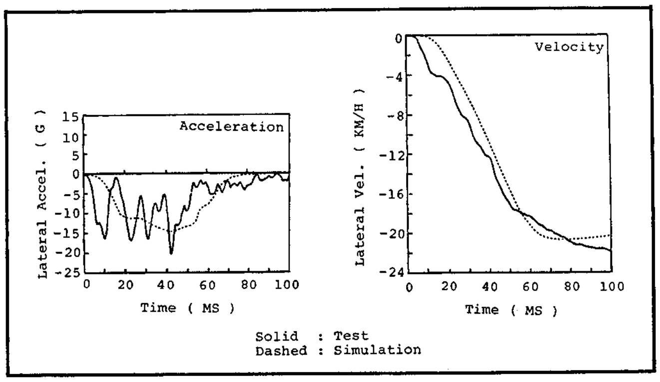

Figure 7. The acceleration and the velocity of the struck vehicle

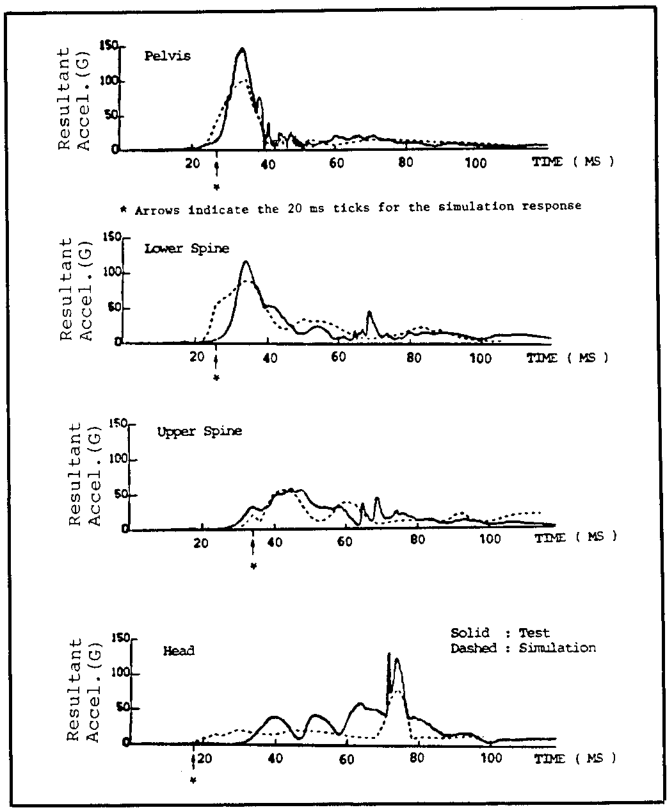

Figure 9. The occupant head and torso acceleration

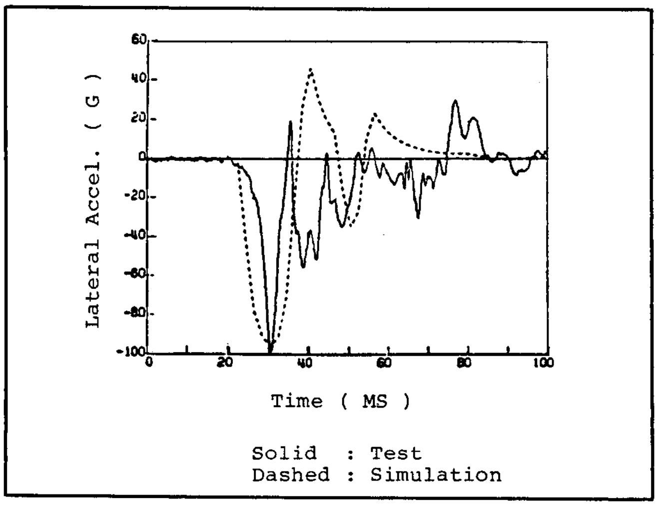

Figure 10. The occupant right upper rib acceleration

where, T12Y is the peak Y (lateral) acceleration at T12 vertebra, and LURY is the Left Upper Rib Y acceleration. In this study, the age parameter will assume the value of 41, which is the median age of the occupant with AIS 3 or greater thorax injury in a side impact in the National Crash Severity Study (NCSS) file. Using other age parameters would only change the TTI by a constant term and have no effect on the observations of the parameter study. The total mass of the dummy segment in this model is 136.7 lbs. The acceleration terms were readily available from the simulation except that the Right Upper Rib Y(RURY) acceleration is used in this study since this investigation is concerned with the passenger side impact as oppose to the driver side impact in the development of this index.

The third index is the rib deformation. In this model, it is represented by the amount of compression in the spring-damper set number one as shown in Figure 2. This quantity represents the relative displacement between the rib cage segment and the

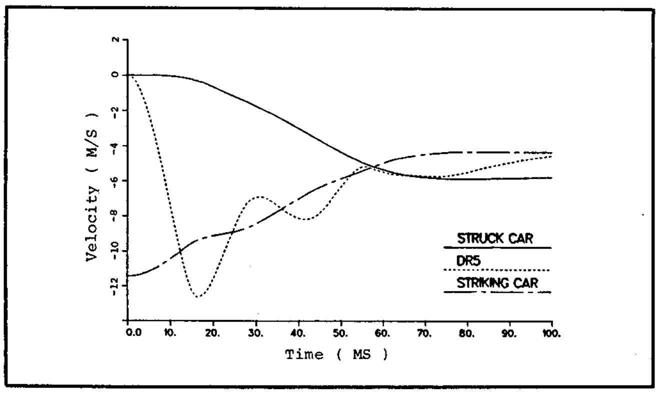

Figure 11. The lateral velocity of the striking car, the struck car and the door

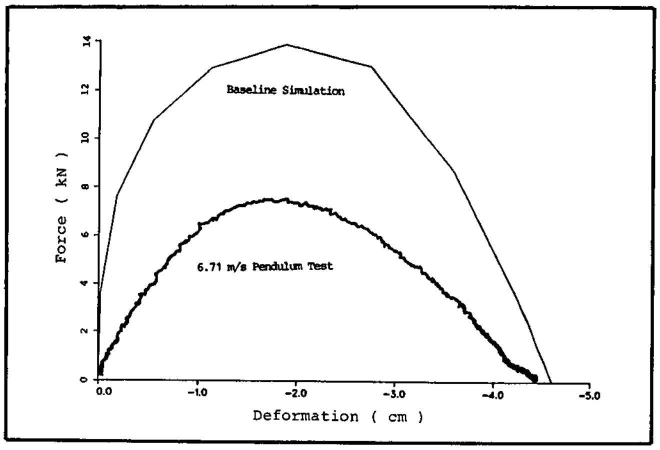

Figure 12. The dynamic force-deflection of the SID thorax

thoracic spine segment and, is an appropriate measure of the rib deformation. The last index used in this work is the Viscous Criterion(VC), which is the product of the chest compression(C) and the rate of compression(V). This index was originally developed by the Biomedical Science Department of the GM Research Laboratories for assessing the thoracic injury due to frontal impact[7]. Currently, there are efforts to extend it to the side impact trauma. The necessary kinematic information for evaluating the VC can also be obtained from the number one spring-damper response. Figure 13 illustrates the relationship among these three response parameters. It can be observed that, for relatively high speed impact as those considered in this study, the shape of VC is primarily dominated by the shape of V.

Parameter Study

This study will be concerned with the following parameter variations: the impact velocity, the side structure stiffness, the clearance between the occupant

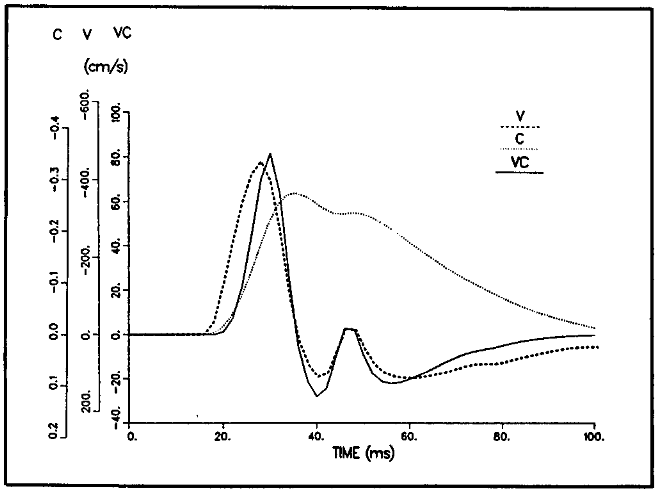

Figure 13. The SID thorax impact response

and the door, and the inclusion of the padding material on the door inner panel. The occupant response to the parameter variation will be evaluated with the injury criteria discussed in the previous section, i.e., chest acceleration, TTI, rib deformation, and VC. Figure 14 shows that all these indices increase as the impact velocity gets higher although different sensitivity was found for each of them. The VC index appears to possess the most change while the TTI has the least amount of change in terms of the percentage of the baseline response. The reduced sensitivity of TTI is due to the invariant age term. Figure 14 also depicts the amount of door intrusion and the door impact velocity on the dummy chest. These two parameters seem to have a linear change to the velocity variation.

The door stiffness variation is accomplished by multiplying the rotational spring constants and the damping coefficients of the side structure linkage joints by a factor ranging from 0.6 to 2.0. These multipliers are used to conveniently alter the door resistance to impact. No attempt was made to relate these numbers to the practical door design. It can be observed from Figure 15 that stiffer side structure would reduce door intrusion and impact velocity on the occupant chest. All four indices exhibit reduction in injury severity as the door stiffness increases. However, a stiffer door design usually requires excessive mass and sacrifice of the interior space[2,3].

In the baseline run, the occupant is about 8 cm away from the door inner panel. This spacing is varied from 0 to 22 cm in the clearance variations.

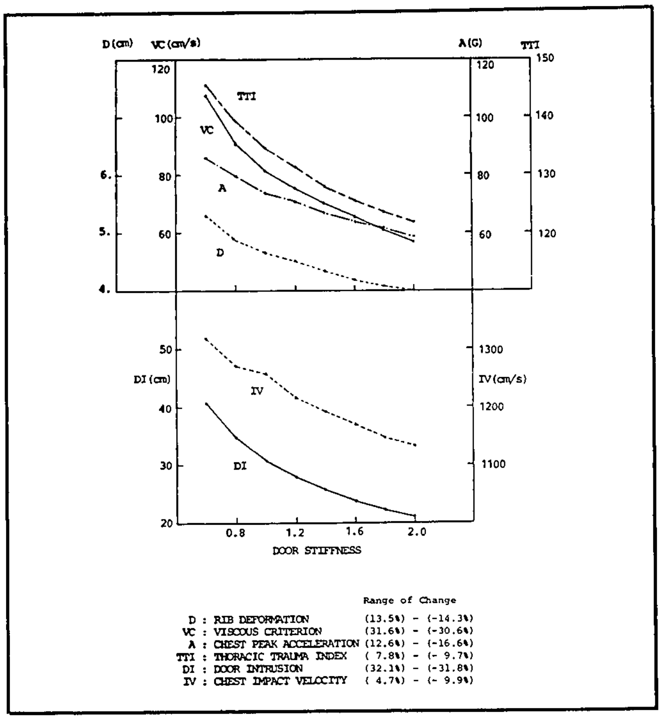

Figure 15. Result of the side structure stiffness variation

Figure 16 shows that more distance from the door inner panel appears to be advantageous in injury reduction. The door intrusion is not significantly affected by the occupant position. Between 0 and 8 cm the impact velocity is lower for smaller clearance. However, there appears to be no injury reduction for

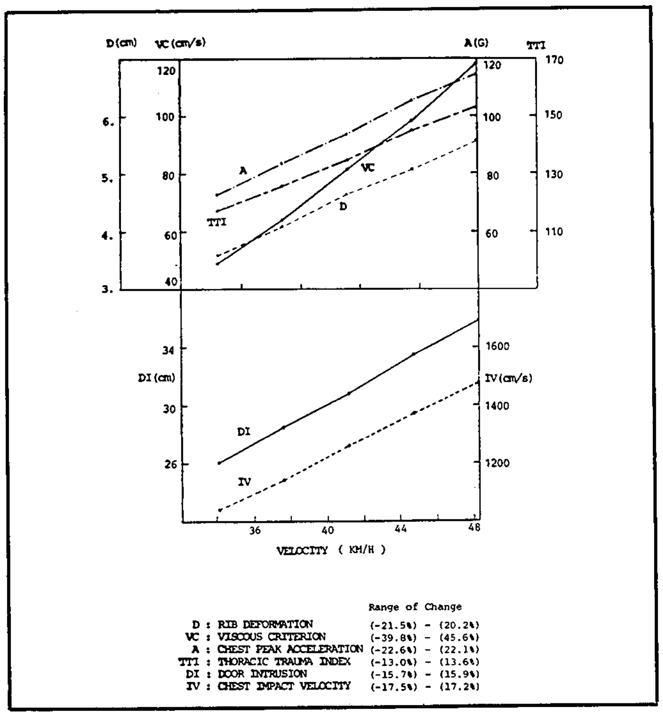

Figure 14. Result of the impact velocity variation

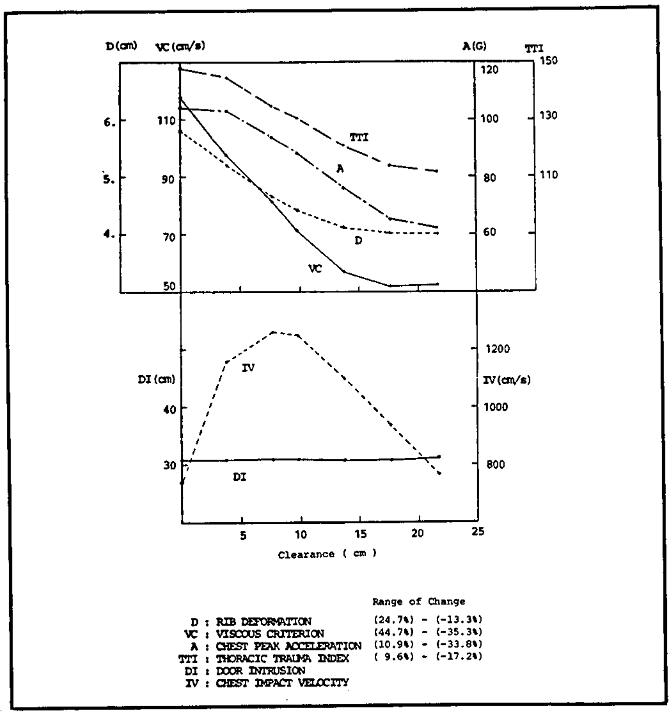

Figure 16. Result of the clearance variation

early contact as shown by the moderate increase on TTI and A, and significant increase on VC and D. This is due to the fact that the occupant contact has little effect on the door velocity.

In the assessment of the padding material, two schemes were used in the simulation. The first variation involves the change of padding thickness for a fixed padding material as demonstrated in the upper half of Figure 17. The occupant is positioned about 14 cm away from the door inner panel with different padding thickness used in each variation. The steeper slope in the second portion of the curve represents the hard contact with the door inner panel and the smaller slope in the first portion of the curve is the padding contact. This scheme would isolate the padding effect to give a true padding assessment as compared to maintaining the spacing between the occupant and the padding surface by moving the occupant away from the door for each padding thickness increment. The latter scheme would result in a benefit for the occupant from the additional space between him and the door inner panel, as illustrated in the previous clearance variation, which is going to obscure the padding effect. The second variation involves fixing the padding thickness, 8 cm in this case, while changing its stiffness as shown in the bottom half of Figure 17. In the extreme cases, no padding is represented by the zero slope in the first portion of the curve, and the stiffest padding is assumed to have the same contact characteristics as the door inner surface.

An interesting phenomenon was found in Figure 18 for the padding thickness variations. As more padding material is used between the occupant and the door inner panel, it is accompanied by a decrease of the acceleration-based indices and an increase of the deformation-based indices. The mechanism behind this is believed to be that the acceleration-based criteria are related to the peak force level while the deformation-based criteria are related to the peak force level as well as the contact duration. With the increase of the padding thickness, the impact force on the occupant chest starts earlier with a longer duration. This effect induced more rib deformation and greater chest viscous response although the acceleration level is reduced due to a softer contact on the padding compared to the hard door panel. This result implies that the use of padding material should be exercised judiciously. Any attempt to reduce the acceleration level by the use of padding must consider the potential for an unacceptable level of chest deformation, and vice versa.

Another observation can be made for the impact force where an increase in peak value is found from 0 to 4 cm padding thickness. This can happen if the use of padding does not reduce the maximum penetration in the thorax/hard door contact by an amount greater than the horizontal distance between these two force-deflection curves.

The padding stiffness variation shows higher stiffness would introduce more deformation as demonstrated in Figure 19. The acceleration-based criteria seem to indicate an optimal padding stiffness for lowest injury severity. The contact force between the thorax/padding surface also indicates a similar trend. This implies that the selection of the padding material must be carried out with an understanding of its effect on both the acceleration and deformation. The

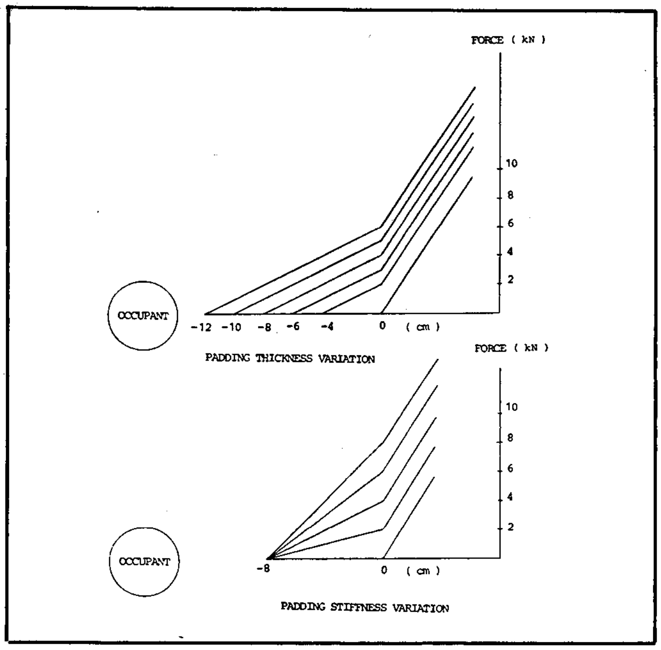

Figure 17. Padding thickness and stiffness variation

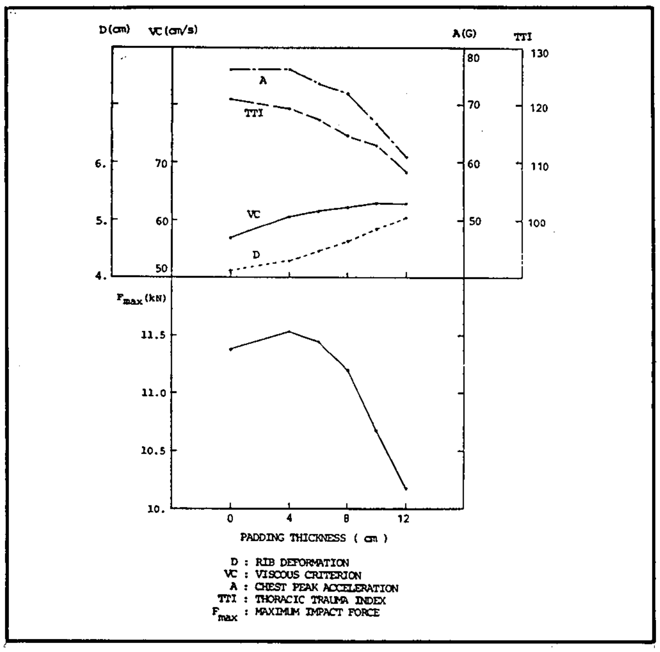

Figure 18. Result of the padding thickness variation

vehicle structure characteristics as well as the impact conditions would also influence the padding material selection.

Experiment Confirmation

With the above observations from the model parameter study, efforts were made to seek experimental evidence for evaluating the predictions. The most systematic data involving these parameter variations were found in a test series conducted by the Motor Vehicle Manufacturers Association (MVMA). The test configuration for this series is shown in Figure 20, where a stationary 1985 Ford LTD 4-door sedan with a Side Impact Dummy (SID) in the right front passenger position was struck at an oblique angle by a NHTSA moving barrier with a deformable energy absorbing impactor surface. The wheels of the moving barrier were crabbed at an angle of 26 degrees so that the barrier front was parallel to the struck vehicle side panel. With the impact speed of 54 km/hr(33.5 mph), this test configuration was intended to simulate a 48.3 km/hr(30 mph) striking vehicle and a 24.1 km/hr(15 mph) struck vehicle.

This series consisted of 8 variations in the vehicle/occupant parameters with each test repeated once, a total of 16 tests. The variations included: (1). the side structure stiffness; (2). the occupant seating position; and, (3). the use of padding material on the door inner panel. The side structure stiffness variation consisted of a baseline and a modified structure in which beams were welded at various side locations for reinforcement. Two occupant seating positions were used in the test: (1). near, in which the dummy was placed against the door inner panel (or the padding surface); and (2). far, in which the dummy was 5 inches away from the door inner panel (or the padding surface). The padding used in the test was 5 inch thick Arcel 506 directly attached onto the door inner panel. Thus, the near side occupant in the padded test cases would have the same position in the vehicle as the far side occupant in the unpadded test cases.

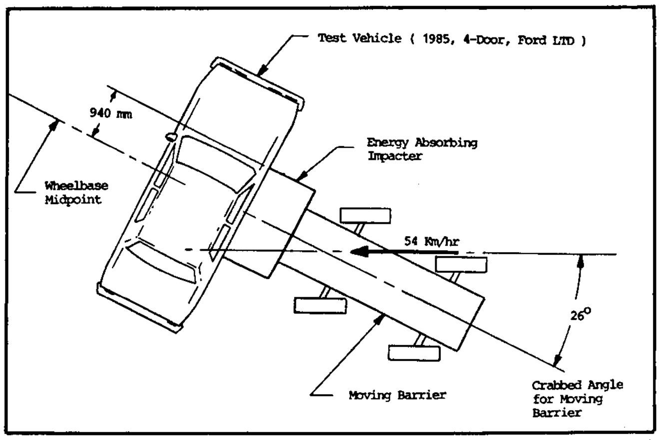

Figure 20. The MVMA side impact test configuration

A summary of occupant response in terms of TTI and rib deformation is given in Figure 21. It should be pointed out that it is well known that the rib deformation of the SID has not been validated against the cadaver thorax behavior. This implies that the absolute number of the SID rib deformation may not be the same as the human response. However, the response trend should still be valid for exercising the test-to-test comparisons. In order to better visualize the effects of each parameter variation and to compare them with the model prediction, three tables were derived from Figure 21. Table 2 is the assessment of side stiffness effect between the baseline and the

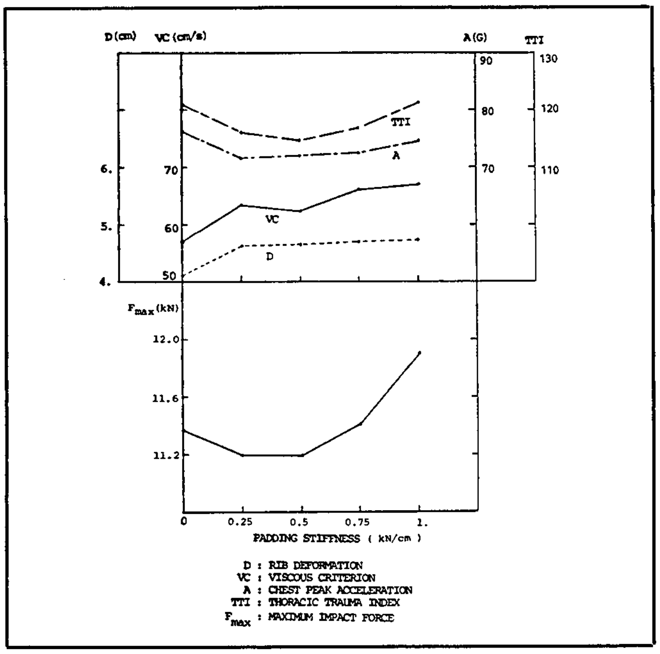

Figure 19. Result of the padding stiffness variation

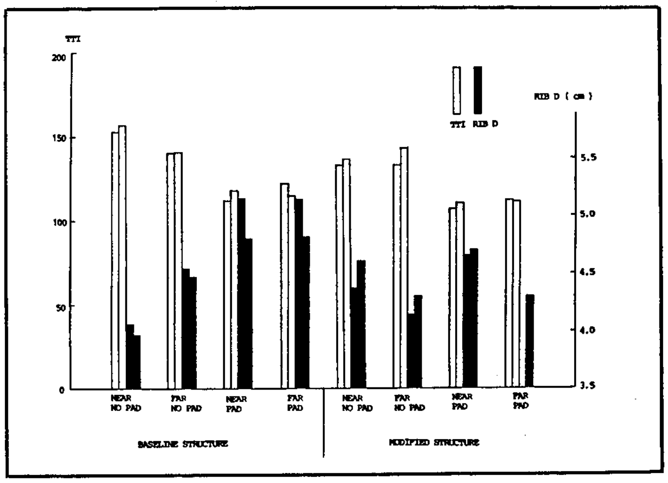

Figure 21. The MVMA side impact test results

Table 2. Door stiffness effect in the MVMA tests.

	BASELINE			MODIFIED		
	TTI	RIB D		TTI	RIB D	
N NP	152.4 156.4		▶	131.5 136.9		●
		4.064 3.962	◀		4.369 4.597	X
F NP	140.3 141.0		?	133.8 143.5		?
		4.547 4.470	▶		4.140 4.293	●
N P	112.4 118.4		▶	107.8 111.2		●
		5.156 4.801	▶		4.648 4.699	●
F P	122.4 115.2		▶	113.0 112.0		●
		5.156 4.826	▶		x 4.293	●

▶ Greater Than
◀ Smaller Than
● Same Trend as Simulation
X Opposite Trend to Simulation
? Inconsistent Data

modified structure. The model predicted that a stiffened structure should reduce the injury indices no matter whether acceleration or deformation is used as the criteria. Since each test was repeated once, both measurements were included in the table. If both numbers in the baseline case were greater (or smaller) than those in the modified structure case, a "greater than" (or "smaller than") symbol was used; otherwise, a "question mark" was used to indicate inconsistent data. The last column was used to designate whether the test data agrees with the model prediction. A circle indicates agreement, a cross indicates contradiction, and a question mark indicates inconsistent test results. Table 2 shows most test results confirmed the model prediction except one contradiction and one inconsistency.

The effect of clearance between the occupant and the inner panel was presented in Table 3 with the same format. This subset yields three confirmations, two contradictions, and three inconsistencies. The high percentage of inconsistent data seems to indicate that the test configuration is not sensitive to the particular clearance variation adopted and therefore test variation may be significant.

For observing the padding effect, it has been indicated that the appropriate comparison would be between the far no-padding and the near padding since these two tests have the same occupant position in the vehicle. The only difference is that, in one case there is 5 inch space between the occupant and the door inner panel while in the other case this space is used by a 5 inch pad. Table 4 indicated that the test data has totally confirmed the model prediction, i.e., a decrease in TTI and an increase in the rib deformation with the use of padding.

Summary

The use of mathematical model allows the study of a wide spectrum of impact conditions and parameter

Table 3. Clearance effect in the MVMA tests.

	NEAR			FAR		
	TTI	RIB D		TTI	RIB D	
B NP	152.4 156.4		▶	140.3 141.0		●
		4.064 3.962	◀		4.547 4.470	X
B P	112.4 118.4		?	122.4 115.2		?
		5.156 4.801	?		5.156 4.826	?
M NP	131.5 136.9		?	133.8 143.5		?
		4.369 4.597	▶		4.140 4.293	●
M P	107.8 111.2		◀	113.0 112.0		X
		4.648 4.699	▶		x 4.263	●

variations, which is difficult to do experimentally. This study demonstrated that with proper modeling, the essence of a complicated physical event can be captured and analysed. The door velocity is an important factor in considering the side impact occupant protection. For the particular vehicle examined in this study, it has considerable variations with an early spike within the time span of door/occupant contact. The fact that this velocity variation is a characteristic of the side structure and is practically not affected by the occupant impact has two immediate implications: (1). Delay of door/occupant contact to avoid the early spike would reduce injury. (2). If sub-system modeling involving only the door and the chest is used, it might be desirable to use a typical door velocity profile as the "forcing function" instead of a constant velocity impact.

The thoracic injury criterion is one of the most crucial issues in the side impact study. This investigation examined both the acceleration-based as well as the deformation-based criteria in the parameter variation study. In some circumstances, both type of criteria give the same indication with regard to injury reduction. Yet most interestingly, conflicting conclusions were observed in certain instances, which reveals the fundamental difference between the two criteria.

The predicted injury potential in terms of both the acceleration and the deformation seems to be an almost linear function of the impact velocity of the striking vehicle within the range of 21 to 30 mph. Stiffer side structure would reduce the door intruding velocity and the amount of intrusion and, thus,

Table 4. Padding effect in the MVMA tests.

	FAR NO PADDING			NEAR PADDING		
	TTI	RIB D		TTI	RIB D	
B	140.3 141.0		▶	112.4 118.4		●
		4.547 4.470	◀		5.156 4.801	●
M	133.8 143.5		▶	107.8 111.2		●
		4.140 4.293	◀		4.648 4.699	●

reduce occupant injury. However, stiffening the side structure enough to make a significant difference without excessive door weight might be difficult to achieve. More spacing between the occupant and the door is also beneficial since this causes the door/occupant impact to occur at a lower velocity.

In the evaluation of padding effect, the acceleration-based criteria indicate that padding on the door inner panel would reduce injury severity while the deformation-based criteria give the opposite indication. From a design standpoint it is necessary to determine whether the acceleration or the deformation is the more relevant measure in assessing thoracic injury, or both need to be considered to achieve a balance. In that case, any attempt to reduce one injury measure should not result in an increase of the other measure beyond the tolerance level. Furthermore, from the acceleration standpoint, the selection of padding material is important to achieve the lowest possible occupant acceleration response.

Acknowledgement

This author would like to thank Mr. J. Stocke of General Motors Current Product Engineering for providing the experimental results used in this study. Special thanks go to Dr. J. Fleck for consultation on the use of Cal3D simulation program.

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Towards Optimised Side-Impact Vehicle Structures

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Abstract

During side-impact accidents there are few structural components between the vehicle occupant and the outside environment. Nevertheless, the number of structural components involved in resisting intrusion of the bullet vehicle is potentially large. Assessing which components to modify so as to improve both the structural performance of the vehicle and reduce potential occupant injuries is a task performed most effectively by computer simulation.

Cranfield Impact Centre have developed techniques for predicting the structural performance of a car during side-impacts and the resulting car motion and occupant kinematics. First, a hybrid approach to predicting the structural behaviour of car side-structures is used. Here, the structural and energy absorbing components and joints are tested to determine their non-linear behaviour (providing a data bank of information which can be invaluable in subsequent design procedures). This non-linear infor-

mation is then input as data to the CRASH-D program to analyse a coarse finite element mesh of the side of the car and hence predict its structural characteristics during side-impacts. The motion characteristics of the side-structure and car centre of gravity can then be predicted. Finally, the kinematics of the occupant and his likely level of injuries, due to his interaction with the intruding side-structure, can be predicted using the Calspan CVS program.

These simulations allow the influence of a large number of structural variations to be quantified rapidly and cheaply. Consequently, structural improvements which require changes to the least number of components can be identified and impressive improvements in side-impact performance can be achieved without gross increases in car weight.

This paper demonstrates the use of these techniques and discusses measures which can improve side-impact performance.

Introduction

During car to car side-impacts there are few structural components between the occupant and the outside environment. Nevertheless, the number of structural components involved in resisting intrusion of the bullet vehicle is potentially large. Despite this, the impact characteristics of the sides of cars are considerably inferior to those of the fronts. This lack of compatibility between car sides and fronts—which has been highlighted before (Refs. 1,2,3, and 4)—usually results in the front structure being capable of absorbing between 2 and 5 times as much energy as the side-structure of current passenger cars.

As a consequence, during side-impacts with contact centred on the passenger compartment, the passenger sat on the struck side of the car is hit by the intruding side-structure and accelerated across the car. This occurs before the lateral velocity of the car as a whole becomes appreciable. The intrusion of the side-structure is the main cause of injuries to the occupant.

Reinforcing the side-structure, to reduce the incompatibility between car fronts and sides, will decrease the relative movement of the side-structure and the car. However, it may cause the lateral velocity of the car to increase at a greater rate. The rate of increase of car lateral velocity may not be influenced too markedly by moderate reinforcements to the side-structure, but reinforcements of a heavy-weight nature i.e. where the mass of the reinforcements becomes significant relative to the mass of the car, will probably maximise this rate. As a consequence the acceleration of the car throwing the occupant against the side-structure will become the main cause of injuries to the occupant.

Therefore, to minimise occupant injuries the strength of the side-structure must be such as to reduce intrusion and in particular the rate of intrusion, yet not so high as to maximise the rate of change of car lateral velocity. In this way the relative velocity between the side-structure and the occupant at first contact will be minimised as will the acceleration experienced by the occupant.

To assess the correct level of side-structure strength for a particular car, a straight forward computer simulation will provide quick and cost effective answers, whether the side-structure to be optimised for side-impact is that of an existing or new car. The use of a technique is demonstrated here.

Analysis Techniques

Analysis techniques for simulating the structural behaviour of car side-structures during impact fall broadly into two categories. First, those which are purely analytical and those which use a hybrid approach combining testing and analysis.

Into the former category fall the large general purpose finite element and finite difference programs. These usually require the availability of detailed drawings of the proposed structure of a car. The computer models analysed with these programs are usually large and detailed often having many thousands of degrees of freedom or a large wavefront. As a consequence long and expensive computer runs are needed. In addition, if it is wished to investigate the influence of a number of separate changes to parts of the model it often takes a considerable time before the results become available. In other words, a large model is often incapable of assessing changes in a timescale which is useful to a design team.

In the latter category come the techniques which in contrast to the former, do not need detailed drawings of the structure, are cost effective and quick to run, and give answers in a short timescale—such advantages are useful to a design team in the early stages of formulating a structure for a new car. However, they do need information from simple bending and torsion tests on the main elements of the side-structure which form the major load paths through the car structure during a side-impact. Although this may at first appear a disadvantage, ultimately a data bank of information can be compiled which is extremely useful for selecting and combining structural properties in computer analyses. Indeed the characteristics of a large number of previous components can be used in a model to perform a parametric investigation and so 'firm up' the structural elements of a new car at an early stage of the design process.

The CRASH-D finite element code written by Cranfield Impact Centre is a program which falls into this latter category. Only the beam elements which form the main load paths through the structure need to be modelled, for which information is available from the practical tests. The resulting coarse finite element model can then be analysed quickly and cost effectively. Subsequently, the dynamic behaviour of the side-structure, and the centre of gravity of the car during an impact can be determined, and finally the kinematics of an occupant predicted.

Use of the Technique

To show the use of this technique, simulations were performed to represent a car being struck on the side by a CCMC mobile deformable barrier travelling at 50km/h perpendicular to the longitudinal axis of the car. The barrier had a ground clearance of 300mm and impact was centred on the passenger cell—without a direct A-pillar contact—thus representing a 'worst case' situation.

The non-linear behaviour of all the components were determined from static bending and torsion tests, then this information was assembled with the geometric details of the side-structure for analysis by the CRASH-D program. An idealisation of the side-structure and the impact area are shown in Figure 1. The model includes all the beam elements and significant sheet metal areas which are likely to influence the collapse mechanism of the side-structure—previous work had shown that a model with a mesh as coarse as this could be used(Ref. 5). The door was not included at this stage, as its properties were not considered to be significant compared to the rest of the structure.

Although all the nodes within the geometric shape of the barrier are initially in contact with it, in view of the coarse model used for the non-linear analysis is was considered adequate to simulate the loads imposed on the vehicle as acting at the extremities of the barrier. An allowance was made for the edges of the barrier crumbling during an actual test. The forces resisting intrusion were generated by nodes 8 and 20 (Figure 2.), with a total force of 24 kN. The turning points on these characteristics are associated with the formation of plastic hinges in the upper and lower B-pillar, waistrail, sill and compression failures of those elements representing the rear seat pan. These characteristics together with data concerning the masses of the car and the barrier, the suspension and inertia characteristics of the car, and the stiffness of the barrier face, were used to simulate the dynamic interaction of the car and barrier. The acceleration and velocity characteristics of the side-structure and centre of gravity of the car were then determined.

This information together with data describing an APROD side-impact dummy, the seat, floor and side-structure of the car were analysed by the Calspan Crash Victim Simulation (CVS) program to determine the kinematics of an occupant. During this simulation the occupant was hit by the side-structure whilst it was moving at its maximum velocity (12.5m/s) and was thrown towards the centre-line of the car (Figure 3). The accelerations experienced by the occupant (Figure 4) were significantly higher than those for front impact. The maximum lateral pelvis velocity was very similar in magnitude to the maximum side-structure velocity, whilst that for the chest was 85% of that velocity (Figure 5).

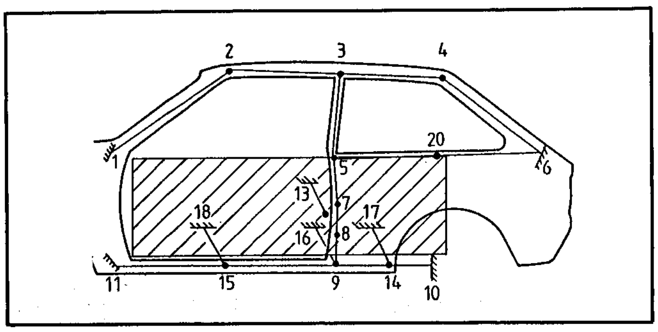

Figure 1. Idealisation of basic structure and impact area

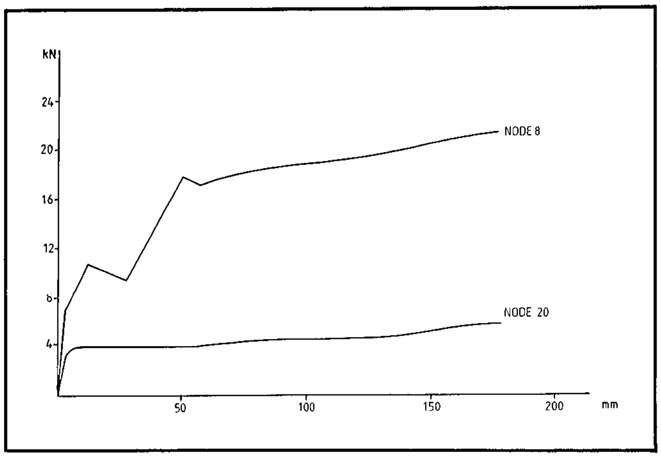

Figure 2. Load deflection curves for basic vehicle

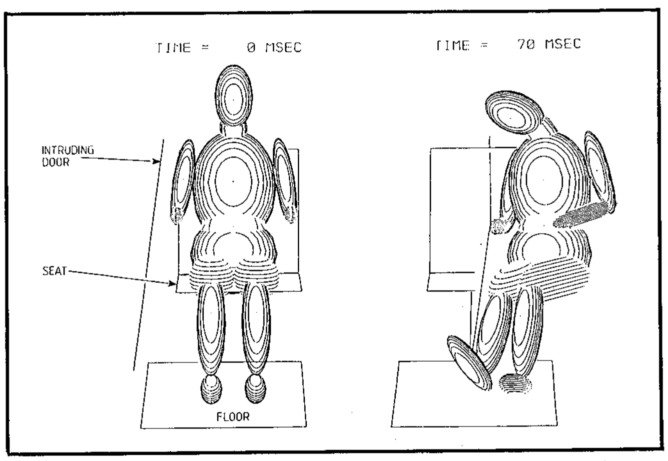

Figure 3. Movement of occupant across vehicle

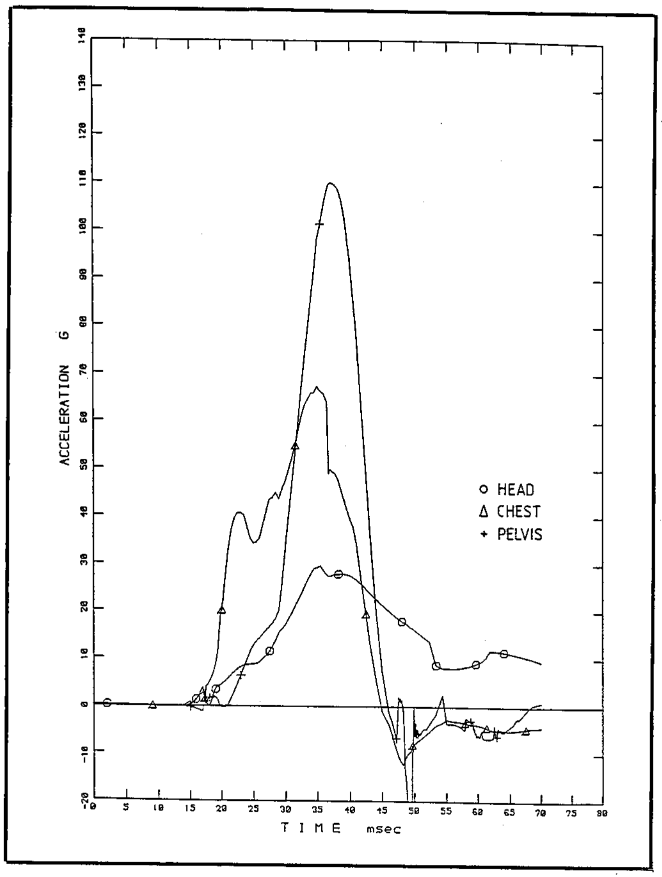

Figure 4. Lateral accelerations of head, chest and pelvis. CCMC barrier to 3 door, basic vehicle

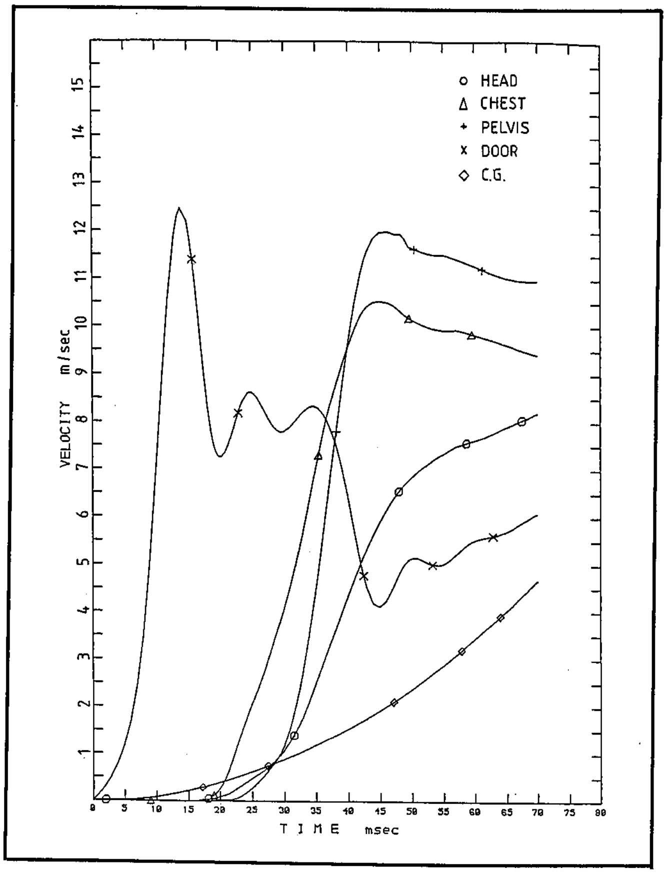

Figure 5. Lateral velocity of head, chest, pelvis, door and vehicle CG. CCMC barrier to 3 door, basic vehicle

Comparison with Tests.

After completion of the simulations the results were compared with those from a staged side impact between a CCMC barrier and the car. The velocity characteristics for the car centre of gravity and the impacted B-pillar from tests and simulations show a reasonable comparison, Figures 6 and 7. Also the velocity and acceleration characteristics of the occupant show a close agreement with the test results Figures 8, 9 and 10. This is despite the coarseness of the model for the non-linear analysis, and the necessary modelling assumptions throughout the simulations.

Discussion.

Using these techniques it would now be possible to undertake parametric studies which would allow the influence of a large number of structural variations to be quantified rapidly. This would allow existing components, which lie outside the impact area, to contribute more to the behaviour of the side-structure—for example the A-pillar and roof—and in conjunction with new structural components to provide additional loads paths through the structure. Consequently, com-

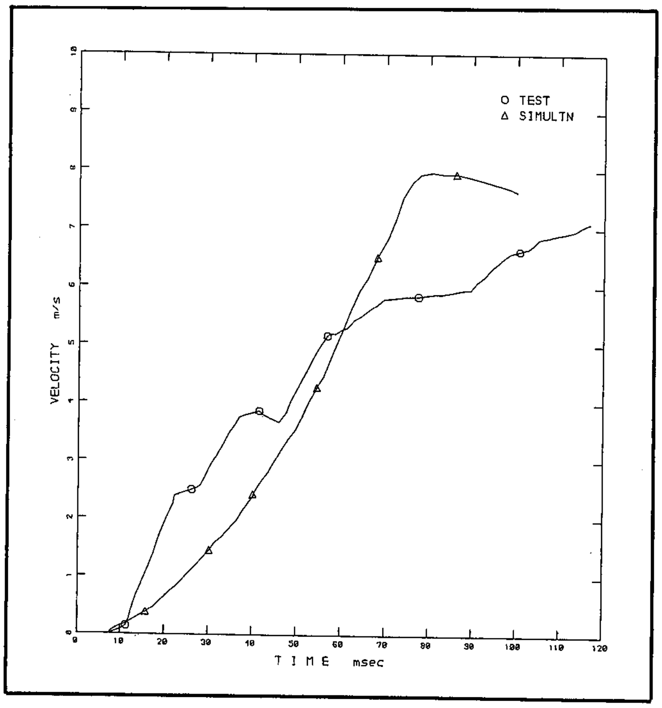

Figure 6. Lateral velocity of car centre of gravity from test and simulation

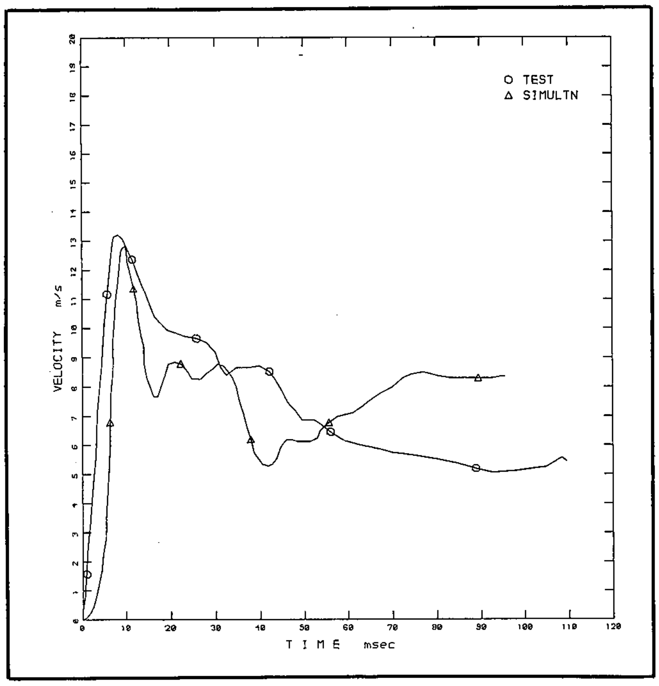

Figure 7. Lateral velocity of B pillar from test and simulation

ponents providing the major forces resisting intrusion and absorbing energy need no longer be in the immediate vicinity of the occupant, but could be dispersed along the whole side-structure. In addition, all the components in the main load paths could be

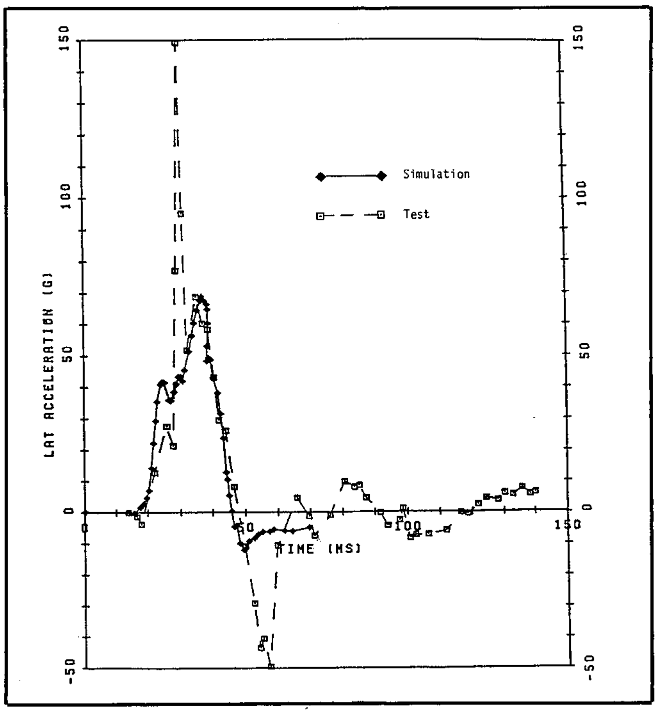

Figure 8. Comparison of chest accelerations from simulation and test

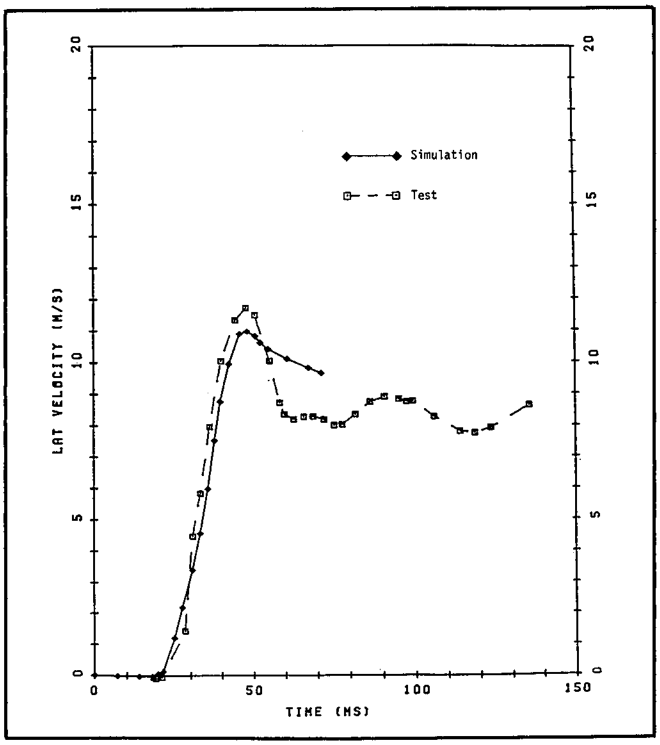

Figure 9. Comparison of chest velocities from simulation and test

made compatible to prevent localised weaknesses from seriously imparing the behaviour of an otherwise 'good' side-structure. As a consequence the occupant would be less vulnerable.

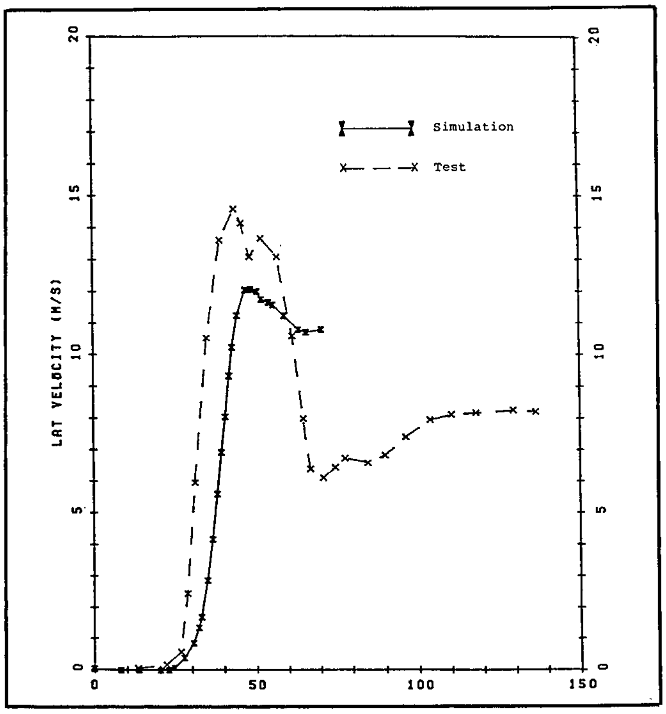

Figure 10. Comparison of pelvis velocities from simulation and test

General Measures to Improve Side-Impact Performance

Redesigning existing components so as to move the causes of bending, axial and torsion failures to other locations can provide significant improvements in side-structure characteristics. Modifications and new components which might add as little as 10 kg weight to a car can make considerable improvements to the side-impact behaviour.

Where redesign of a component is prohibited for reasons of cost a simple measure which can improve collapse behaviour is to foam fill the component. This generally has the effect of improving the energy absorption capabilities of the component after the onset of collapse, rather than improving the elastic properties of the component. However, the density of the foam is all important. Interestingly, foam filled components often also have improved noise and vibration characteristics.

Weight for weight foam filling usually results in a component which is more efficient in terms of energy absorption than the same component with a metal reinforcement.

With careful redesign and modest weight increases it is possible that side-structure speeds during the contact period with the occupant can be reduced by 3m/s. The consequent reduction in potential occupant injuries is obvious.

The use of padding on the inner panel of a door can offer protection to an occupant. However, too often, especially where packaging dimensions are very tight, the padding bottoms out before the occupant has been accelerated to the same speed as the intruding side-structure. If thicker padding were fitted it could reduce the gap between occupant and side-structure and might even worsen injuries if side-structure speed were higher at first contact.

A combination of structural improvements and door padding will probably give the best improvement in occupant protection during side-impact.

Conclusions

1. The use of a hybrid approach involving practical testing and computer simulations can be used to predict the structural behaviour of a vehicle during a side-impact, and the consequences for an occupant. Even with the use of coarse finite element meshes, accuracy is acceptable and offers a means of rapid assessment of structural alternatives.

2. Structural improvements do not need to add significant weight to a vehicle to be effective in side-impact, but they do need to be structurally efficient. Achieving structural compatibility between adjacent components in a side-structure can assist in improving side-impact performance.

Acknowledgement

Thanks are due to Ford Motor Company Ltd. for allowing the publication of the work described in this paper, which was the subject of collaborative research between Ford and Cranfield Impact Centre.

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Development of the European Side Impact Test Procedure and Related Vehicle Improvements

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Abstract

A study has been made of the proposed European test procedure, using TRRL mobile deformable barrier (MDB) faces built to the EEVC specifications and a First Prototype EUROSID dummy. A series of tests using a current small car were compared to assess repeatability. The results from these tests were compared with those from tests on similar cars which had been modified to improve protection in side impacts. This series included tests on the demonstration safety car, ESV 87. Also reported are impact tests using an earlier design of MDB face constructed by Fritzmeier and using a car as "bullet" vehicle. Details of the changes incorporated in the TRRL MDB face are discussed. A computer simulation model has also been used to study the side impact injury process.

The study showed that the test procedure was effective, giving reasonable but not perfect repeatability. The importance of careful control of test parameters was demonstrated some of which have important implications for the test requirements. Both the TRRL MDB face and the EUROSID dummy performed well, clearly showing the desirability of having multiple instrumented ribs and using multiple criteria for assessing thoracic injury risk.

Introduction

The relative importance of injury in side-impact accidents has increased because injuries in frontal impacts have been reduced by improved car design and the high use of front seat belts. Most at risk, in side impacts, are those people seated on the impact side of the car adjacent to the impact area. The priority areas for their increased protection have been shown to be the head, thorax, abdomen, pelvis and femur(1,2,3).

Head injuries are most frequently caused by hitting the cant rail, whereas those to the thorax, abdomen, pelvis and femur are mainly related to contact with the side of the car, usually the door. In the case of femur injury, this is usually due to being bent over some angular intrusion.

In the proposed European test procedure, protection for the head, thorax, abdomen and pelvis would be assessed by a full scale impact test. Because of the variability in the location of head contact, a separate headform test is also specified. At this stage, no provision is made to assess protection for the femur.

The full scale test consists of an impact, perpendicular to the side of the car, with an EEVC mobile deformable barrier (MDB) at 50 km/h(4). The level of protection afforded is measured using a European Side Impact Dummy, EUROSID(5), seated adjacent to the impact.

Within a programme of work coordinated by the European Experimental Vehicles Committee, TRRL has been collaborating on the development of the side impact test procedure, the mobile deformable barrier face and the EUROSID dummy.

In order to understand the side impact injury process, to develop protective measures and to validate the test procedure, a series of impact tests on standard and modified cars has been carried out supported by computer simulation studies. This work has formed the basis of the demonstration safety car, ESV 87.

The paper describes this work, the results obtained and discusses injury tolerance criteria, with particular reference to the thorax.

The Dynamics of Side Impact

Factors Influencing Injuries in Side Impacts

In car to car side impacts, momentum is transferred from the "bullet" car to the "target" car and energy is absorbed by both cars. The rate of momentum transfer and the proportion of energy absorbed by the bullet car may be raised by increasing the stiffness between the side of the target car and its main mass and by reducing the dynamic stiffness of the bullet car's front. In the absence of any bounce this should reduce the lateral velocity of the door and any resulting injuries. If the loads are transferred from the bullet car to the target car, through parts of the structure not immediately adjacent to the occupant, then the stiffness and effective mass of the door will be lower when it hits the occupant. The benefits of transferring the loads from a low bumper to the car's sill have already been demonstrated(6,7).

Although the impact causes the target car to be accelerated laterally, the frictional forces between the occupant and his seat are insufficient to accelerate him with the car. Consequently, the occupant's lateral velocity is still virtually zero at the time he is hit by the incoming door. When struck by the door, the side of the occupant's body is rapidly accelerated to the velocity of the incoming door and the loads imposed on his side are transferred through his skeleton and soft tissue to accelerate the main mass of his body.

The manner in which this happens determines the type and severity of any injuries sustained.

Padding, positioned between the door and the occupant can, by its own collapse, extend the time during which the side of his body is accelerated and reduce peak loads. The padding stiffness needs to be carefully selected to optimise protection over a range of impact conditions.

The Simulation Model

The simulation model (Figure 1), in its present state of development, represents a simple one-dimensional lumped mass system, with similarities to other models which have been reported(8,9,10). In the model, the car's side is based on data from static and dynamic tests on a Morris Marina car, carried out by its manufacturers(11). Particular emphasis has been placed on realistic modelling of the highly non-linear characteristics of the structural 'springs', which represent the elastic, plastic, and collapse phases of the impact response, together with appropriate rebound characteristics when the load is removed. The model is being updated in the light of new information from the current crash tests. For example, the desirability of modelling the door in at least two dimensions, has become clear. This will take account of differences in motion at different heights and will be necessary to take account of multiple load paths into the main car structure.

The occupant is represented by a model of the EUROSID dummy using a single combined rib(12). To date the analysis has been mainly related to thoracic injury, so only the thorax has been modelled. However, recent studies have suggested that the inclusion of low level padding with the deliberate intention of causing early movement of the pelvis could reduce subsequent loadings on the thorax and hence reduce thoracic injury. In future the pelvis will be included in the model. However, there is considerable uncertainty as to how to represent the connection between the pelvis and the thorax. In this respect, it is unlikely that any of the current dummies give a good representation of live humans. A further refinement would be

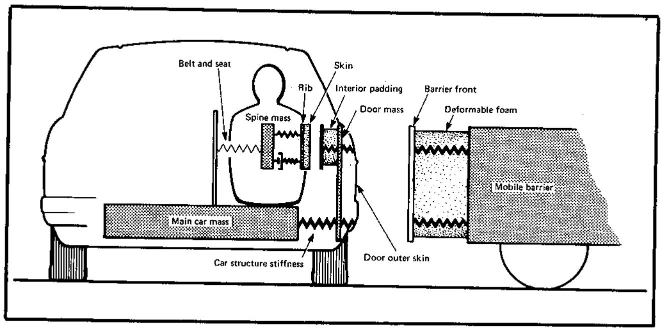

Figure 1. Simulation model of EUROSID dummy in side impact

to include more than one rib. The crash tests have shown that there are considerable differences in the loading of EUROSID's three ribs.

A number of injury tolerance criteria are calculated in the simulation runs, including those used in assessing the data from the full scale tests, peak thorax compression(13), Viscous Tolerance Criterion(14) and Thoracic Trauma Index(15). One of the objectives of the simulation work has been to improve our understanding of the relationships between these criteria, under different test conditions.

In the model, the MDB face is a simple representation of the EEVC force/deflection characteristics. It would be possible to substitute other characteristics to represent impacts with such objects as a rigid pole. Alternatively a car front could be represented using data from an existing frontal impact model.

The main use of the simulation model has been to gain an understanding of the complex processes which occur in side impacts. These are difficult to interpret with the limited data obtainable from full scale impacts. A study of the sensitivity of the injury parameters to variations in the model parameters can also be made. The model parameters varied in the studies include:

1. Initial gap between the door padding and the thorax
2. Thickness of the padding
3. Force / deflection characteristics of the padding.
4. Stiffness of car structure in terms of load transfer to its main mass
5. Stiffness of the MDB face
6. Mass of the front of the MDB face and of the door

Impact Dynamics

The results from simulation runs and full scale tests show that the forces and accelerations imposed on the occupant vary significantly with small changes in the input parameters. This is mainly because of the effects of bounce and the way the structure collapses.

A typical side impact starts with contact between the bullet car and the door. The light outer skin of the door bounces away at an initial velocity which is usually higher than the impact velocity. It then starts to pull on the rest of the car's structure so that it slows down until it is again hit by the bullet car. At the same time, some load is transmitted directly to the car's structure through the A and B-posts, and through the sill. The amplitude of the door oscillation decreases with successive bounces and eventually the bullet car's front and the door attain a common velocity. While the target car's structure is collapsing, the front of the bullet car is also deforming. The details of the door's motion will depend on the relative stiffnesses of the target car's side structure and the bullet car's front(7).

The motion of the inner skin of the door may not follow that of the outer skin very closely and its oscillations may be considerably less. The speed of the door, at the moment of contact with the occupant, can have a major effect on injuries. If it is too fast, injuries result from the initial contact. If it is too slow it will not accelerate the occupant sufficiently to avoid injury from later contacts. As the velocity at first contact is affected by the initial gap, this is an important parameter in determining the degree of injury.

Methods of Minimizing Injury

The traditional methods of minimizing occupant injury have been to stiffen the door and provide padding. Full scale tests and early simulation runs have suggested that padding is more effective than structural stiffening(9,16).

For cars with similar stiffness characteristics, the level of intrusion is a measure of the severity of the collision. Consequently, in real accidents there is a correlation between the amount of intrusion and the severity of injury. However such a correlation does not necessarily imply that for cars of radically different stiffnesses suffering the same severity of impact, the level of injury would still be related to intrusion. Current work supports the contention that there is no such simple relationship.

In any collision, the occupants must eventually experience the same change of velocity as the car they occupy. This is mainly dependant on the impact velocity and the mass ratio. This change in velocity will be affected if the cars separate after the impact. As the car's mass is kept low to reduce manufacturing and running costs, the change in velocity may be regarded as being beyond the control of the designer, unless the amount of bounce can be changed.

The most important changes which can be made are to reduce the initial impulse on the occupant by incorporating padding and to increase the time over which the change of velocity occurs. Reducing the initial impulse may reduce the type of injuries which can be related to the Viscous Tolerance Criterion and the Thoracic Trauma Index. However in general, this will have little effect on the peak thorax compression, which is related to loading over a longer time period.

To maximise the time over which the velocity change occurs, it is desirable to start the occupant moving as early as possible in the impact. Although increasing the stiffness will gain time by making the best use of the crush characteristics of the bullet, too stiff a structue can delay the time at which the door padding makes contact and starts to accelerate the occupant. Simulation suggests that there is an optimum stiffness for any particular set of circumstances, and that excessive stiffness above this optimum can actually increase injuries. The optimum is probably considerably stiffer than existing vehicles.

It can be seen that a limited amount of intrusion, early in the impact, may be advantageous. Having provided sufficient stiffness to make use of the crush characteristics of the bullet vehicle, further improvements should take the form of increased door padding, to reduce the gap between the occupant and the door, to reduce peak forces and to lengthen the period over which the change in velocity occurs.

Outline of Simulation Results

Variation of the model parameters does not, unfortunately, show any simple relationship between parameter values and injury, from which a simple optimisation of vehicle design could be derived. However some general principles do emerge:-

1. Increased structural stiffness would be beneficial up to a maximum of about 2.5 times the stiffness of current vehicles.
2. Provided that there is some padding to reduce peak forces, it would be desirable to minimise the gap between the occupant and the door.
3. The padding should be as thick as possible.
4. Reducing the stiffness of the bullet car's front should have a big effect on reducing injury levels.

The simulation shows that even moderate increases in stiffness of the side structure and moderate thicknesses of padding can produce useful reductions in the Viscous Tolerance Criterion and Thoracic Trauma Index, but have little effect on peak thorax compression. Two alternative conclusions can be drawn from this observation:-

1. If peak thorax compression is a good measure of injury, there is little that can be done to the target car to reduce injury. The amount of stiffening and padding necessary would not be practicable in present styles of production cars.
2. If reductions in Viscous Tolerance Criterion and Thoracic Trauma Index genuinely relate to improvements in protection, then peak thorax compression, by itself, is not a good criterion to use.

These simulation studies suggest that the relationship between thorax compression and injury is complex. Laboratory tests, using simple impactors on cadavers, should investigate the effect of varying the mass of the impacting object over a wide range. Recent studies(17) have shown that this can have an effect on many aspects of thoracic injury. Experimental side impact tests with animals or cadavers have mostly been conducted with standard production cars, so that injuries and the corresponding thorax compression measurements have been providing a comparison of impact severity, rather than a direct comparison of the effectiveness of protective measures.

The European Test Procedure

Development of the EEVC Mobile Deformable Barrier Face

In the light of problems identified in a joint validation programme carried out by INRETS, BASt and TRRL(18), TRRL took on responsibility for redesigning the face of the polyurethane Mobile Deformable Barrier (MDB), to meet the EEVC specifications.

This redesign was aimed at improving control of the collapse mechanism of the face and adjusting its dynamic stiffness to obtain a better match with the EEVC specifications. The redesign involved making the following changes:-

1. Moving the core holes to the leading half of each block, to reduce the frontal mass and improve the collapse characteristics.
2. Improving the adhesive bond between the foam blocks to prevent uncontrolled disintegration, in particular to prevent the vertical expansion of the collapsing lower blocks from pushing the upper layer of blocks away from the impact site.
3. Increasing the strength of the wrapping to improve containment and prevent pieces of foam from breaking free and moving away from the impact zone.
4. Adjusting the foam densities and the hole sizes to improve the force / deflection characteristics.
5. Adopting the use of continuously produced foam to improve repeatability and maintaining closer control over the foam characteristics.
6. Enclosing the complete face in two large polyethylene bags to contain the dust produced as the MDB face disintegrates.

The redesign improved the force / deflection characteristics in validation tests against a dynamometric wall. The curve for the total face lay almost entirely within the specified corridors (Figure 2).

For the individual blocks the fit was not quite as good but repeatability had been substantially improved. The repositioning of the holes was found to have greatly reduced the early inertial peaks, seen in tests on the earlier design, and the face collapsed in a more controlled way. Any tendency for the top layer of foam to be pushed out of the way by the bottom layer was eliminated. One penalty of the improved control was an increase in elastic recovery. Because of this, the after test thickness of the MDB face was above that specified. This is not thought to be important as the recovery does not start to occur until well after the time at which the peak values of the injury parameters occur.

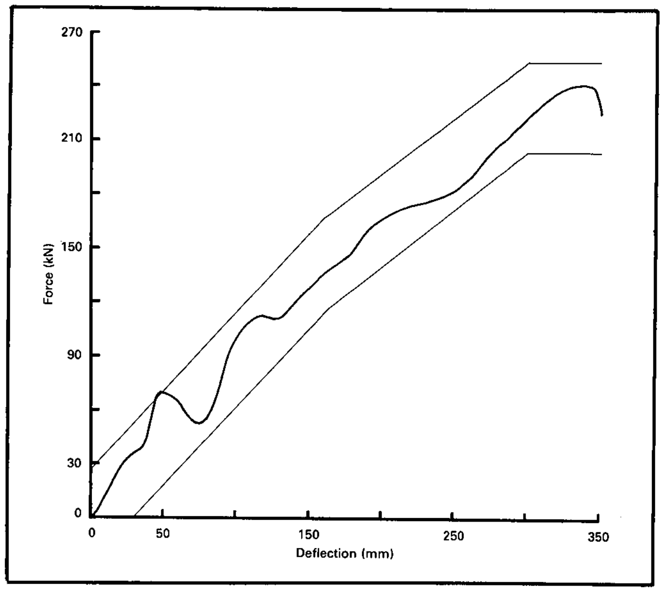

Figure 2. Force/deflection trace for the TRRL built EEVC mobile deformable barrier face

Whether there would be any benefit in further smoothing of the force / deflection trace is doubtful as the frontal stiffness of a real car does not increase smoothly with deflection. Already the force / deflection characteristics of the MDB face are smoother than those of a car. Smoothing removes the peaks which may be responsible for initiating failure of parts of the car's structure. What is wanted is a repeatable force / deflection trace, rather than variable ones within a specified corridor.

The EUROSID Dummy

The EUROSID dummy has been designed to be capable of measuring the parameters considered to be correlated to injury risk as follows:-

HEAD	: Triaxial accelerometers to give peak acceleration and for the calculation of HIC(19).
THORAX	: Thorax compression and rib acceleration at each of three ribs and spine acceleration near to the first thoracic vertebra (T1). From these, all the proposed thoracic injury parameters can be obtained: peak thorax compression, Viscous Tolerance Criterion, Thoracic Trauma Index and peak spine acceleration.

ABDOMEN	:	Switches to detect simultaneous force and penetration of a specified amount.
PELVIS	:	Force transducers at the ilium and public symphysis and accelerometer at the sacrum.

Tolerance Criteria for the Thorax

A detailed study of data from animal tests, cadaver tests, and the investigation of real accidents, shows that there are a number of different types of thoracic injury, which can have serious or fatal consequences. An earlier paper(17) suggested that different types of injury criteria were relevant to the different types of injury.

There are three criteria for thoracic injury in current use which correspond fairly closely to these different types of injury. These are peak thorax compression, Viscous Tolerance Criterion and Thoracic Trauma Index. There is also spine acceleration at T1, but this is not thought to correlate well with thoracic injury.

All three criteria have been calculated in the simulation runs and impact tests. The criteria, their method of measurement in EUROSID, and their likely relevance to injury are described below:-

Peak Thorax Compression, Cadaver drop tests from different heights have been reported, with the side of the cadaver striking either a rigid plate or energy absorbing foam(13). Thoracic injuries, resulting in eight or more rib fractures in the cadavers were shown to be highly correlated to the relative deflection of the half thorax. The original criterion proposed a limit of 35 percent of the half thorax width of 150 mm. Allowing 10 mm compression of the flesh, this would be equivalent to a rib deflectiron of 42.5 mm.

In EUROSID this compression is measured by a transducer on each of the three ribs. In view of the original maximum compression criterion, the ribs were designed with a bump stop which gave a rapidly increasing stiffness above 50 mm deflection. Subsequent work on the bone condition of the cadavers used in the tests has suggested that a higher value of peak compression should be used as the criterion. In this work, six or more rib fractures corresponded to a thorax compression of 48 mm. For the population most likely to be exposed in accidents this would somewhat larger, possibly up to about 60 mm. Compressions as large as this would require a greater stroke from the measuring devices.

It may be noted that, in the tests used to develop this criterion, the cadavers collided with massive and relatively smooth structures. Under these circumstances the ribs are likely to fail in bending at a point remote from the impact, which may not always be representative of the conditions experienced in real accidents. The tests may also have been unrealistic in terms of spread of the impact loads across the chest.

Viscous Injury Criterion (VC). Injuries to the chests of animals and cadavers have been correlated with the normalised compression and the velocity of compression for frontal impacts(14). A good correlation was found between injury risk and the peak value of the instantaneous product of the normalised compression and the velocity of compression, which occurs before the maximum thorax compression is reached:-

VC = normalised thorax compression x thorax compression velocity

The normalised compression is measured as a proportion of the half-chest width of 150 mm. The rib compression velocity is measured in m/s. Using these units, the risk of serious injury increases rapidly for maximum values of VC above 1.0 m/s. In the analysis of the experimental crashes VC was derived separately for each rib of EUROSID. The thorax compression velocity was derived by numerical differentiation of the thorax compression data.

It appears that the Viscous Tolerance Criterion provides a measure of damage in rupture type injuries, caused by distortion and the relative movement of different organs. It probably serves as a proxy for the amplitude of the shear wave propagated through the thorax after impact.

Thoracic Trauma Index. This was based upon a series of tests with cadavers impacted against rigid and padded walls, impacted in cars and struck with impactors. Thorax injuries were correlated against various combinations of rib and spine acceleration. Cadaver mass and age were also taken into account to produce the Thoracic Trauma Index.

TTI = 1.4 x AGE + 0.5 x (RIB-G + SPINE-G) x MASS/165

AGE	=	Age of the occupant in years. In this paper the "kernel" TTI has been used where AGE is set to zero.
RIB-G	=	Maximum rib acceleration, after being passed through a 100Hz Finite Impulse Response filter. The value of RIB -G refer to the maximum of the upper or lower rib but in this paper the value for each of the three ribs of EUROSID is quoted separately.
SPINE-G	=	The maximum spine acceleration, after the same FIR filtering. The spine acceleration should be measured at T12, but in these tests it was calculated indirectly. The quoted data refers to the middle rib, but this is unlikely to give a very different result from that at T12.

MASS	=	The whole body mass, used to standardise with cadavers of different mass. With the EUROSID dummy it has been set to 165.

This gives:-

Kernel TTI = 0.5 x (RIB-G + SPINE-G)

Although the definition refers to rib and spine accelerations, the effect of the heavy filtering is to make the filtered values correspond more closely to the short term change in velocity. The rib acceleration is normally the larger component of TTI, and this can be related to the measure of the compression wave amplitude as a criterion for contusion type injuries to the lung(17). TTI may also be a good measure for impact fractures of the rib, where the fracture is close to the impact point. These are likely to be produced by high velocity impact from relatively light masses, in contrast to bending fractures produced by large masses moving more slowly. Both types of impact to the thorax may occur in side impact accidents.

Vehicles and Full Scale Tests

Add-On Modifications

A series of full scale impacts has been carried out to validate the side impact test procedure. To check repeatability, three standard production cars (W1-W3) were impacted. To see whether the test procedure could discriminate between cars with different levels of protection, the opposite sides of two of the cars were strengthened and padded in identical ways and subjected to a further impact (W5-W6). These tests were also intended to show whether side impact protection could be improved with "add-on" modifications which could be made available for fitting to cars in use.

The modifications consisted of reinforcements and padding added to the car without disturbing the original structure. To strengthen the floor in front of the driver, a glass reinforced plastic tray was bonded into the footwell. A "bolt in" steel frame was used to reinforce the sill and a low door beam was added. To strengthen the floor, to the rear of the front footwell, three cross members were added. The middle section of the B-post was strengthened by the addition of an internal square tube. The rib area padding used was 50 mm thick and its crush strength was about 120 kN/sq m. In the pelvis area the padding was 100 mm thick and its crush strength was about 185 kN/sq m.

The padding stiffness was selected following some preliminary pendulum impacts against the thorax of EUROSID. In the tests different stiffnesses of padding, 50 mm thick, were placed between the impactor and the thorax. In these preliminary tests, the lowest

Figure 3. ESV 87

rib compressions were obtained with cored rigid polyurethane foam of about 120 kN/sq m stiffness.

Modifications Incorporated in ESV 87.

The demonstration safety car, ESV 87, is intended to show what improvements could currently be incorporated in a typical small car, without significantly compromising its functional capacity (Figure 3). These modifications were in addition to those for pedestrian and frontal impact protection(20,21).

As the proposed test procedure does not allow for improvements in compatibility between the bullet and target cars, changes have been restricted to the side of the car.

To increase the rate of momentum transfer, the door stiffness was increased and changes were made to improve the transfer of loads to the main mass of the car. This was achieved by incorporating a tall beam in each door (Figure 4) with modifications being made to ensure that the loads were taken into the A and B posts. As the standard car has sills which are designed to help prevent the doors riding over them, no additional latching to the sill was provided.

The A-posts of the standard car are well supported at waist level but extra support was provided at the floor. The strength of the lower part of the B-posts was also increased. To support the top of the B-post, a lateral roof beam was fitted and to support the bottom, the floor was again strengthed.

The sills were reinforced by the addition of an internal fillet which also resisted rotation about its longitudinal axis. To transfer loads from the sill and to improve the support of the seat mounts, the lateral stiffness of the floor was increased. To prevent the seats from breaking free from the floor, their frames were strengthened and stronger, interlocking, seat runners were used.

By seating people of sizes ranging from about fifth to ninety-fifth percentile in their driving positions, it was possible to determine the amount of padding that could be provided without restricting occupant space

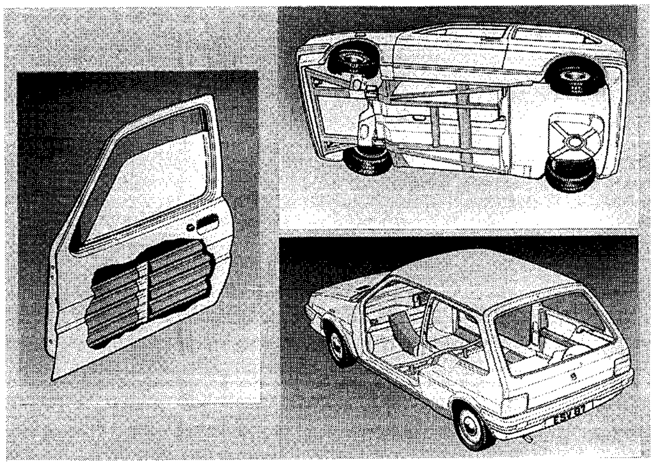

Figure 4. Door, underside and interior structural modifications to ESV 87

too greatly. Door padding of up to 50 mm was thought to be acceptable in this car.

At the time of the full scale test, the work on padding stiffness had not been carried out. For the test rigid polyurethane foam of the lowest stiffness (185 kN/sq m) was used.

Other Full Scale Impact Tests

The impact test on ESV 87 was carried out using a TRRL built MDB face at a ground clearance of 250 mm. When the tests to validate the test procedure were carried out a proposed change to the ground clearance had been made. These tests were performed with the amended ground clearance of 300 mm. As tests on standard cars were carried out at each height, a comparison between the test heights was possible. Two further comparison tests were also performed. One was a car to car test using a second standard car as the "bullet" car. This car was ballasted to the same weight as the MDB trolley. The other test used an EEVC barrier face built, to an earlier design, by Fritzmeier. This MDB face did not incorporate the modifications adopted for the TRRL face.

Results of Full Scale Impact Tests

Repeatability Tests with Standard Cars

Three standard production cars (W1-W3) were impacted by TRRL built MDB faces with a ground clearance of 300 mm. Although the variation in the impact speed was a little greater than that required by the proposed test specifications, the results obtained were fairly consistent (Table 1). The forces exerted on the MDB trolley were similar and any variation can be mainly attributed to the variation in impact speed. The motion of the car was also consistent. Both the lateral acceleration of the main mass of the car and the lateral velocity of the door had very similar time histories (Figure 5).

The values of HIC for both standard and modified cars were all below the proposed criterion level of 1000. However, because there is a large variation in the location of occupant head contacts in accidents, and because small changes in impact location with the head of the dummy can result in a large variation in dummy response (see results for W1,W2 and W3), it is considered that supplementary headform tests would be needed to evaluate a vehicle fully.

The thorax parameters measured by EUROSID were also seen to be reasonably consistent. Each of the tolerance values were calculated for each of the three ribs. It is proposed that the value from the worst affected rib be used in assessing the car. Some variations were seen in the values of each of the parameters measured and the worst affected rib was not the same in each case. This variation may be due to the relative positions of the door's waistline and the ribs on the EUROSID. In this car, the top roll on the door, just below the window, crossed the top and middle ribs of the dummy. Which of these ribs took the greatest load varied in the tests. Provided the highest value is used, this variation is probably not very important. The fact that EUROSID can detect loading at three separate rib positions probably is important. The thorax compression measurements were high and in two cases the measuring devices bottomed out.

Tests on Cars with Add-On Modifications

The modified cars (W5 and W6) were subjected to a similar test. Although there was a small difference in impact speed, the tests were nominally identical. The peak barrier forces were similar, and the measurements of door intrusion gave almost the same value. The pelvic injury criteria and HIC values were also similar for the two tests, confirming that in many respects the tests were indeed similar. However, examination of the chest injury data reveals a different story. In W6 rather larger thorax compressions and increased VC and TTI Values were recorded for the top rib. These difference in injury parameters suggests that there were real differences between the tests. Detailed examination of the vehicles showed that they had collapsed in slightly different ways. This would have caused the top roll of the door to have intruded more violently in W6 than in W5. The reason for the different collapse behaviour has not yet been fully established, but it points a warning to the problems of repeatability in side impact tests. An apparently small difference in the way the two cars collapsed appears to have had a sizeable effect on the measurements which relate to thoracic injury.

Table 1. Summary of Results from Full Scale Impact Tests

Target car	Car(a)			Car + Add-On		Car	ESV 87	Car	
Bullet vehicle	TRRL 300(b)					TRRL 250(c)		Fritz(d)	Car
Test No.	W1	W2	W3	W5	W6	V4	V5	W12	W13
Target car weight (kg)	980	980	980	980	980	915	914	944	962
Impact velocity (km/h)	52.2	49.3	51.1	50.4	47.9	51.8	52. 2	49.3	49.7
Peak force on bullet (kN)	139	130	138	168	155	171	174	113	196
Dynamic door intrusion (mm)	—	304	308	171	—	302	238	341	218
Static door intrusion (mm)	290	243	243	120	129	270	157	268	182
Peak door velocity (m/s)	19.1	18.1	18.7	16.4	16.4	15.7	11.7	19.2	10.6
HIC	780	290	860	320	285	142	190	493	186
THORAX COMPRESSION									
Top rib (mm)	46	40	36	39	49	49	46	45	24
Middle rib (mm)	50+	47	50+	44	48	45	47	50+	22
Bottom rib (mm)	48	43	45	39	43	34	42	38	18
VISCOUS TOLERANCE CRITERION (m/s)									
Top rib	1.27	1.05	0.68	0.46	1.37	0.95	0.70	0.83	0.28
Middle rib	1.52	1.33	1.29	0.60	1.21	1.14	0.87	1.11	0.48
Bottom rib	1.24	1.32	1.16	0.54	0.64	0.88	0.87	0.95	0.34
THORACIC TRAUMA INDEX									
Top rib peak acc. (g)	211	210	169	147	225	225	142	163	122
Middle rib peak acc. (g)	190+	221	180	90	135	235	149	193	157
Bottom rib peak acc. (g)	198	225	185	123	121	186	161	182	148
Spine peak acc. (g)	111	109	110	66	68	75	65	91	60
Kernel TTI (worst rib)	161	167	148	106	147	155	113	142	109
PELVIS (3 ms exceedence)									
Pubic symphysis force (kN)	11.8	12.1	10.8	10.1	10.7	8.0	7.1	—	7.0
Lateral acc.(g)	—	99	87	46	42	78	73	86	100

(a) Standard car
(b) TRRL face at 300 mm ground clearance
(c) TRRL face at 250 mm ground clearance
(d) Fritzmeier face at 300 mm ground clearance

Results of a Full Scale Impact Test on ESV 87

A car modified in a similar way to that of the demonstration car, ESV 87, was impacted with a TRRL built MDB face but in this case the ground clearance was 250 mm, as specified in the test procedure at the time. The results were compared with those from a similar test on a standard car. For the tests, a First Prototype EUROSID was used fitted with plastic skeleton arms. At the time of the tests, the weight of the modified car was one kilogram less than that of the standard car. This was mainly due to the removal of underseal and sound deadening material.

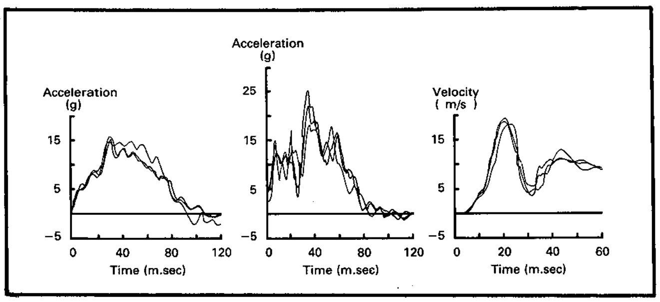

Figure 5. Trolley acceleration, car acceleration and door velocity time histories for impacts with three standard cars (W1 - W3)

The increase in stiffness of the car resulted in a reduction in intrusion and an increase in the rate of momentum transfer. The static displacement of the door, at a point just ahead of the dummy's thorax, was reduced by over forty percent, though the reduction in dynamic displacement was only about twenty percent. The greater elastic recovery of the modified car indicates that care needs to be exercised in drawing conclusions from static intrusion observations.

The velocity time history of the door was modified by the structural changes. The velocity was lower during the early part of the impact with the peak velocity being reduced from 15.7 m/s to 11.4 m/s. During the period of time that values were used in calculating VC and TTI, the door velocity of the modified car had increased above that of the standard car. However, this is probably less relevant than the door velocity when it first impacts the occupant. In ESV 87, the ribs started to accelerate at about 17 ms when the door velocity was about 10 m/s. In the standard car, rib acceleration commenced at about 21 ms when the door velocity was about 14.5 m/s. At this time in the impact, the door acceleration is changing rapidly and more needs to be known about its importance before firm conclusions can be drawn.

Each of the measures for thorax injury were lower in the modified car, with the exception of thorax compression. However, some care is needed in interpreting the data on thorax compression. Some problems were experienced with sticking of the thorax compression pistons with the First Prototype EUROSID thorax. In the test with the standard car, the bottom piston appeared to stop moving and remained stationary for a while giving an artificially low value for compression. Sticking may also have occurred with a number of other compression measurements but close examination of the transducer signals suggests that this would not have had a major effect on the quoted values.

The highest value of VC in the modified car was 0.87, compared with 1.14 in the standard car. With kernel TTI, the highest value from the modified car was 113 compared with 155 for the standard car.

The extent to which these reductions would influence injuries in accidents is not yet known. Although VC has only been validated for frontal impacts there is general agreement that it should also be valid for side impacts. An initial target value of about 1.0 has been suggested. Validation work on TTI suggests that the probability of sustaining an AIS 4 or greater injury increases rapidly at values of between 120 and 130, with age taken into account. If either of these tolerance levels are valid, the improvements should translate into real reductions in injuries.

The work on padding, which was done after these tests, indicates that the padding used in this car was too stiff. Padding with a static compression strength of about 185 kN/sq m was used in the test. It is now thought that a compression strength of about 120 kN/sq m. would be more appropriate. This weaker padding should reduce the injury measurements, though at this stage it is not clear by how much.

So far, less effort has been put into protecting the other regions of the body. However, head accelerations were quite low in both the standard and modified cars. HIC values below 200 were recorded in both tests.

The prototype version of EUROSID used in the tests, was not equipped with switches on the right hand side of the abdomen, so their operation could not be checked in these test. However, in other more severe tests on the left hand sides of standard cars, the switches were not activated.

The pelvis lateral acceleration and the load at the pubic symphysis were quite high, though those for the modified car were a little lower than those for the standard car.

Although EUROSID currently has no instrumentation to measure femur bending, no angular intrusion was produced in the impact with the modified car. It is likely therefore that the bending moment would have been small and that femur injury would not have been a problem in this impact.

Repeatability of the TRRL Built MDB Face

In the three identical tests on standard cars (W1-W3) the MDB faces were seen to perform in a repeatable way. The time history profiles of the force on the trolley, the lateral acceleration of the car and the velocity of the door were consistent from test to test (Figure 3). This shows that the current design of MDB face can produce repeatable results. Where variations have been seen they have been more associated with differences in the car than in differences in the response of the MDB face.

The Effect of Changing the MDB Face Ground Clearance

The results from a test on a standard car, using a ground clearance of 250 mm (V4). were compared with those having a ground clearance of 300 mm (W1-W3). Increasing the ground clearance reduced the peak barrier force significantly. This is because at the increased height the barrier mainly impacts the door and the A and B-posts, whereas at the lower height it also impacts the stiff sill. However, the maximum dynamic and static intrusion measurements were virtually unchanged. Unfortunately, because of instrumentation problems the door velocity information for the impact at the lower ground clearance may not be accurate. However, the best estimate suggests that the peak door velocity was lower than with the higher ground clearance.

Increasing the ground clearance affected the injury parameters measured on EUROSID. The HIC was increased, which may have been due to a change in roll about the car's longitudinal axis. This has not yet been studied in detail. The peak thorax compression was increased slightly but there was little effect on TTI. The VC was also higher. As for the pelvis, the load at the Pubic Symphysis was increased significantly. It should be noted that the make-up of TTI was also somewhat different; the 250 mm high barrier gave a lower contribution from spine acceleration, but a higher contribution from the rib acceleration.

Comparison Between the Fritzmeier and TRRL built MDB faces

Comparing the results of the test using the Fritzmeier built MDB face (W12) with those using the TRRL faces (W1-W3), important differences were seen. A visual inspection of the cars, after the impact, gave the impression that the Fritzmeier MDB face had been more aggressive than the TRRL face. The static deformation was about 15 percent greater and the dynamic deformation was about was about 12 percent greater, when compared with the test using the TRRL face at the same impact speed (W2). However, the barrier force was consistently lower with a peak of only 113 kN compared with 130 kN for the TRRL face.

With the Fritzmeier MDB face, the initial acceleration of the door was greater. This may have been associated with the higher frontal mass imposing a greater inertial force, as the solid foam at the front of the face was decelerated to collapse the cored foam behind it. This inertial force may have initiated failure of the car's structure in a different way. Overall the damage to the car indicated that the Fritzmeier MDB face had put more of its loading through its lower half. This may have been due to the top half being pushed upwards away from the impact, as has been seen in the validation tests.

All the thoracic measurements, with the exception of thorax compression, were lower with the Fritzmeier face, but the pelvis load on the ilium was a little higher. Unfortunately, the data channel recording the load at the pubic symphysis failed to record.

Comparison Between Car to Car and MDB Trolley to Car Impacts

Comparing the impact into a standard car by a similar ballasted car with impacts using MDB faces, clear differences could be seen. Although the peak force on the car was higher than any using the MDB faces, the peak door velocity was much lower, as was the extent of the intrusion. Loads from the car were transferred into the sill much more than with the MDB face, even when its ground clearance was set at 250 mm. The tests using a ground clearance of 300 mm were much less representative of this car. Loads from the leading edge of the car's bonnet were much less and consequently all the injury parameters were much lower, with the exception of the lateral acceleration of the pelvis.

It is clear that the EEVC specification for the MDB face is not representative of cars with low, soft bonnet leading edges, set well behind the bumper line. The changes to car fronts needed for pedestrian protection, such as those incorporated in the demonstration car, will tend to produce even softer bonnet leading edges, set further behind the bumper.

Conclusions

The European side impact test procedure worked well in a series of tests using TRRL built EEVC Mobile Deformable Barrier (MDB) faces. These tests produced a great deal of valuable information, useful both as an aid to selecting the test conditions and for desiging cars to provide protection. The results raised a number of issues of importance. As had been realised the impact situation is complex, but unexpected results may be explained by reference to a mathematical simulation of the impact situation for the thorax.

The test procedure using the EEVC MDB and the EUROSID dummy appears to be sufficiently repeatable for use in legislation, provided that account is taken of the findings that small changes in the car and in the positioning of the impact may have quite large effects.

Rigid polyurethane foam versions of the MDB face can perform satisfactorily and repeatably, but differences in the results using the Fritzmeier and TRRL designs were found. Neither barrier conforms precisely to the EEVC specification, which itself takes inadequate account of inertial effects, but the TRRL barrier is known to conform more closely. It would probably be more satisfactory to specify a particular design, manufactured under closely controlled conditions, rather than continue with the present specification with its fairly wide force / deflection corridor. The ground clearance used is also critical.

Measurements from EUROSID have enabled a valuable understanding to be gained, of the side impact protection situation. The dummy was generally durable, but care is needed with its instrumentation and the rib system pistons should be modified to overcome sticking. The balance of the impact between the pelvic and thoracic regions must be carefully arranged to optimise protection, though it will be extremely difficult to design a dummy with an authentic connection between the pelvis and the thorax.

The pelvic measurements on EUROSID appear to indicate loads satisfactorily. Although injuries to the abdomen and the liver and spleen are reported in accidents, the abdominal event switches did not record excessive impacts in any of these test.

The loading of the thorax and the corresponding likelihood of injury to the lungs and adjacent organs presented a complex picture in these test results. Thorax compression appears to be related to the total impulse applied through the ribs to the thorax. Viscous Tolerance Criterion and Thoracic Trauma Index were also measured and clearly relate to the violence of the initial impact, though they measure different aspects, and are probably suitable for assessing the risk of many types of serious injury. It is probably desirable to record the rib and spine acceleration components of TTI separately as well. It is concluded that for the present all of these measurements should be produced from the records of all tests.

The findings of this series of tests have important implications for the design of cars. Small cars cannot easily be designed to pass this test procedure at present suggested test levels and performance criteria. With improved design and padding the various thoracic criteria, apart possibly from maximum thorax compression can be reduced to acceptable levels.

The critical importance of the height, shape and stiffness of the bullet vehicle is clear. The EEVC MDB face, representing the average of European cars of the 1970's, presents a severe test because of the height and stiffness of its upper half. This is particularly so when it is positioned, as now proposed, 300mm above the ground. The front of the small car, used for this work, gave much less severe loadings.

ESV 87 on display to the 11th ESV Conference incorporates practical modifications and the paper shows the extent to which these are successful in improving side impact protection. A different set of modifications have also been tested with generally similar improvements in protection, although the results are different in detail.

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Subsystem Testing for Head to Upper Interior Safety

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Abstract

A study was conducted to develop test procedures and hardware to measure the head injury potential of vehicle upper interiors. The test procedures were derived primarily from the highway accident data files. The impact velocities were derived by using median delta-v levels from injurious accidents, along with laboratory crash pulses and occupant seating information.

A number of potential approaches for upper interior testing were reviewed and one particular approach—a free motion headform (FMH), was examined. The results of Hybrid III dummy and FMH testing are compared and discussed. The results were found to agree well and the FMH was noted to accurately reflect Hybrid III head response levels.

The Hybrid III and FMH were further studied with math modeling. Ranges of stiffness and A-pillar angle outside those used in testing were examined with the model. Once again the FMH was noted to reasonably represent a full dummy response over the range examined.

Background

It has been estimated that approximately 2,000-3,000 fatalities and 8,000 serious head injuries result each year from head/upper interior impacts. Both Government and industry are involved in reducing the injury likelihood from such occurrences.

Both full scale and subsystem test procedures have been investigated for thoracic protection. There seems to be general agreement that subsystem testing is a preferable method of measuring head impact responses. The reason for this is that in highway accidents, head impacts occur with a variety of upper interior surfaces. It is not practical to produce impacts into such a variety of surfaces using full scale crash testing. In fact, the full scale tests to date do not result in head impacts with the upper interior.

In addition, head surrogates are easily adapted to subsystem type hardware. Versions of hardware for propelling head surrogates into interior surfaces have been documented by several sources in both Europe and the U.S.

Objectives

The main purpose of this study was to identify potential test conditions and examine hardware approaches for upper interior safety testing. Highway accident data, laboratory test data and math modeling are used together to achieve these objectives.

Derivation of Test Conditions From Accident Data

Methodology. The methodology used consisted of two main steps:

1. A computer conducted review of accident databases to identify most frequently occurring impact zones, median △-V's and orientations for serious-to-fatal head/face injuries (AIS> = 3) due to impacts with the vehicle upper interior.
2. An analysis of laboratory crash test data to estimate an appropriate head speed given the △-V derived from accident data.

The combined National Accident Sampling System (NASS) 1981-84 and National Crash Severity Study (NCSS) files were used for this data review.

Accident Database Review

Shown in Table 1 are the total number of cases and the number of injuries for each impact area for the combined data set. The actual number of cases totalled 96, however due to multiple injuries to some occupants, the number of head/face AIS≧3 injuries used for our analysis was 116. It was found that A-pillar contacts account for approximately 57% of the upper interior induced head/face injuries. Contact with the visor/front header and roof side header account for 26% and 13%, respectively, of the head/face injuries. The roof and B-pillar accounted for approximately 4% and 0%, respectively, of the head/face injuries. The programs used to extract the information from the NCSS and NASS databases are contained in Appendix A.

It is found that, for the sets of accident databases examined, the majority of head impacts occurred in three areas - the visor or front header, the roof side header and the A-pillar. A very small number of impacts occur to the roof itself and the B-pillar. Because of this fact, testing of the A-pillar and roof rails was judged to be a higher priority than the roof or B-pillar region.

Table 1. Head/face AIS ≥ 3 injuries by upper interior surface.

Contact Area	Total Cases n	Total Cases %	Total Injuries n	Total Injuries %
A-pillar	51	53.1	66	56.9
Visor/Front Header	28	29.2	30	25.9
Side Header	12	12.5	15	12.9
B-pillar	0	0.0	0	0.0
Roof	5	4.3	5	5.2

In addition to the overall distribution by area of contact, two other interactions were looked at:

- Contact Area by Principal Direction of Force (PDOF)
- Contact Area by △-V (in 5 mph increments)

Each of the interactions will now be discussed.

Table 2 contains the distribution of Contact Area by PDOF. The principal directions of force looked at were 9 to 12 o'clock and 1 to 3 o'clock, representing lateral, oblique and frontal impacts to the forward portion of the vehicle.

The combined PDOFs of 1,2 and 11 o'clock account for approximately 68% of the A-pillar impacts (n = 66). The remainder of A-pillar impacts have corresponding PDOFs of 10 and 12 o'clock (15% and 17%, respectively). No A-pillar impacts occurred at the 3 and 9 o'clock directions.

The majority (93%) of head/face-to-visor/front header impacts (n = 30) occur in 12 o'clock (73%) and 11 and 1 o'clock (10% each) accidents, with the remaining 7% equally distributed between 2 and 3 o'clock accidents.

Approximately 87% of head/face injuries resulting from impacts to the side header (n = 15) occur in oblique impacts (2 o'clock - 40% and 10 o'clock - 47%). The remaining 13% of head/face-to-side header injuries occurred in 9 o'clock accidents.

The frequency distributions for Contact Area by △-V are contained in Table 3. Delta-v ranged from 1 to 60 mph, and for our purpose, were divided into 5 mph increments.

For each area of contact, the △-V cummulative percentage distributions were derived and plotted to obtain the median △-V. Figure 1 shows the △-V cummulative percentage distribution for head/face-to-A-pillar impacts (n = 66). The corresponding distributions for head/face impacts to the visor/front header (n = 30) and side header (n = 15) are shown in Figures 2 and 3, respectively. For A-pillar impacts, the median △-V is 24 mph. The median △-Vs for visor/front header and side header impacts are approximately 28 mph and 21 mph, respectively.

Table 2. AIS ≥ 3 head/face injuries, contact area by PDOF.

PDOF	A-pillar n	A-pillar %	Visor/Front Header n	Visor/Front Header %	Side Header n	Side Header %	Total Injuries
3	0	0.0	1	3.3	0	0.0	1
2	15	22.7	1	3.3	6	40.0	22
1	14	21.2	3	10.0	0	0.0	17
12	11	16.7	22	73.3	0	0.0	33
11	16	24.2	3	10.0	0	0.0	19
10	10	15.2	0	0.0	7	46.7	17
9	0	0.0	0	0.0	2	13.3	2
Total	66	100.0	30	99.9	15	100.0	111

Subsystem Test Conditions

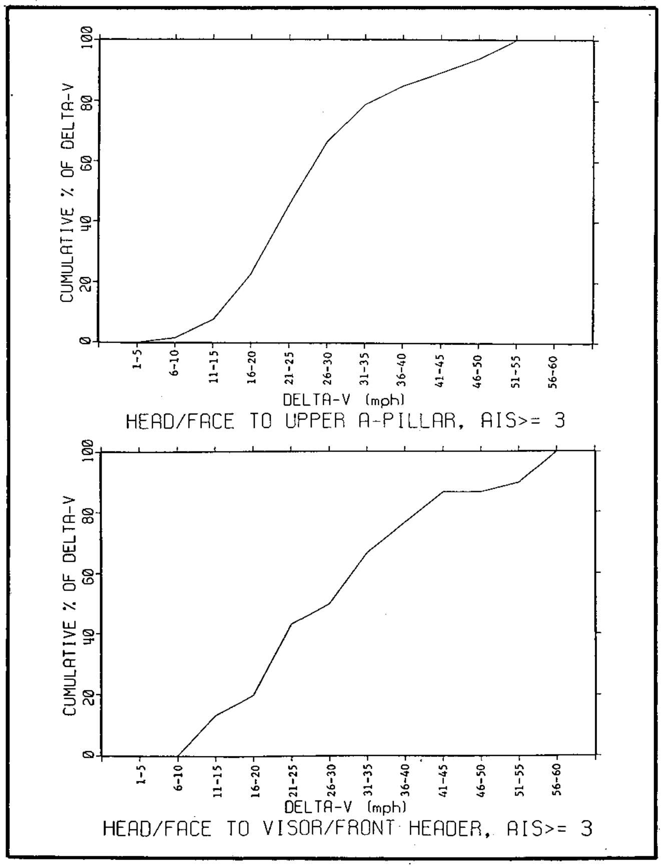
The accident data analysis resulted in the observation that front and side roof rails and the A-pillar regions are contacted more than roof or B-pillar regions in serious injury accidents. Impacts were noted to occur both perpendicularly and obliquely into these vehicle components. A subsystem test into the upper interior would conceptually consist of propelling a head surrogate into one of these interior components at some angle and at a velocity representative of the portion of the injuries desired. The selection of a particular angle need not come from the accident data, but might come on the basis of a worst case, which would be perpendicular to the impacted surface. The selection of a velocity is probably more important and there are likely differing views as to the most appropriate method. An example of how this might be done will be shown.

Impact Velocity. The distribution of Δ V for A-pillar, visor/front header and side roof rail impacts was shown in Figures 1, 2 and 3. The median values for serious to fatal injuries were found to be 24 mph, 28 mph and 21 mph respectively. These values represent the total velocity change of the vehicle compartment during the accident, and do not reflect the velocity at which the head contacted the interior. A procedure was derived to estimate the relative velocity between an occupant's head and the vehicle contact area in collisions of the above severity. This is outlined as follows:

- Laboratory collisions of similar Δ V were used for analysis. It was assumed that the occupant did not interact with the vehicle prior to the time of head contact, i.e., the head velocity was constant during the time the vehicle was accelerating toward the head.
- The percent relative velocity (between the vehicle and occupant) was plotted against the relative displacement for various collision modes, full frontal, frontal into pole, MDB to vehicle side and side to pole.
- The distance from the head to the contact area was obtained for dummies seated in several vehicles.
- The head contact velocity was derived from the velocity displacement plots, using the measured head distances.

Table 3. AIS $\geq$ 3 head/face injuries, contact area by Δ-V.

Total Δ-V (mph)	A-pillar n	A-pillar %	Visor/Front Header n	Visor/Front Header %	Side Header n	Side Header %	Total
1-5	0	0.0	0	0.0	0	0.0	0
6-10	1	1.5	0	0.0	0	0.0	1
11-15	4	6.1	4	13.3	3	20.0	11
16-20	10	15.2	2	6.7	2	13.3	14
21-25	15	22.7	7	23.3	5	33.3	27
26-30	14	21.2	2	6.7	5	33.3	21
31-35	8	12.1	5	16.7	0	0.0	13
36-40	4	6.1	3	10.0	0	0.0	7
41-45	3	4.5	3	10.0	0	0.0	6
46-50	3	4.5	0	0.0	0	0.0	3
51-55	4	6.0	1	3.3	0	0.0	5
56-60	0	0.0	3	10.0	0	0.0	3
Total	66	100.0	30	100.0	15	99.9	111



Figures 1 and 2. Injury Delta-V distributions for upper interior impacts

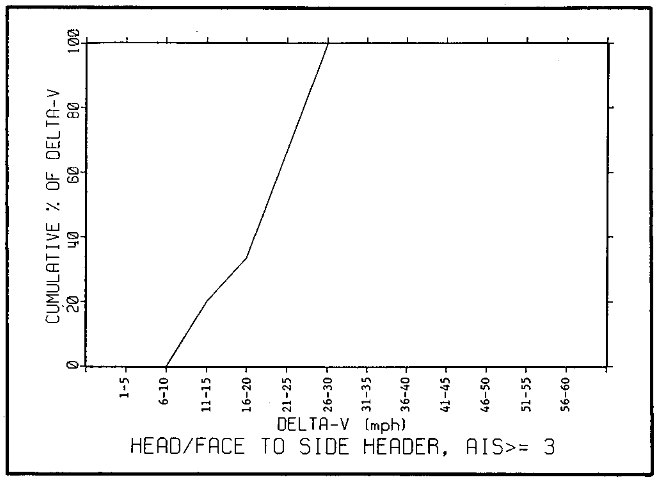


Figure 3. Injury Delta-V distributions for upper interior impacts

As noted from the procedure, two types of information were needed to estimate a head impact velocity. The first is a plot of velocity change (or % velocity change) vs. head/vehicle displacement, and the second is a measured distance from the head to the component of interest in a particular vehicle. Table 4 lists the laboratory crash tests conducted at the Transportation Research Center of Ohio, which were used for estimating head impact velocities. An attempt was made to find collisions which had the same or very similar delta-v as the highway data, 24 mph, 28 mph and 21 mph. A judgement had to be made as to which collision mode to use for A-pillar impact speed analysis. The available laboratory collisions were frontal and side collisions, none of which resulted in A-pillar contact. It was judged that an oblique impact to the corner of a vehicle would most likely cause the occupant to move toward the A-pillar. Such an impact was sought but not found. A frontal offset impact was used for this purpose, and the assumption was made that this would render an acceptable estimate of a corner impact response. The example crash pulses for header/front rail and side rail impacts were from frontal and side impacts respectively.

The delta-v, displacement plots are contained Figures 4-6. Each plot was obtained by computing the velocity and displacement time histories from the recorded acceleration data. The delta-v time history was then computed as the change from the initial velocity, and the delta-v values were divided by the total delta-v to obtain the percentage of total delta-v time history. This was then plotted against the displacement time history to obtain the percent delta-v, displacement plot.

The distance from a seated dummy to the interior contact areas of a few vehicles is contained in Table 5. The distances to the front and side headers were obtained from crash test reports using the pre-test dummy to interior measurements. These reports did not contain measurements to the A-pillar region. A few such measurements were estimated from an employee seated in available vehicles and the horizontal distance to the pillar roughly measured.

The selected crash pulses were used along with the above travel distances to estimate the head impact velocity in each case. An example of how the velocities were derived will be given for the A-pillar region. The measured distances resulted in an average distance from the head to the pillar of 14.5 inches. Using

Table 4. Laboratory crash tests used for estimating head impact velocity.

TRC Test No.	Approximate Delta-v	Condition	Application
850814	26	Omni/Celebrity Offset Frontal	A-Pillar
850614	30	Omni/pole Frontal, Ctr	Front Rail
841129	30	Fuego/Fuego Frontal, Ctr	Front Rail
840926	30	Celebrity/FRB Frontal	Front Rail
851202	20	MDB/Spectrum Side	Side Rail
840629	20	Rabbit/Pole Side	Side Rail

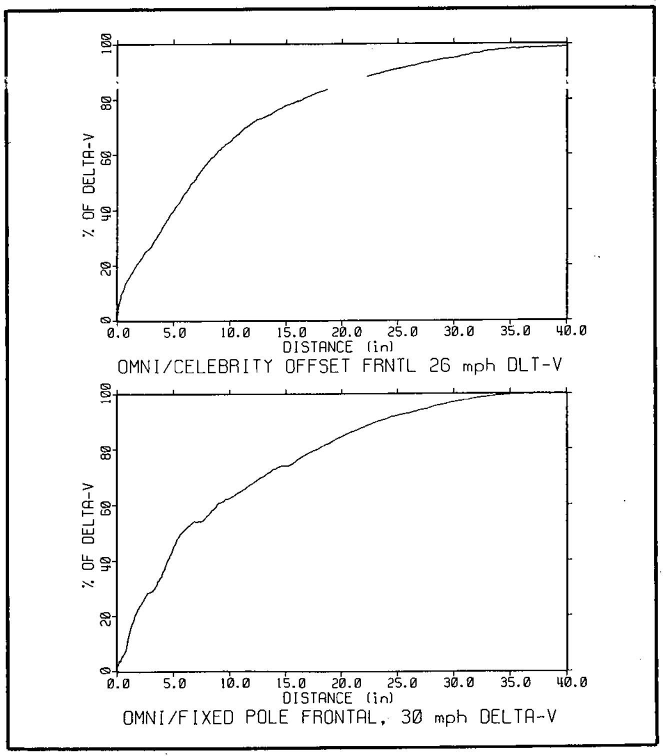


Figure 4. Percent total Delta-V versus seated distance

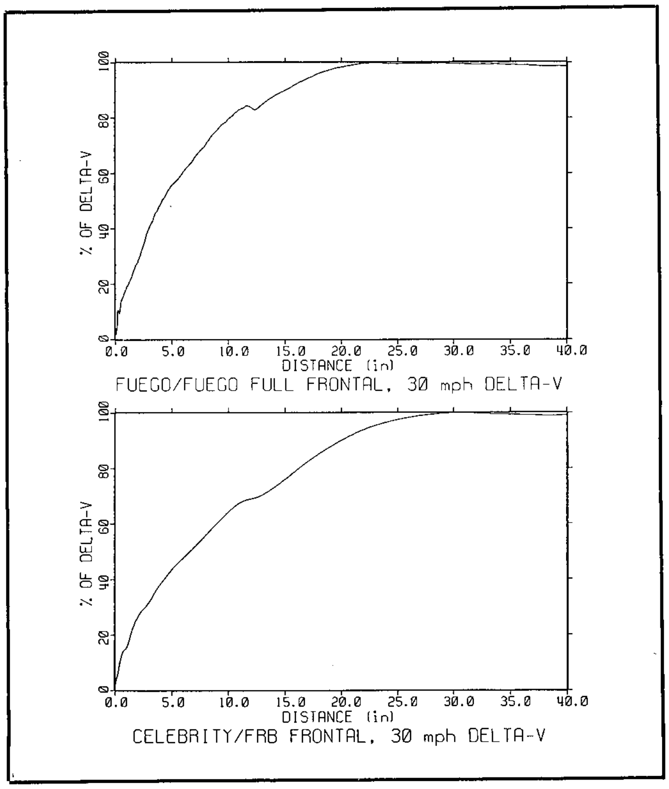


Figure 5. Percent total Delta-V versus seated distance

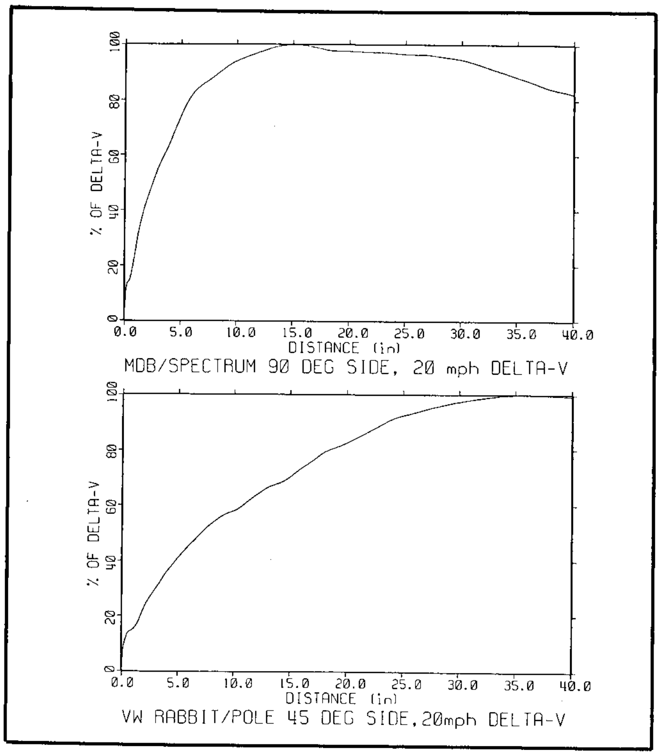


Figure 6. Percent total Delta-V versus seated distance

the Omni/Celebrity crash pulse of Figure 4, 77% of the vehicle delta-v is achieved at 14.5 inches of distance. Since the accident delta-v for A-pillar impacts was derived to be 24 mph, the head impact speed corresponding to this would be 77% of 24 or 18.5 mph. The derived head impact speeds are contained in Table 6.

Test velocities based upon the above approach would be approximately 18-23 mph for the front rail, 19 mph for the A-pillar and 10-17 mph for the side rail. These values could easily be rounded off to 20 mph for the front rail and A-pillar and 15 mph for the side rail. There appears to be enough data to substantiate that the head impact velocity for a given accident configuration is less than the total accident delta-v.

Table 5. Driver head to interior structure distances (in).

Test No.	Vehicle	A-Pillar	Front Header	Side Header
840113	'81 Granada		10.0	6.4
840209	'81 Horizon	14	8.6	5.9
831219	'81 Citation		11.0	6.5
840307	'81 Rabbit		12.9	6.5
840302	'82 Sentra		8.0	5.1
840216	'81 Omni		9.6	6.2
840316	'82 Civic		9.5	5.0
840223	'80 Concord		11.9	4.9
850710	'83 Mazda 626		10.6	7.6
851202	'85 Spectrum		13.9	6.5
851213	'84 Celebrity		14.1	6.3
------	'84 Accord	15		
	Mean	14.5	10.9	6.1
	Standard Dev.		2.1	.8

Table 6. Head impact velocity estimates (mph).

Crash Test Config.	A-Pillar @ 14.5"	Front Rail @ 10.9"	Side Rail @ 6.1"
Omni/Celebrity Off. Frontal	18.5		
Omni/Pole Front, Centered		18.2	
Fuego/Fuego Full Frontal		23.2	
Celebrity/FRB Full Frontal		19.0	
MDB/Spectrum 90° Side			17.3
Rabbit/Pole 45° Side			9.7

In summary, the NCSS and NASS data files were searched, and three upper interior areas were found to contribute to the bulk of serious head/face injuries. The three areas were front header, A-pillar and side header. Injuries were noted to occur from force directions both perpendicular and oblique to these surfaces. Median delta-v values for serious injuries were found to be 28 mph, 24 mph and 21 mph respectively for the 3 contact areas. It was shown that perpendicular impacts with a surrogate at 20 mph, 20 mph and 15 mph would be reasonable based upon the laboratory data analyzed.

A review of various types of hardware which might be used will be given in the next section.

Upper Interior Test Strategies

Test Alternatives

There are four levels at which head/pillar impact tests can be conducted: 1) full crash tests, 2) sled tests, 3) component tests, and 4) sub-component tests. As shown in Table 7, each test varies according to the degree of detail with which the occupant, vehicle, and head impact conditions are represented.

In general, there is a tradeoff between increased control and reduced realism. Likewise, reduced test cost can be gained at the price of diminished realism. While full vehicle crash tests provide maximum realism, the small target area of A-pillars and roof rails make control of the head impact location extremely difficult and repeatability virtually impossible. Component tests, on the other hand, offer complete control of the impact point, but sacrifice the interaction of the head and the dummy body. Sub-component tests are the least expensive but account for neither head-neck interaction nor A-pillar/frame compliance.

Earlier NHTSA research(1,2) has shown the component test to be one of the more promising approaches for studying head/pillar impacts. The component test is inexpensive, easily repeated, and provides a reason-

Table 7. Head/pillar test alternatives.

Test Level	Occupant Surrogate	Vehicle	Head Impact Conditions
Crash Test	Full ATD	Full - Moving	Function of Crash Pulse
Sled Test	Full ATD	Full - Moving	Specified Head Speed
Component	ATD Component	Full - Fixed	Specified Head Speed
Sub-Component	ATD Component	Component - Fixed	Specified Head Speed

able indication of full dummy head responses. The next section will discuss the validation criterion for component tests as a class, and will describe several current approaches for developing a head/pillar impact device.

Component Test Design Requirements

The objective in designing a component test is to replicate the head responses of the full dummy by using or representing a single component of the dummy, such as the dummy head. In addition, the correlation between component test and the full dummy test must hold across the range of current and potential passenger vehicles. In particular, this range must include vehicles with padded as well as unpadded A-pillars and roof rails.

For our study, the merits of a proposed component test will be judged by how well it compares with the responses of a Hybrid III dummy subjected to similar impact loading in a HYGE sled test. Our measures of head response will be the Head Injury Criterion (HIC) and the peak head resultant acceleration.

It should be noted that the headform is not required to look like a human head. Our criterion requires only that the impactor replicate the head responses of the full dummy. The objective is to simulate the response of the full dummy—not the appearance of the dummy.

Component Test Alternatives

Both NHTSA and the automotive industry have experimented with several variations on the head component test. Early NHTSA research performed head/rail testing using both a guided rigid 6.5" aluminum hemisphere and a guided P572 headform (1). The rigid hemisphere was used to measure pillar and rail stiffness, while the P572 headform was used to measure head responses.

General Motors has reported the use of a similar technique(3). The GM head impactor is a guided device which uses a hemispherical impact surface. GM has experimented with both an uncovered (or hard) impact surface and a "skin" covered surface.

An alternative to the guided impactor is the free motion headform. The free motion headform is especially constructed to allow the device to be fired in free flight at an A-pillar or roof rail. The free motion aspect of the device allows angled A-pillar impacts, glancing blows, and the measurement of rotational head accelerations—significant advantages over guided headforms.

Searle(4) has developed a free-flight headform to be used in a more realistic evaluation of vehicle interior components. The headform consisted of a smooth, rigid aluminum sphere of 165 mm diameter and having a mass of 5.5 kg. The procedure proved to be repeatable and demonstrated the ability to discriminate among different vehicle components. However, problems were encountered in velocity measurement, and the device had inadequate biofidelity for accident reconstructions.

NHTSA(5) has reported on the development and validation of a free-motion headform (FMH) based on the Hybrid III headform. The FMH launching mechanism was found capable of achieving repeatable head impact velocities. In addition, the headform was found capable of demonstrating a significant difference between vehicle components. The majority of NHTSA head/pillar tests to date have been conducted using the FMH.

Viability of a Free Motion Headform Test

In the previous section, it was shown that several types of hardware are currently being used to represent human head impacts. In this section, one type of hardware, the FMH will be examined more closely to determine the viability of such testing. The headform consists of a Hybrid III dummy head, modified to be compatible with a launching apparatus. For this study, it was assumed that the particular launching apparatus used to propell the FMH is not important. The analysis of this test approach does not mean that NHTSA has selected or will select this for head safety testing. In fact, research is currently being conducted to determine the viability of a simpler hardware approach, a guided hemi-spherical headform.

The viability of using a FMH type of test will be examined on the basis of biofidelity. The biofidelity was determined from drop test results and from comparisons of sled testing with the Hybrid III dummy. The sled testing and FMH testing were also modelled and the model was then used to predict the comparison in conditions other than those tested.

Calibration Response of the FMH

One of the advantages of the FMH approach is the potential to use the head drop test, which was devised for dummy head calibration, to calibrate the FMH. The FMH of this study was basically a Hybrid III head, the only alteration being the removal of the skull cap on the back of the head and substitution of a flat steel plate. The overall weight of the head was about the same (10.6 lb).

The calibration drop test for the Hybrid III head consists of suspending the head at a height of 14.8 inches over a specified steel surface. The alignment of the head is also specified and results in an impact to the forehead region. The resultant acceleration is to be maintained between 225-275 g's for acceptance. A sample result from dropping the FMH under these same conditions is shown in Figure 7. A peak response of 246 g's was obtained which is well within the acceptance range.

Comparison Testing of Hybrid III Dummy and FMH

Sled Description. A series of tests were conducted in which direct comparisons were made between the responses of a Hybrid III dummy and the FMH. The purpose of the testing was to create a realistic and controlled environment in which the only difference was the surrogate. In sled tests the surrogate was the Hybrid III dummy and in FMH tests, the surrogate was a Hybrid III headform, modified for launching from a propulsion device. It was hoped that this testing would allow quantification of the influence of the rest of the body on head impact response.

It was decided that a vehicle sled buck would not be used due to the difficulty of maintaining repeatability and the uncertainty of degradation of the pillars and rails with each impact. A sled buck was fabricated for the Hyge sled which was generic in representation of an occupant compartment of a vehicle. Figure 8 contains a photograph of the generic sled buck arrangement. The upper interior contact surface consisted of a length of steel tubing, fixtured at variable positions. The tubing orientation was varied to represent various A-pillar angles. A separate fixture with a horizontal bar was used to represent a roof rail. The seat of the sled was rigid and covered with aluminum plate. The seat pan was horizontal and the seat back angle was adjustable. The seat back was used to level the Hybrid III dummy head in the seated position.

The steel tubing used to represent the upper-interior contact surfaces was selected on the basis of stiffness in dynamic impacts. It was noted in an earlier program(2) to be roughly the same stiffness as a Chevrolet Citation A-pillar. The A-pillar tubing was 1 inch square by 18.75 inches long by .11 inch wall thickness. The mounting was such that the ends were simply supported, i.e., not fixed or pinned. The impact point on the tubing was roughly 6 inches from the top end. The roof rail was represented with the same tubing, of 24.75 inch length. The roof rail tubing was oriented horizontally and impacted directly in the center.

Padded tests were also performed on the sled buck. The padding (Dytherm 4.2) was the same as that used in the earlier study of A-pillars and was found effective in reducing head response into a Chevrolet Citation pillar. The same padding was used for the roof rail simulation tests, even though the roof rail is much less stiff than the A-pillar. Consequently, the response reduction for the padded roof rail tests was not as much was possible.

For the A-pillar simulations, the location of impact on the head was determined by the angle of the pillar. The head motion was horizontal and the pillar contacted the dummy forehead near the calibration area of the head. For the simulated roof rail impacts, the tubing was oriented horizontally and impacted the dummy just above the eyebrow area.

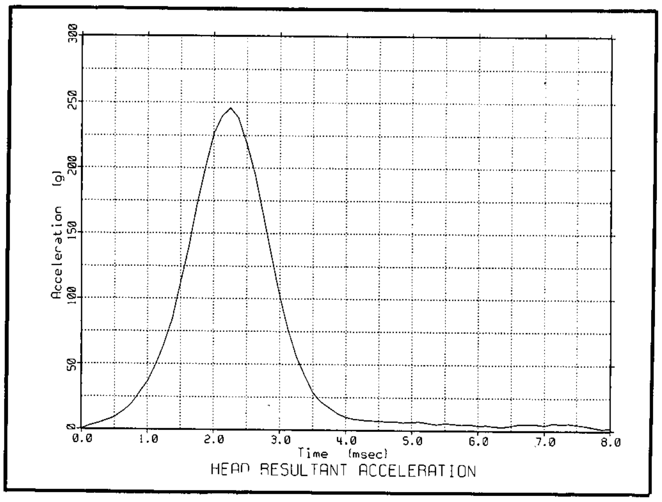


Figure 7. FMH resonse in Hybrid III drop test

FMH Description. Figure 9 shows the hardware used for the FMH testing of the generic environment. The fixturing used to hold the steel tubing for simulation of pillars and roof rails, is identical to that used on the Hyge sled buck. The tubing and orientation of the head are also identical.

Test Measurements. The Hybrid III head contained 9 accelerometers. Three were located at the center of gravity of the head and 3 additional pairs were located at fixed distances from the c.g. along the 3 principal orthogonal axis. In addition, the forces and moments at the head/neck juncture were measured for the sled tests with the whole dummy. The accelerometer data from the head responses was processed to yield both the translational and rotational motions.

Conditions Simulated. The baseline configurations consisted of two A-pillar angles and one roof rail configuration. The A-pillar angles were 45 and 60 degrees from horizontal. Each pillar and rail configuration was tested at 20 and 25 mph nominal impact

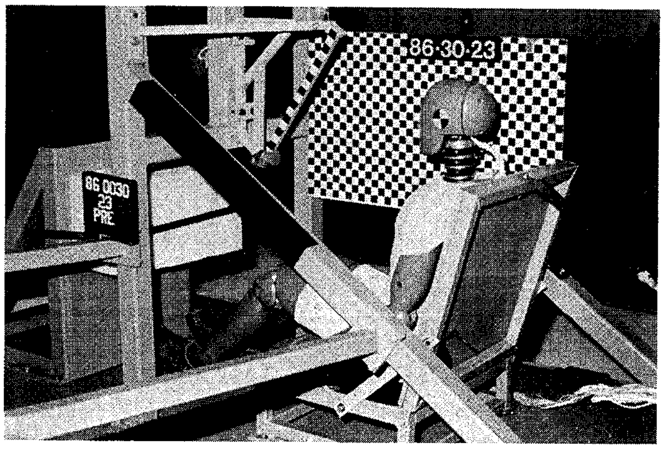


Figure 8. HYGE sled buck with simulated A-pillar

Figure 9. FMH test apparatus with simulated A-pillar

speeds. The total number of baseline tests was then 6. In addition, each test was repeated with a 1" thick piece of padding material added to the impact site. The total tests was then 12 sled and 12 FMH, not counting a small number of repeats of some test conditions.

Comparison-Test Results. A summary of the test conditions and the results of the 24 tests (12 Hybrid II sled and 12 FMH) are contained in Table 8. The sled responses are listed on the left half of the table, and the FMH responses on the right. In comparing the acceleration responses, the FMH results were noted to agree well with the full dummy results. The differences ranged from 0% to 8% with the average being about 5%. The acceleration resonse-time histories of the sled and FMH were plotted as overlay presentations and are contained in Appendix B.

A graphical summary of the comparison of the sled and FMH results is shown in Figure 10. Again, it was noted that the FMH responses followed the same trends and were very close in value to the whole dummy head responses at both speeds, and across all baseline and padded conditions. This would lead to the conclusion that the FMH is a realistic representation of a Hybrid III dummy head impact under similar conditions.

The HIC responses are shown in Figure 11. It was noted that at 20 mph, both the acceleration and HIC responses of the two types of testing agreed reasonably well in the 6 baseline and padded conditions. The padded responses at 25 mph also agreed quite well. The HIC responses from the 60 degree A-pillar and roof rail testing at 25 mph, did not result in good agreement between whole dummy and FMH test results.

The angular accelerations were also computed from the translational accelerations of the head. The peak values are shown in Table 8. The time histories of the rotational accelerations are contained in Appendix B. These were also plotted as overlays for the whole dummy tests and the FMH tests to allow easy comparison. It was again noted that a high degree of similarity exists between the FMH results and the whole dummy results.

Math Modeling of Hybrid III and FMH

The eventual goal in developing a component test is to develop a tool for evaluating the safety performance of all passenger vehicles in the fleet. To meet this objective, it is not enough to show that a proposed component test performs well for one vehicle. The component test must perform across the spectrum of current and future production vehicles. For our study, this requirement translates into the requirement that there be reasonable correlation between the FMH and the Hybrid III dummy across the fleet.

Table 8. Summary of sled and FMH responses.

Generic HYGE Sled Tests

TRC Test #	Test Vel. (mph)	Test Condition	Head Res. (G)	HIC	Res. Rot. Acc. (rad/s*s)
701	20	45/Unpadded	212.3	1552	21017
702	20	45/Padded	147.8	836	11881
703	20	60/Unpadded	251.0	2169	21741
704	20	60/Padded	175.3	1318	12467
737	20	Header/Unpadded	258.9	1423	19397
738	20	Header/Padded	193.3	1237	14133
739	26	45/Unpadded	248.3	2281	24529
740	26	45/Padded	194.1	1606	16565
746	26	60/Unpadded	305.1	3480	27365
747	26	60/Padded	246.1	2751	19675
752	26	Header/Unpadded	362.8	1665	29074
753	26	Header/Padded	241.2	1671	16335

Free Motion Headform Tests

VRTC Test #	Test Vel. (mph)	Test Condition	Head Res. (G)	HIC	Res. Rot. Acc. (rad/s*s)
57	20	45/Unpadded	205.9	1431	20950
58	20	45/Padded	135.4	714	15044
62	20	60/Unpadded	269.8	2112	25226
65	20	60/Padded	166.7	1137	13226
71	20	Header/Unpadded	260.0	1640	16108
70	20	Header/Padded	183.7	1352	11330
77	26	45/Unpadded	227.0	2013	24569
76	26	45/Padded	179.7	1478	9720
79	26	60/Unpadded	297.0	2769	26093
80	26	60/Padded	241.1	2588	21778
89	26	Header/Unpadded	383.1	2317	22284
86	26	Header/Padded	242.9	1755	14155

of a rigid hemisphere into a 18.75" length of 1" square tubing used in the experimental phase of the program. The force-deflection properties of the roof rail were computed from the acceleration-time response of a 15 mph normal impact of a rigid hemisphere into a 24.75" length of 1" square tubing. The sled model also contained a vertical "instrument panel" which was given the force-deflection properties of a Chevrolet Citation passenger dashpanel.

Modeling Results

The strategy of the analytical modeling phase of the program was to extend the results of the laboratory phase of the project to design conditions beyond those tested experimentally. In particular, the performance of the FMH was evaluated analytically across the fleetwide range of expected pillar/rail pitches and pillar/rail stiffnesses. Looking beyond current designs, each A-pillar was examined both in its baseline (unpadded) state, and with a 1" covering of Dytherm 4.2 padding. Finally, the performance of the FMH was investigated at the reference speed of 20 mph, and at the higher severity speed of 25 mph.

Table 9. Analytical modeling of head to pillar/rail impacts.

				... Sled ...		... FMH ...		... % dif ...	
Angle	Speed	K	Pad	HIC	G	HIC	G	HIC	G
35	25	2020	0	1620	171	1400	185	13.6	−8.2
45	25	2020	0	2492	214	2332	227	6.4	−6.1
60	25	2020	0	3689	260	3902	297	−5.8	−14.2
75	25	2020	0	4597	286	5207	312	−13.3	−9.3
90	25	2020	0	4828	286	5562	321	−15.2	−12.2
35	20	2020	0	906	132	800	148	11.7	−12.0
45	20	2020	0	1406	167	1333	182	5.2	−8.7
60	20	2020	0	2082	205	2234	223	−7.3	−8.8
75	20	2020	0	2616	228	2978	250	−13.8	−9.6
90	20	2020	0	2755	230	3180	257	−15.4	−11.8
35	25	2020	1	1012	137	845	134	16.5	2.1
45	25	2020	1	1616	176	1406	163	13.0	7.3
60	25	2020	1	2464	217	2292	199	7.0	8.5
75	25	2020	1	3108	241	3259	244	−4.9	−1.1
90	25	2020	1	3312	243	3549	256	−7.2	−5.2
35	20	2020	1	516	90	482	107	6.6	−19.1
45	20	2020	1	805	117	805	131	0.0	−12.1
60	20	2020	1	1227	149	1350	161	−10.0	−8.1
75	20	2020	1	1548	168	1743	171	−12.6	−1.8
90	20	2020	1	1654	171	1845	174	−11.5	−1.4
45	25	1000	0	1437	151	1376	160	4.2	−6.0
45	25	1500	0	1971	185	1865	196	5.4	−6.0
45	25	2500	0	2951	239	2737	253	7.3	−5.7
45	20	1000	0	814	117	786	128	3.4	−9.0
45	20	1500	0	1112	144	1066	157	4.1	−8.5
45	20	2500	0	1666	187	1565	202	6.1	−8.0
90	25	1000	0	2794	198	3290	226	−17.8	−14.1
90	25	1500	0	3826	245	4452	277	−16.4	−13.1
90	25	2500	0	5706	321	6520	358	−14.3	−11.5
90	20	1000	0	1599	160	1880	181	−17.6	−13.3
90	20	1500	0	2182	197	2545	222	−16.6	−12.4
90	20	2500	0	3256	257	3730	286	−14.6	−11.2

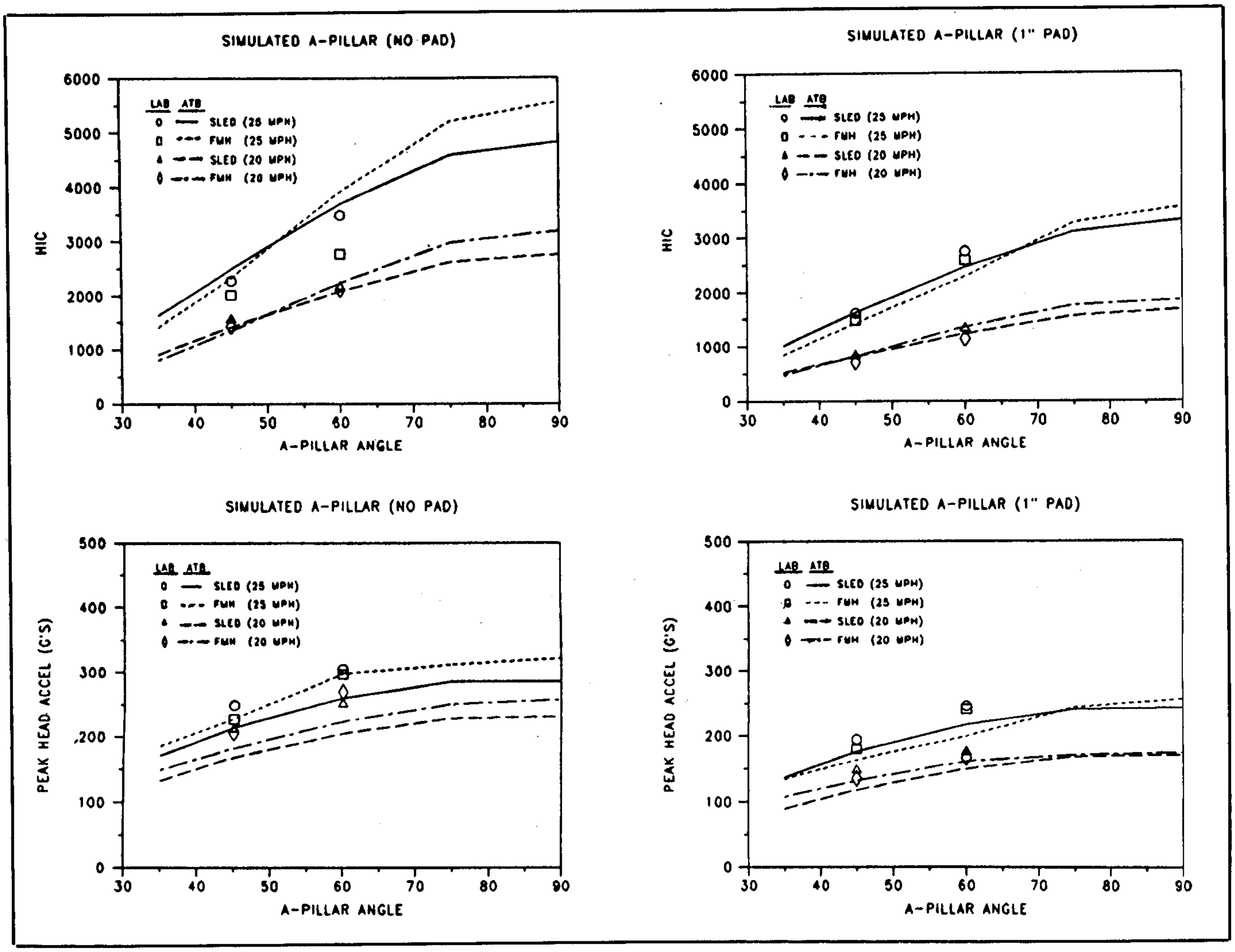


Figure 12.

tween the FMH and the Hybrid III dummy across the fleet.

Technical Approach

Our study examines this issue along two parallel approaches—experimentally and by analytical modeling. The strategy will be to compare the performance of the FMH and the full dummy across the full range of expected pillar/rail design parameters.

Essentially, the design of a pillar or rail can be defined by three parameters:

- Pillar geometry (pitch)
- Pillar stiffness
- Padding characteristics

Both the experimental and the analytical modeling phases of the program have examined the performance of the FMH at a number of these pillar/rail designs parameter combinations. To test the sensitivity of the FMH to fluctuations in impact speed, the performance of the device was also evaluated at both the accident data derived (or reference) speed of 20 mph and at a higher severity speed of 25 mph.

As discussed in the previous section, the experimental comparison of the FMH and the full dummy was conducted at 12 different design/severity configurations. The comparison was conducted at two A-pillar angles and for one roof rail configuration. The stiffnesses were near the average measured on several fleet vehicles. These three interior components were tested at the two speeds and in both padded and unpadded conditions.

The second prong of our program investigated the FMH/sled correlation by analytically modeling and comparing the head responses of the FMH and full Hybrid III dummy. The analytical modeling was conducted using the ATB (Articulated Total Body) computer package—a three dimensional model of the dynamics of a vehicle occupant subjected to an automobile collision(6).

The Analytical Model

For our study, the FMH was modeled as a one-dimensional mass with the inertial properties of the Hybrid III headform. The sled tests were modeled as a three mass system composed of the headform, the neck, and the torso. Articulation of the three mass system was provided by joints at the juncture of the head and neck and at the juncture of the neck and torso.

The A-pillar was modeled as an infinite plane having the required pitch. The force-deflection properties of the A-pillar were computed from the acceleration-time response of a 15 mph normal impact

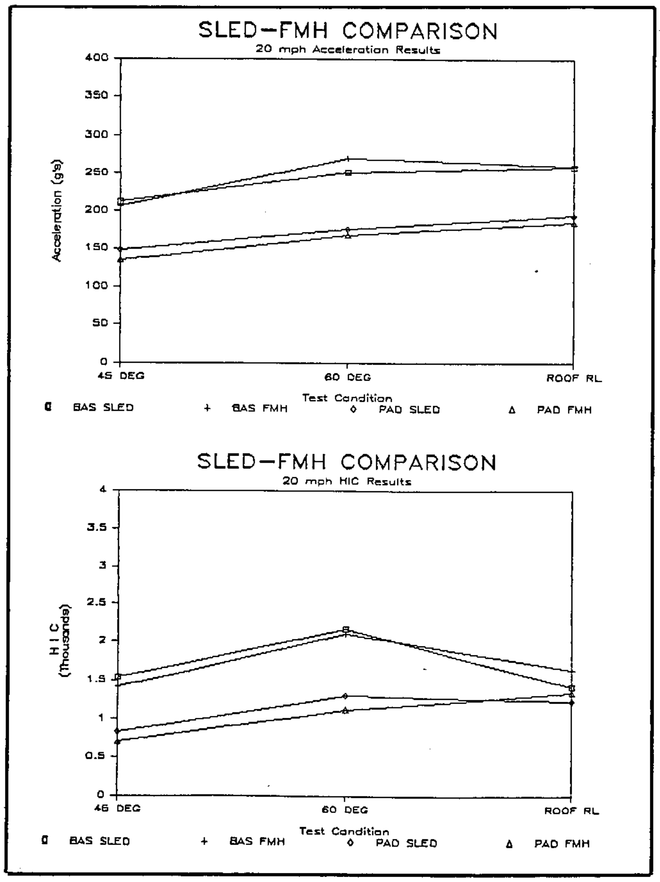


Figure 10. Summary of 20 mph test responses

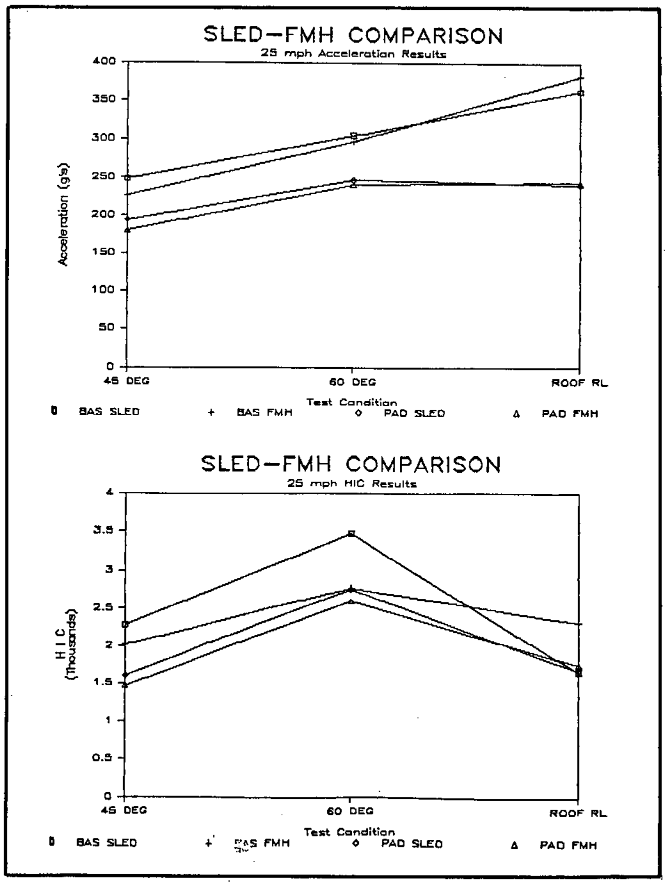


Figure 11. Summary of 25 mph test responses

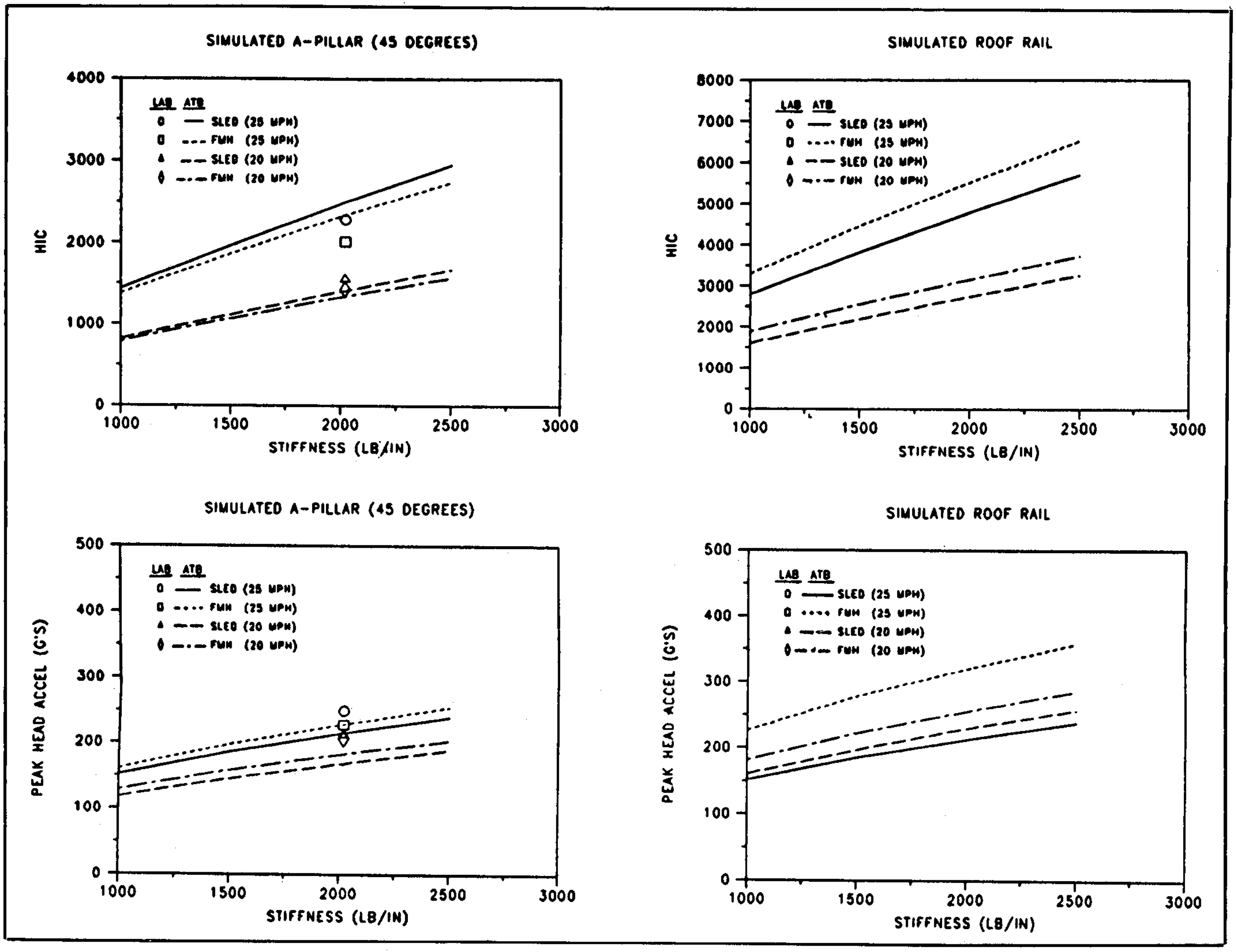


Figure 13.

The results of that evaluation are tabulated in Table 9, and are presented graphically in Figures 12 and 13. The results of applicable tests from the experimental program have been overlaid on the ATB results. The reader will note that our models are in fair agreement with the results of the experimental program.

Reinforcing the observation of the experimental program, the results are very encouraging. The head responses of the FMH are within 20% of the head responses of full dummy across the range of pillar/rail designs considered. In addition, the model also suggests the following:

- The FMH performs as well in padded environments as in baseline environments.
- The best agreement between FMH and full dummy is observed for A-pillars of the lower pitches (<45°) utilized in current production cars. Correlation is least good for roof rails (pitch of 90°).
- The FMH is a better indicator of the full dummy at its accident derived speed of 20 mph than at the higher severity speed of 25 mph.
- For the unpadded 45° A-pillar, the agreement between FMH and full dummy is relatively independent of the stiffness of the pillar.

Conclusions

This paper has reported on the development and validation of the Free Motion Headform (FMH)—a component test device for evaluating head-to-pillar/rail impacts in passenger vehicles. Our conclusions are:

- Examination of the accident statistics and laboratory crash pulses suggests a head impact test speed for A-pillars and front roof header of 20 mph, and a head impact test speed for side roof rails of 15 mph.
- The results of 24 tests (12 Hybrid III sled and 12 FMH) showed good agreement between the head responses of the FMH and

the full dummy when subjected to similar impact conditions.

- The results of analytically modeling the FMH reinforce the results of the experimental evaluation of the device. The head responses of the FMH were within 20% of the full dummy across the spectrum of expected pillar/rail designs.

Examining specific pillar/rail designs, both the experimental and analytical evaluations of the FMH lead to the following conclusions:

- The FMH performs as well in padded environments as in baseline (unpadded) environments.
- The best agreement between the FMH and full dummy is observed for A-pillars of lower pitch (<45 degrees) utilized in current production cars. The experimental program also showed good agreement for roof rails.
- For the unpadded 45 degree A-pillar, the agreement between FMH and full dummy is relatively independent of the stiffness of the pillar.

The experimental and analytical phases of NHTSA's pillar/rail research program have examined a broad spectrum of pillar/rail design parameters, and, in both cases, have come to the same conclusion. In impacts with a simulated A-pillar, the FMH provides good to excellent indication of the head responses of a full Hybrid III dummy subject.

Acknowledgments

The authors gratefully acknowledge the efforts of Randa Radwan and Claude Melton in modeling and processing test data, and Susan Weiser in preparing the manuscript.

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Appendix A

Programs Used to Extract NASS Data
SAS Program for NCSS Database

```
DATA VEH;
SET VEHICLE0;
IF ((VGADPR='F' OR VGADPR='R' OR
   VGADPR='L') AND (01<=VBDYSTY<=04)
   AND (VAPPVEH));
KEEP CASENO WEIGHTFA VEHNO VAPPVEH
   VMODEL VBDYSTY VVEHWT VCONTPR
   VDOFPR VGADPR VSVAPR VTDDPR VEX-
   TEP;
DATA SEVACC;
MERGE ACCIDENT SEVERITY;
BY CASENO;
IF DVTDAM1 NOT=.;
KEEP CASENO WEIGHTFA VEHNO NUMVEHIN
   TYPEIMPA DVTTRA1 DVTDAM1;
DATA OCC;
SET OCCUPNT0;
IF ((SEATAREA=1 AND (LOCATION=1 OR
   LOCATION=3) AND (RESTRINV=0 OR
   RESTRINV=8) AND ((BODYREG1-'H' OR
   BODYREG1='F') OR (BODYREG2-'H' OR
   BODYREG2='F') OR (BODYREG3='H' OR
   BODYREG3='F')) AND ((CONTACT1=18)
   OR (CONTACT1=11) OR (CONTACT1=19)
   OR (32<=CONTACT1<=35) OR
   (CONTACT2=18) OR (CONTACT2=11) OR
   (CONTACT2=19) OR
   (32<=CONTACT2<=35) OR
   (CONTACT3=18) OR (CONTACT3=11) OR
   (CONTACT 3=19) OR
   (32<=CONTACT3<=35)) AND
   ((3<=AIS1<=6) OR (3<=AIS2<=6) OR
   (3<=AIS3<=6)));
SWITCH1=1;
KEEP CASENO WEIGHTFA VEHNO SEATAREA
   LOCATION RESTRINV BODYREG1
   BODYREG2 BODYREG3 CONTACT1
   CONTACT2 CONTACT3 AIS1 AIS2 AIS3
   OVERALLA SWITCH1;
DATA VEHSACC;
MERGE VEH SEVACC;
BY CASENO VEHNO;
SWITCH2=1;
KEEP CASENO WEIGHTFA VEHNO VAPPVEH
   VMODEL VBDYSTY VVEHWT VCONTPR
   VDOFPR VGADPR VSHLPR VSVAPR
```

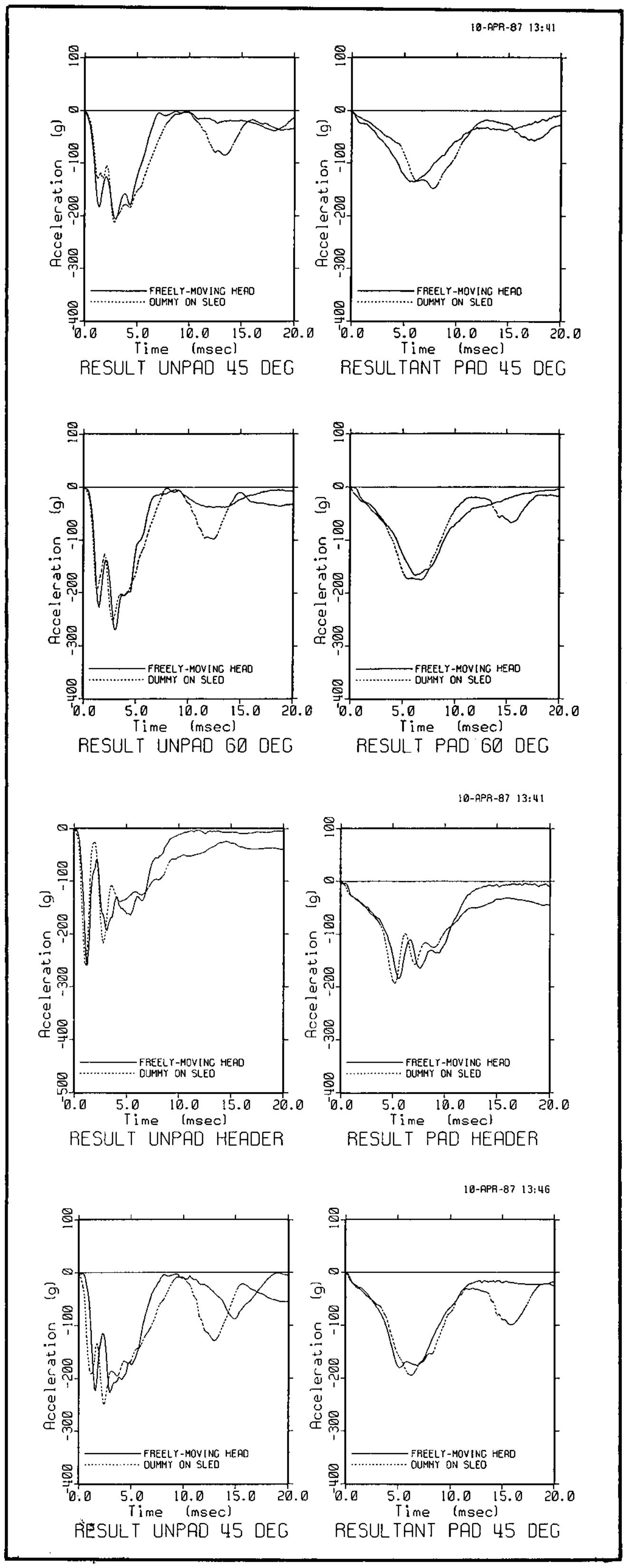
```
    VTDDPR VESTEP NUMVEHIN TYPEIMPA
    DVTTRA1 DVTDAM1 SWITCH2;
DATA VEHOCC;
MERGE VEHSACC OCC;
BY CASENO VEHNO
IF SWITCH1 AND SWITCH2;
DROP SWITCH1 SWITCH2
PROC PRINT DATA=VEHOCC;
TITLE1 HEAD AND FACE 3<=AIS;
```

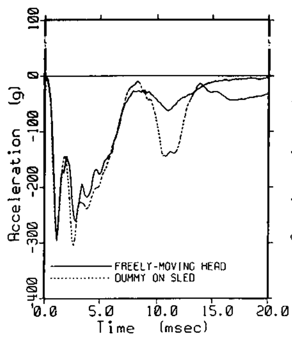
SAS Program for NASS Database

```
DATA OCC;
SET SASNAS.OCCUPANT;
IF ((01<=SEAT__POS<=03) AND
    ((OBODYRG1='H' OR OBODYRG1='F') OR
    (OBODYRG2='H' OR OBODYRG2='F') OR
    (OBODYRG3='H' OR OBODYRG3='F'))
    AND ((OCONTCT1=06 OR OCONTCT1=13
    OR OCONTCT1=14 OR OCONTCT1=31 OR
    (33<=OCONTCT1<=34)) OR
    (OCONTCT2=06 OR OCONTCT2=13 OR
    OCONTCT2=14 OR OCONTCT2=31 OR
    (33<=OCONTCT2<=34)) OR
    (OCONTCT3=06 OR OCONTCT3=13 OR
    OCONTCT3=14 OR OCONTCT3=31 OR
    (33<=OCONTCT3<=34))) AND
    ((03<=OAIS1<=06) OR (03<=OAIS2<=06)
    OR (03<=OAIS3<=06)));
SWITCH1=1;
KEEP CASE__ID CASE__NO NATWT PSUWT
    VEH__NO OCC__NO SEAT__POS EJECTION
    NMAN__AVAI MAN__REST AUT__AVAI
    AUT__REST PSU OAIS1 OAIS2 OAIS3
    OBODYRG1 OBODYRG2 OBODYRG3
    OCONTCT1 OCONTCT2 OCONTCT3
    OTREATMT OCC__ROLE SWITCH1;
DATA VEH;
SET SASNAS VEHICLE;
IF (01<=BODY__TYP<=06 OR
    10<=BODY__TYP<=13 OR
    50<=BODY__TYP<=58) AND
    (01<=DV__SOURC<=02);
SWITCH 2=1;
KEEP CASE__ID CASE__NO NATWT PSUWGT
    VEH__NO VEH__ROLE BODY__TYP
    CLOCK__PR CURB__WT DEFLOCPR DFOR-
    CEPR DISTRIPR DV__SOURC DV__TOTAL
    EXTENPR IMP__TYPE PSU LONGITPR
    MAKE OBJ__CNPR VERTICPR SWITCH2;
DATA VEHOCC;
MERGE OCC VEH;
BY PSU CASE__ID VEH__NO;
IF SWITCH1 AND SWITCH2;
DROP SWITCH1 SWITCH2;
PROC PRINT;
TITLE1 HEAD AND FACE 3<=AIS;
```

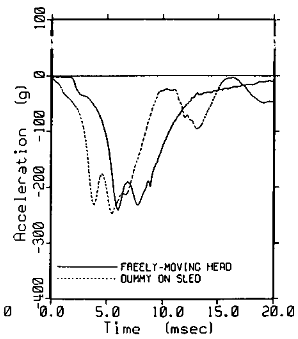
Appendix B

Response Summary of HYGE Sled and FMH Test Results: Overlay of Acceleration Time Histories

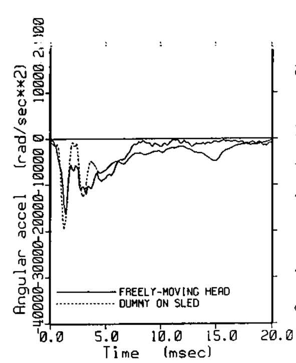




RESULT UNPAD 60 DEG



RESULT PAD 60 DEG



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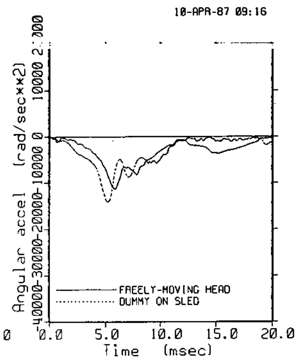
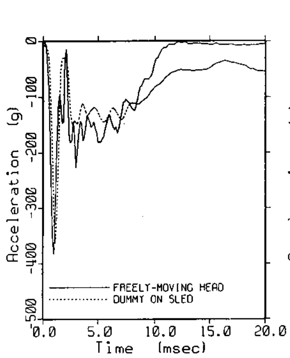
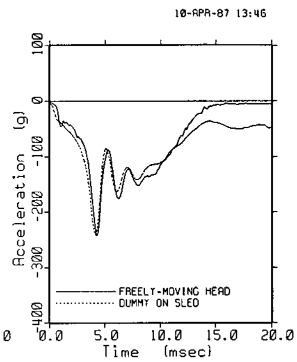


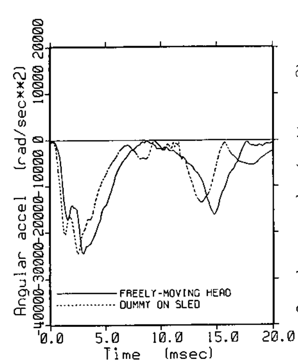
RESULT PAD HEADER



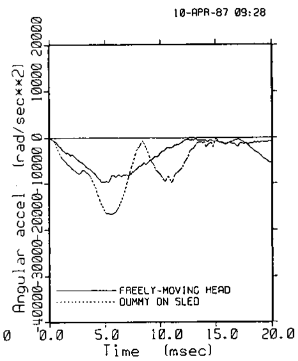
RESULT UNPAD HEADER



RESULT PAD HEADER



RESULT UNPAD 45 DEG



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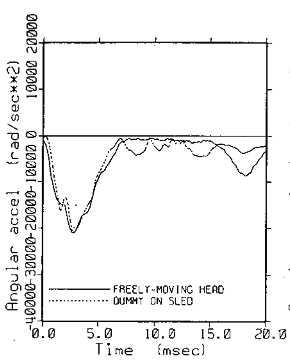

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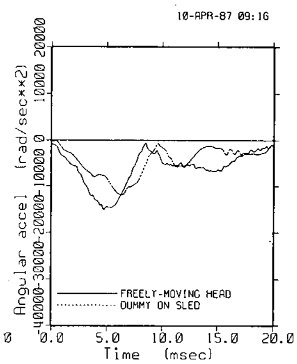
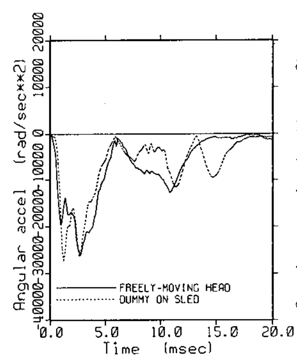
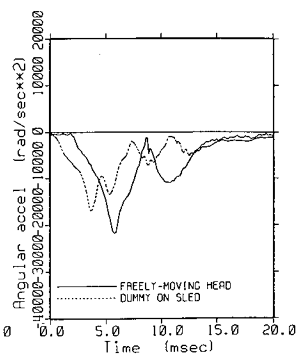


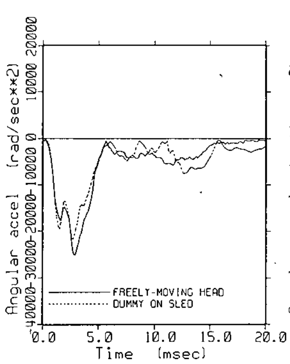
RESULT PAD 45 DEG



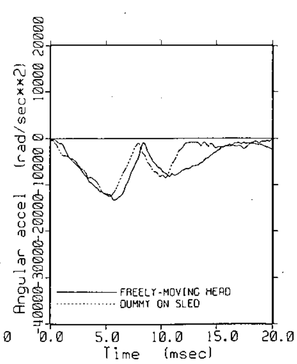
RESULT UNPAD 60 DEG



RESULT PAD 60 DEG



RESULT UNPAD 60 DEG



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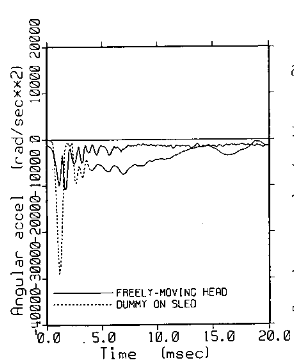

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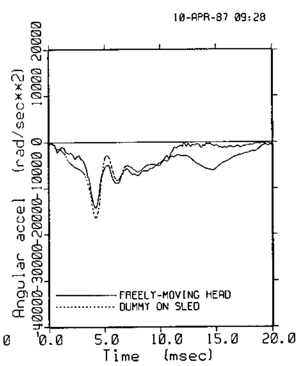

RESULT PAD HEADER

An Investigation Into Vehicle Side-Door Opening Under Accident Conditions

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Abstract

Accident studies in the UK have identified a number of instances where the door latch has been activated following a side impact, allowing the door to open and the occupant to be ejected. The latch activation occurs in cars having a pull rod linkage connecting the latch and interior release handle, through disturbance of the linkage following an impact by the occupant's shoulder or upper arm on the door inner panel.

An investigation into the phenomenon was conducted by MIRA and tests carried out which successfully reproduced this effect in the laboratory. It is proposed that this would be useful in predicting the behaviour of a vehicle door system as a development or legislative tool.

The test comprises an impact on the door's inside surface at 24.1 km/h (15 mile/h) by a resilient bodyform of 36 kg (80 lb) mass, reproducing the shape of the upper torso and shoulder.

In addition, a number of design changes to the door structure and linkage arrangement have been proposed to alleviate the problem.

Introduction

It has long been recognised that preventing the ejection of an occupant in an accident is crucial in minimising his resulting injuries. Accident studies have consistently refuted the popularly-held view that it is beneficial to be "thrown clear" and shown that the risk of injury is greatly increased where ejection occurs.

Although seat-belts are to some extent effective in preventing ejection, it is the vehicle's side door which remains the principal defence against its occurence. If the effectiveness of the door is destroyed by either mechanical failure or inadvertent operation of the latch, a restrained occupant may suffer partial ejection. Similarly, an unrestrained occupant, such as a rear-seat occupant in a four-door car, may suffer complete ejection.

The importance of maintaining an effective barrier between the occupant and the exterior is reflected in the adoption of standards such as FMVSS 206 and ECE 11 as a mandatory requirement in most industrialized western nations. These standards stipulate minimum strength requirements for the latches and hinges, and also require the latch not to release when subject to acceleration. Compliance with the ECE 11 standard has been compulsory for all new cars in the UK for several years, and accident studies have demonstrated its effectiveness in reducing occupant ejection through door bursting(1).

However, a study of UK accidents for the period 1978 to 1980 has identified several instances where occupants have been ejected in accidents following the opening of the car door. In every case, the door appeared to open not as a result of mechanical failure of the door-hinge-latch system but through inadvertent activation of the latch. It is believed that this occurred as a consequence of an impact by the occupant against the inside face of the door disturbing the linkage between the latch and the interior release handle.

Concern over these accidents led to the UK Department of Transport commissioning MIRA to investigate the phenomenon by attempting to reproduce the effect under laboratory conditions. It is this study which forms the basis of this paper.

Accident Data

The Risks Associated with Occupant Ejection

Investigation into cases of accidents involving occupant ejection provide a valuable source of information into the mechanisms of this type of accident.

The incidence of these cases amongst accidents as a whole underwent a significant change as a result of the enforcement of mandatory standards for hinge and latch performance during the 1950's and 1960's. This served to reduce the number of occupant ejections resulting from the mechanical failure of these components(1).

Despite this reduction, ejection remains a very common feature in accidents involving death or serious injury. This is well illustrated in the paper by Carlsson(2) which studied cases of ejection amongst a sample of 10,000 accidents involving Volvo cars. Considered as a proportion of all of these accidents, ejection occurred in only 0.5% of cases. Yet amongst the accidents involving occupant fatality, ejection occurred in 12%. A similar study of the NCSS records(3) put this figure even higher, at 27%. Clearly, despite its low incidence this phenomenon is frequently associated with this most severe category of accidents.

This does not imply, however, that ejection was responsible for the extent of injury in all of these cases. To assess the role of occupant ejection in this respect it is necessary to distinguish between those cases where, on the one hand, ejection occurred as a consequence of the severity of the accident, and on the other where the accident injury was made more severe as a consequence of the ejection. In other words, we need to assess the separate roles of the ejection and the remaining accident conditions as a contribution to the resulting injuries.

It is, however, difficult to measure this directly, but several studies have attempted to assess the role of ejection in causing injuries. O'Day and Scott(4) conducted a "back-to-back" comparison of injuries to selected pairs of occupants, where one of the pair was ejected from the vehicle and the other remained inside. This reported a 22% rate of fatality amongst those ejected, compared with a 9% rate of fatality amongst those not ejected. At the other end of the severity scale, the proportion which received no hospital treatment was 3% for those ejected but 21% for those not ejected. The paper by Carlsson(2) quotes the figures for the risk of severe or fatal injury as being 48% for ejected occupants compared with 8% for belted (presumably non-ejected) occupants.

On the basis of these figures, it is therefore likely that the level of injury is significantly increased by the ejection process. Although crashes, where ejection occurs, appear to be the most severe ones, this does not seem sufficient to explain the high level of injury amongst ejected occupants.

The Mechanism of Occupant Ejection

Before the introduction of mandatory standards for latch and hinge performance typified by FMVSS 206 and ECE 11, a large proportion of ejections were caused by latch failure. The engineering improvements brought about as a result of these regulations have served to reduce the number of cases of this type from the accident figures and give greater prominence to other ejection mechanisms, for example, ejection via the window apertures or through the disruption of the passenger compartment structure. This pattern of distribution has remained essentially unchanged until the present. However, in the UK despite the requirement for all new vehicles to be fitted with latches which meet the ECE 11 requirements for latch strength, hinge strength and the latches' resistance to opening under acceleration, cases of door opening continue to form a recognisable proportion of ejection accidents.

A recent study by Green et al of the Loughborough Institute for Consumer Ergonomics and the Leicester Royal Infirmary(5) illustrates this.

In this study of serious accidents involving ejection in the UK, of 919 vehicles in all, door opening occurred in 129 (12.7%) of the vehicles. This was divided between 79 (7.8%) side door opening and 66 (6.6%) tailgate opening. On 16 cars, both of these opened, and some had openings of more than one side door. These openings were associated with 12 occupant ejections or 12% of the total ejections registered in the study.

The principal reason for non-ejection seems to be that the occupant was not seated next to a door which opened. Amongst exposed occupants, that is, occupants who were unrestrained in a seat next to a door which opened, 30% of such front seat occupants were ejected and 55% of rear seat occupants.

Finally, this study investigated the mechanism of latch release. The greatest proportion of these occurrences, some 51% were considered to be due to activation of the linkage, compared with 28% due to latch failure, 5% due to handle activation and 16% due to other mechanisms.

To obtain an indication of the total number of ejection cases in the UK it is possible to assign the proportion of ejection cases in various detailed studies with comparable totals for the UK as a whole. Such an approach is, of course, subject to the many pitfalls of matching sample and population, and so such figures must necessarily be crude estimates only.

If we apply the 0.5% ejection rate amongst all injury accidents in the Volvo study to the 246,000 such accidents in the UK in 1985, we can estimate a total of 1230 occupant ejections. Alternatively, the figure of 12% given as the proportion of ejections amongst occupant fatalities, when compared to the UK total fatalities of 5165, gives 620 cases. A third approach applied to a wider range of car types is to consider the 40 cases of complete ejection and 16 cases of partial ejection in the Loughborough study. The study was stratified so that serious or fatal injuries represented 30% of the study, giving a total of 526 such cases in the study against 70980 for the UK. On this basis, therefore, we would expect to see 5400 complete ejections and 2160 partial ejections.

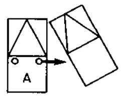
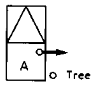
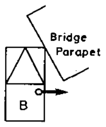
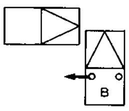
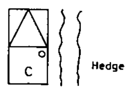
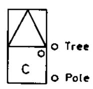
Allowing for the disparity in these figures, therefore, we could say that as a minimum figure, 620 ejections occur in the UK per year. If the proportion of cases due to various causes reflects the figures in the Loughborough study, we might expect to see, at least, 140 ejections via the side door. About half of these may be expected to be due to linkage failure, that is some 70 ejections.

The Injury Mechanism-Occupant Trajectory

The mechanism of injury, that is, the circumstances of the accident, the dynamics of the occupant, and their interaction with the door structure and the latch mechanism may be illustrated by reference to six representative accidents chosen from a selection of accident studies collected in the UK.

Table 1 summarizes and illustrates the relevant features of the accidents. The door activation investigation was limited to three specific vehicle models—A, B and C—and two situations have been selected for each of the vehicles concerned.

Table 1. Accidents featuring door latch release.

Accident No.	Vehicle	Collision Configuration	Occupants	Ejection	Severity AIS
1	A	 A	Driver F Passenger	Yes Possibly	2 to 3 1 to 3
2	A	 A, Tree	Driver	Yes	5 (Fatal)
3	B	 Bridge Parapet, B	Driver	Yes	3
4	B	 B	Driver F Passenger	NO Yes	- 2 to 3
5	C	 C, Hedge	Driver	No	1
6	C	 Tree, C, Pole	Driver	No	1

Accident Number 1 concerns a vehicle of type A which was struck on the front right corner from a 5 o'clock direction by a vehicle of similar size, with a severity estimated to be equivalent to an ETS of 30 mph. The unrestrained driver struck the inside of the door, denting the inner panel at the beltline. The driver's door latch released and the door opened. The driver was ejected via the open door and sustained injuries of MAIS 3 in the form of a comminuted fracture of the right ulna, and a fractured scapula. The front seat passenger was reported to have been ejected via the same route, but her injuries are not known.

Accident Number 2 again concerns a vehicle of type A whose driver lost control on a bend and struck a tree. The impact with the tree occurred on the right hand rear door at an angle of 4 o'clock, with a severity estimated to be equivalent to a 35 mph pole impact. The driver, the sole occupant, was unrestrained, he struck the door with his hip and chest, causing moderate distortion of the inner panel. The door released and the occupant was ejected, receiving a fatal injury in the form of contusions of the liver.

Accident Number 3 concerns a vehicle of type B which struck a bridge parapet with the right front corner in a 2 o'clock direction. The impact speed has not been estimated, but there was reported to be minor intrusion of the right footwell. The unrestrained driver struck the right hand door, which opened, allowing the driver to be ejected. The driver suffered a fractured rib.

Accident Number 4 concerns a vehicle of type B which was struck on its front left corner in a 9 o'clock direction by a vehicle of 10% greater mass. The collision speed was estimated to be equivalent to a 30 mph impact. The unrestrained female front seat passenger struck the left front door. The latch released and the door opened, leading to complete ejection and injuries to the head and left scapula of MAIS 3.

Accident Number 5 concerns a vehicle of type C which rolled over and struck a hedge on its right hand side. The impact with the hedge and low bank was distributed over most of the side of the car and was in a 3 o'clock direction. The unrestrained male driver struck the door inner panel with his shoulder, causing the door to unlatch. In this case, the presence of the hedge prevented the door from opening completely and the occupant was not ejected. The resulting injuries were slight and not caused by contact with the door. Despite the insubstantial nature of the hedge at door height, the vehicle suffered moderate intrusion at rocker panel level, presumably from the earth bank or the root system of the hedge. This accounts for the severity of the occupant door contact.

Accident Number 6 concerns a vehicle of type C. The car lost control and struck a tree just forward of the right A-pillar area. This was followed by a roll-over and a further impact with a pole on the rear right quarter panel. Both impacts were estimated to be in a 3 o'clock direction. During the accident the driver struck the right door waist rail with his shoulder and the door released and opened. The driver was restrained; this prevented complete ejection through the door and he suffered only minor injuries (AIS 1).

Although this is only a small sample, it nevertheless represents a wide range of accident conditions in terms of severity, accident type and resulting injuries. Despite the small sample size, however, there are a number of conclusions that we can draw from this study.

a) In each of the accidents, a similar pattern of damage to the door inner panel has been noted and is shown in Figure 1. This was in the form of a dent or crease at the belt-line. Since this is strongly linked with the impact by the occupants shoulder it appears that it occurs before, and not as a result of the door opening. Furthermore, since in most accidents there is very little damage to the door and B-pillar apart from this, it would appear to be the sole cause for the unlatching.

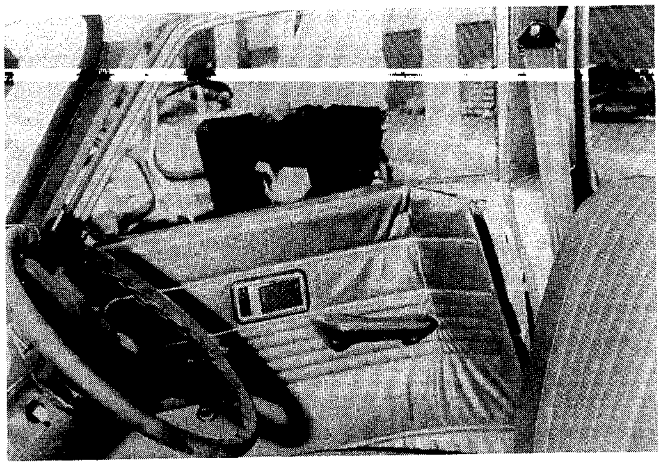


Figure 1. Road accident—door inner panel deformation

b) The recurrence of the same type of vehicle in different accidents suggests that the problem may be related to specific designs of door system.

c) The mechanism of door opening is clear from examination of the door layout. In each type of car involved in the accidents, the linkage between the latch and the interior release handle is in the form of a pull-rod routed along the inner surface of the door, at a height very close to the waist rail. It appears that the impact by the occupant's shoulder has caused the linkage to activate the latch.

d) The absence of damage to the latch, striker pin or surrounding structure suggests that the phenomenon occurs as a result of low energy impacts, less than the energy needed to deform the door structure significantly. This is confirmed by the low level of shoulder injury in those occupants who were not ejected as a result of the door opening. If this is so, it suggests that, firstly, the conditions for door bursting are the same as for ejection, namely an occupant projected outward. In addition to this, if the door bursts under a relatively low force, this implies that an occupant projected against it would experience only a small change in velocity and would exit the open door and strike some external object at a higher velocity than if the door offered more resistance.

Experimental Investigation

MIRA was commissioned by the UK Department of Transport to conduct an experimental investigation of this phenomenon. The terms of reference of this study were to attempt to reproduce this phenomenon in the laboratory, using doors from vehicles known to be prone to this type of occurrence. In addition to this, it was hoped that other features observed in the accident studies, particularly the pattern of door damage, would also be reproduced.

In initial discussions with the sponsors, it was felt that one of two types of test might offer a suitable method of achieving this. The first of these was a quasi-static test, in which an intrusion device, representing the shoulder of the occupants, was forced slowly into the inside face of the door until unlatching occurred. The second was in the form of an impact test using a bodyform representing the torso, shoulder and upper arm of an occupant, projected against the inner panel of the door. However, before the development work on these tests commenced, some additional work was done to gather data on the forces and deflections required to release the latch on undeformed door specimens.

Investigation into Latch Release Parameters

Front doors from two types of car known from the accident studies to be subject to this effect were selected and mounted in a test fixture. A small hole was made in the outer skin of the door opposite the centre of the latch release rod and a thin cable looped around the rod. The cable was then pulled outwards until latch release occurred, the force and deflection of the rod being noted at the time of release. The results of the tests are given in Table 2.

It can be seen from the table that the two specimens from Vehicle A unlatched at a similar force but with some variation in deflection, presumably due to manufacturing and assembly tolerances. The latch from Vehicle B required the same order of force to release but less deflection.

The deflections appear to be rather less than the degree of plastic deformation observed in photographs of the accident vehicles.

It was observed that on one of the latch systems that the locking linkage and latch linkage had some

Table 2. Results of latch rod deflection tests.

Sample	Door Release Force N	Door Release Deflection mm
B1	200	15
A1	178	30
A3	182	40

common components. A similar test was conducted on the interior lock linkage rod which was incorporated into the handle system to enable lock release from inside the car.

The result of the test on the lock linkage was interesting, in that it required a similar load and deflection to unlock the latch as it does to unlatch it. The design of latch effectively isolates the latch from the interior release when the lock is activated. However, the result suggests that even if the lock was activated, it might not prevent the door from unlatching, since the lock would be de-activated by the time the door was unlatched.

Development of Quasi-Static Test Method

A quasi-static test was devised similar to a side door intrusion test but from inside the car.

These tests were conducted by mounting a door specimen by its hinges on a fixture, with the latch in engagement with a striker pin. An intrusion device was constructed from 50 mm diameter steel tube to represent the humerus of the occupant. This was mounted on a hydraulic ram which forced it slowly into the inner panel of the door. A typical test arrangement is shown in Figure 2.

The initial test was conducted with the intrusion device axis inclined at 20° to the vertical in a longitudinal plane and aligned with the centre of the longest unsupported section of rod. Later tests used a similar tube mounted vertically, a striker pin mounting allowing it to pivot, a 100 mm diameter intrusion device and a device allowing the intrusion device to pivot in a transverse plane. A total of nine tests were conducted on front doors from vehicles of type A and B.

Despite these variations only one test, the first, where activation was not repeated, resulted in unlatching of the door. All of the others continued until the door failed mechanically or until deformation of the door rendered unlatching impossible. From observations during the tests, it was felt that the test method was not producing a similar pattern of deformation to that seen in the accident studies. In particular, the test was leading to rotation of the latch about a vertical axis which prevented the striker from clearing the latch.

Development of Dynamic Test Method

It is clear that the closest simulation of the conditions existing at the door on the instant of release, as seen through the accident case studies, would be achieved through some form of impact test. Such a test would need to reproduce the important parameters such as the geometry and mass of the impacting occupant, yet be simple enough to allow the test to be conducted at moderate expense.

It was felt that a suitable test method would be broadly similar to the Bodyform Impact Test specified to measure the effectiveness of the Steering Assembly in Regulations such as FMVSS 203 and ECE Reg 12. Although the bodyform used in this test is not suitable for representing lateral impacts, the use of a common propulsion device would clearly allow the test to be carried out more easily, and provide good repeatability.

As in the previous series of tests, a basic test method was established which was common to all tests, with variations introduced to attempt to achieve repeatable latch release.

The basic test utilised a bodyform constructed from the upper torso and arm of a Sierra fiftieth-percentile male anthropomorphic dummy. This was adapted to enable it to be mounted on a sled-type propulsion device used for ECE Reg 12 tests by fixing a frame to the rigid thoracic spine of the dummy. The whole unit was then ballasted to the ECE Reg 12 bodyblock mass of 36 kg. The arm was fixed at the elbow to constrain articulation and the centre of gravity position adjusted so that it coincided with that of the bodyblock specified by ECE Reg 12. This test arrangement is shown in Figure 3.

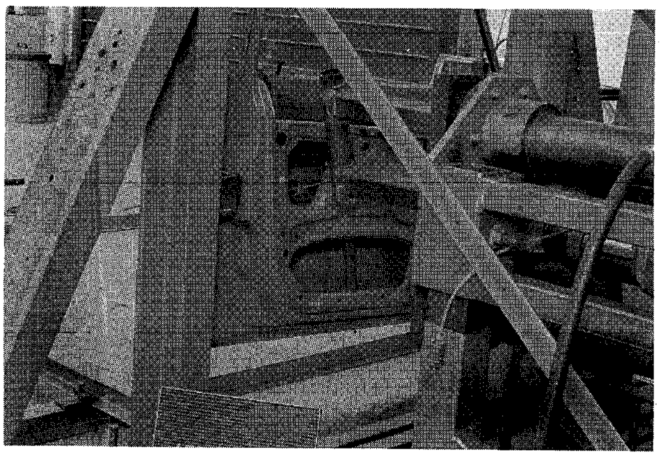


Figure 2. Static door latch activation test

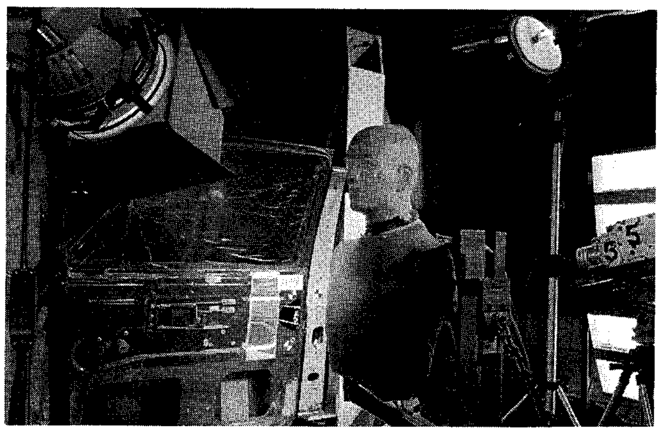


Figure 3. Dynamic door latch activation test

To allow information to be gathered which would be useful in the investigation, an accelerometer was mounted on the dummy spine, measuring in a lateral direction.

As in the ECE Reg 12 test, the bodyform was mounted on its propulsion sled, the sled was propelled to impact velocity and arrested a short distance from the door, projecting the dummy forwards to strike the door in free flight.

An impact velocity of 24.1 km/hr was chosen initially as it seemed that this represented the accidents studied earlier in the project. It was felt that the choice of velocity was not important except that the kinetic energy of the bodyform should exceed a certain threshold value necessary to deform the door and that a higher velocity would merely lead to a higher final velocity of the bodyform as it was projected through the door after unlatching.

The initial tests were conducted with a bodyform having an upper arm attached to the torso at the shoulder; in later tests, the upper arm was fixed at the elbow. This change meant that the dummy's ribs no longer contributed to the performance of the bodyform, leading to a simpler device. Other variations were introduced by varying in the impact velocity and the means of mounting the striker pin on the rig. A total of 19 tests were conducted in this series, using front doors from Vehicle A and these are summarised in Table 3.

The most interesting aspect emanating from the analysis of the results from these tests was the effect that realistically representing the surrounding vehicle structure of the door had in allowing latch activation to occur. The door closure seal pressures were reproduced by encorporating actual seals from the vehicle concerned so that normal opening and closing efforts were maintained. In the early tests the striker was attached to a simple box section to represent the B-pillar. To investigate the forces acting on the latch and striker the components were instrumented with displacement transducers and tri-axial loadcells. The effects of configuring the test rig in these ways tended to lead to poor repeatability. Analysis of high speed cine suggested that the "B" pillar location was too rigid. In an effort to overcome this problem the rig was adapted to encorporate the body "B" pillar fixed at its extreme ends. The fixing position was designed to represent the "B" pillar connection at the cant rail and the rocker panel, thus increasing the degree of representativeness of the door system in the rig. During the second half of the test programme it became clear that a test which very accurately reproduced the circumstances in real world accidents had been achieved. The door was opening on more than 80% of occasions and the damage to the door was virtually identical to that seen in the accidents used to model the test. An example of the damage is shown in Figure 4.

Validation Testing

The investigation had up to this point in time focused on reproducing the circumstances of real-world accident situations with vehicle components that had exhibited the problem. The next stage of the study was to investigate the possibility of the same problem occurring on vehicles that were not currently included in the national accident data file. Trials were conducted to evaluate the performance of a random selection of six models of vehicle readily available on the UK market.

Each test was conducted using the front door taken from cars which had been purchased new for a crashworthiness evaluation programme. Two specimens of each type of door (both left hand) were tested. The complete "B" pillar was removed from the vehicle to mount the striker for the test, and the vehicle was carefully measured before removal of

Table 3. Results of dynamic impact tests.

Test	Test Configuration	Latch Activation		Bodyform Data		Test Result
		Yes	No	km/hr	(g)	
1&2	Striker supported on hollow box section 10 SWG	*		24.8	30	Door Opened
3	Triaxial Loadcell on Striker. Rotary Pot on Latch Lever. Latch Rod Removed		*	24.7	25	Door Remained Closed
4,5 &6	As 3 but with Latch Rod Fitted. 16, 20, 27 km/hr		*			Door Remained Closed
7	As 3 but 24 km/hr Impact Speed	*		24.8	28	Latch Released but Rig Moved, Trapping Door
8	Striker Supported by B-pillar Secured at Top and Bottom	*		24.5	44	Door Opened
9	As 8	*		24.5	52	Latch Pawl Jammed
10&11	As 8	*		24.3	44	Door Opened
12	As 8		*	24.4	47	Door Remained Closed Latch Mechanism Corrosion
13	As 12. Latch Lubricated	*		24.4	37	Door Opened
14	As 12. But with Interior Trim	*		24.4	37	Door Opened

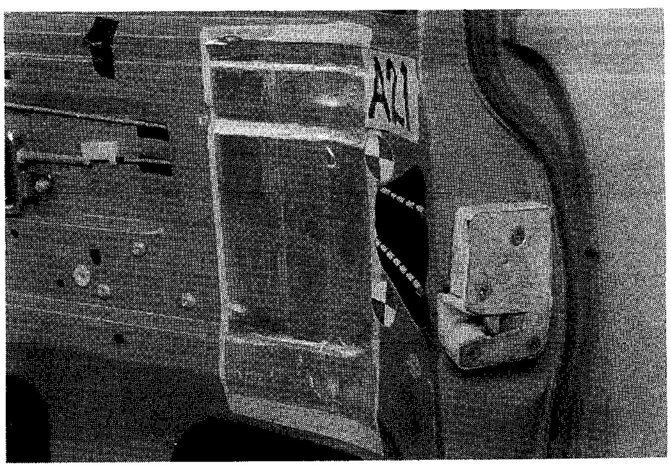


Figure 4. Typical door inner panel distortion

these components so that the relationship of the latch and striker could be accurately reproduced in the test.

The tests were conducted in accordance with the final test method used in the development trials, that is, propelling the bodyform at a 24.1 km/hr impact velocity, with the arm of the bodyform aligned toward with the front seat R-point of the vehicle, to strike the door in free flight.

The results of the tests showed that the door on two of the models activated and released, three failed to activate and one activated but failed to release completely. However, in the case of one of the models where the door opened, the unlatching did not persist long enough to allow the door to pass the intermediate-latched position and the door remained closed.

Engineering Evaluation of Results

The evidence gathered in experimental investigation has suggested that the occurrence of door unlatching is determined to a great extent by features in the design of the door and the latch linkage. In other words, the potential for this type of accident seems to lie in specific types of linkage rather than every linkage of the pull-rod.

It is apparent that in most cases the linkage rod is situated in a vulnerable region of the door and that the degree of outward lateral deformation of the rod and the force to achieve this deformation is small.

The performance of any particular design using a pull-rod appears to be determined to a great extent by the pattern of deformation of the door when struck on the inner face. In particular, it is whether the deformation of the door extends into the end face adjacent to the latch and the behaviour of the rod itself under impact which is crucial in deciding whether the door will open or not.

This suggests that the simplest solution may be to increase the beltline stiffness to reduce intrusion but not enough to cause serious shoulder injury. Alternatively, additional structural stiffness could be incorporated into the corner of the door, in the region of the latch, which will more easily transmit the bending into the end face of the door. However, this will work best where the latch incorporates a fairly restrictive slot, around the head of the striker pin. Doors having a more open slot, or a channel in the striker would not benefit from this.

An alternative solution to the problem, still using the pull-rod linkage, might be to incorporate some free-play into the handle so that it was free to move when the rod is displaced in a direction opposite to its normal travel. If sufficient movement were allowed, it is probable that the rod would not activate the latch under accident conditions.

The problem could also be solved by using other forms of mechanism to release the latch. A "bowden" cable is used in some of the luxury cars produced in Europe. Provided that a generous length of outer casing is incorporated, this would be free from activation problems when routed away from the critical area.

General Observations

The investigation has shown that the opening of car doors as a result of inadvertent unlatching is a possibility in a significant number of accidents and that some action is necessary in order to reduce its occurrence. The choice of an appropriate course of action is a complex one, since there are many considerations which will ultimately determine the practicability of any solution which might be adopted.

In the UK the Department of Transport consider that the necessary changes in vehicle design should be stimulated by amending the existing ECE Regulation concerned with these components. This item has been on the agenda for discussion by the ECE Group of Rapporteurs on General Safety matters (GRSG) on a number of occasions over the last two years. The occurrance of the accident circumstances in question for a variety of traffic environments has clearly to be established. The implications of seat belt use by the occupants of vehicles has a bearing on the efficacy of any proposed changes. In Europe, restraint use is generally quite high for front seat occupants, although wearing rates are variable over a large group of countries(7). However, restrained occupants can be partially ejected and seriously injured, and the majority of rear seated occupants are not restrained. Therefore a significant risk of injury as a result of latch activation is present.

Engineering solutions which will reduce the incidence of inadvertant door opening are not difficult or expensive to introduce, and this research study has demonstrated that a simple component test procedure can be used to evaluate design changes. The test has all the desirable characteristics for a crashworthiness assessment tool in having the ability to discriminate levels of safety performance for various vehicle safety designs. It has a high degree of repeatability and reproducibility and a low level of test complexity. The research has provided yet a further example of a component test that could provide real benefits for improving vehicle crashworthiness.

Conclusions

1) Passenger car door design, for good crashworthiness, remains an important area for future improvements even though legislation currently exists concerning the performance of hinges, latches and side door intrusion strength.
2) The component (sub-system) test can provide an effective contribution as a development

and legislative tool in reducing the level of road accident injury.

Acknowledgements

The opinions, findings and conclusions expressed in this paper are those of the authors and do not necessarily reflect the views of MIRA.

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Motor Vehicle Manufacturers Association Subsystem and Full Vehicle Side Impact Test Procedure Development Program

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History

In 1982, the MVMA undertook a comprehensive research program to develop and evaluate potential improvements in side impact protection(4). The program was structured with several tasks including:

- Accident Data Analysis
- Effectiveness
- Injury Criteria
- Develop Subsystem Test Procedure
- Build Subsystem Test Device
- Develop Full Vehicle Test Procedure
- Build or Acquire Full Vehicle Test Equipment
- Evaluate Full Vehicle and Subsystem Test Procedures
- Evaluate Benefits of the Test Procedures

The accident data analysis has been completed and was reported previously(1). A review of the accident data show that the upper torso to the side surface of the vehicle is the predominate injury category for serious to fatal injuries. These injuries involved fractured bones and lung damage caused by the side surface of the vehicle. The abdomen to the side surface is the second most frequent injury category where the side surface interaction with the kidney, spleen and liver were the major injury mechanisms.

The test effectiveness portion of the study concluded that for any given thorax to side surface test procedure, three factors are important(7). These are:

- A high level of repeatability and reproducibility plus a low level of test complexity;
- Ability to discriminate levels of safety performance in different vehicle designs;
- Ability to accurately represent the injury category and injury severity occurring in the real-world under a variety of conditions which reflect real-world occurrences.

The hardware and test development and evaluation portion considered two basic approaches: full vehicle and subsystem. MVMA did not choose a full vehicle test over the subsystem test since both have merit. Both test procedures were supported in this program to develop information and answers to basic questions that are needed to implement a meaningful side impact test procedure.

This report summarizes much of the hardware development and evaluation including the results of 16 full vehicle tests and 12 subsystem tests comparing the variability of each, their ability to discriminate potential countermeasures and provide estimates of the repeatability and reproducibility of the respective test devices.

The countermeasures considered for full vehicle testing consisted of interior padding, door stiffness and occupant spacing relative to the door. Countermeasure considered for subsystem testing was interior padding. The objectives of both full scale and subsystem test series was to:

1) Estimate the relative ability of various response measures from each test device to discriminate changes due to application of the countermeasures.
2) Estimate the variability and correlation of the various response measures.
3) Make statistical inferences concerning interaction of the various countermeasures.
4) Determine the effects of various levels of data filtering on the results.
5) For full vehicle testing, provide estimates of dummy reproducibility and repeatability by introducing two different SID dummies in the full vehicle matrix.*

Completed Testing

A series of 16 full vehicle tests were conducted under highly controlled conditions using the proposed National Highway Traffic Safety Administration (NHTSA) side impact test procedure. See Figure 1.

The test vehicles were sequentially produced 1985 Ford LTD's identically equipped to reduce vehicle variability.

Eight of the vehicles were modified structurally to insure a measurable response difference from the standard vehicle. These modifications consisted of strengthening the "A" pillar of the vehicle and installing a channel between the "B" pillars. Crush data showing Force vs. Displacement for the structural modifications are shown in Figures 2 and 3.

Eight of the vehicles were equipped with padding. This padding was selected of sufficient capacity to prevent "bottoming out" at the proposed test velocity. A pad approximately 5 inches (12.7cm) of Arcel 506 (32.1kg/m^3) was selected.

Finally 8 of the vehicles were tested with the dummy spaced 12.7cm (5″) from the door padding in various combinations with the other 2 countermeasures above. The test matrix for the full vehicle testing consisted of replicating each of eight test conditions composed of a matrix of:

Structure
Padding
Spacing of the dummy from the door

The test matrix is shown in Table 1.

* Separate estimates of reproducibility and repeatability were not obtained in the subsystem testing because only one test device was available.

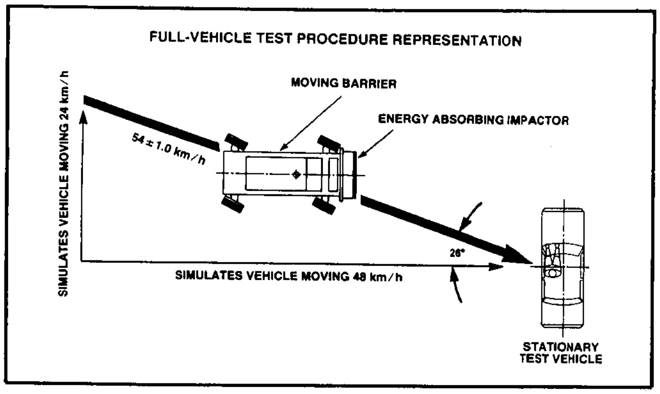


Figure 1

To avoid interactions with the driver controls which might confound the results the decision was made to conduct the tests with the dummy in the passenger position.

Measurements obtained from the SID are shown in Figure 4. Additional accelerometers were placed on the test vehicle right front door at the centerline, midrear, upper centerline and midfront as well as on the left sill. This information was used to determine the striking velocity of the dummy and accordingly was used in the selection of the subsystem test velocity. The moving barrier had accelerometers at the center of gravity and the rear member.

The primary analytical technique employed was the Analysis of Variance (ANOVA) of maximum values

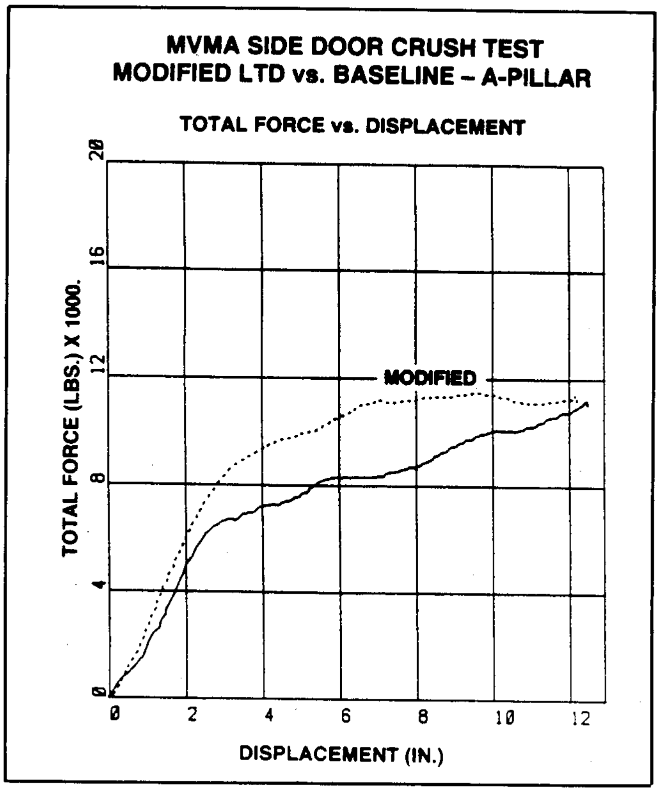


Figure 2

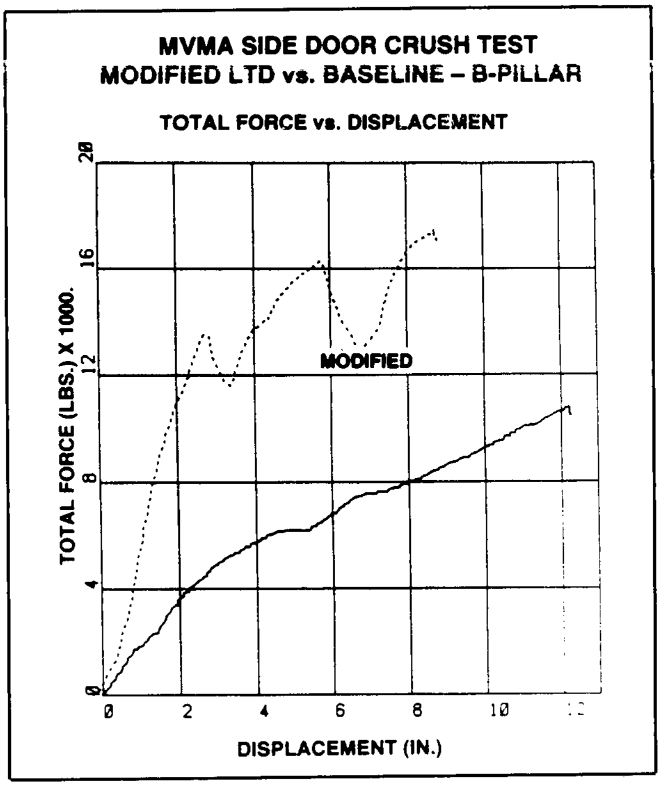


Figure 3

although analyses were done using sampled time histories and vector methods. Both repeatability and reproducibility were studied regarding the sensitivity of the test procedure to discriminate substantive changes in test conditions(2).

A summary of the full vehicle data analysis follows:

- Padding had the most significant effect on dummy responses.
- For the conditions tested, dummy spacing did not have a significant effect on the dummy response.

Table 1.

SUMMARY OF TEST CONDITIONS

Test Date	Test Struct.	Door Conditions	Space	Dummy	Impact Velocity m/h	LTD Weight Lbs
5/29/85	B	H	N	B	33.6	3247
6/30/85	B	H	N	B	33.3	3239
6/07/85	B	H	F	A	33.4	3261
6/17/85	B	H	F	A	33.4	3234
6/26/87	B	P	N	A	33.6	3216
7/03/85	B	P	F	B	33.5	3214
7/15/85	B	P	F	B	33.6	3215
7/16/85	B	P	N	A	33.5	3182
8/05/85	M	H	F	B	33.5	3262
8/08/85	M	P	N	B	33.6	3249
8/18/85	M	P	N	B	33.5	3253
8/30/85	M	H	F	B	33.5	3242
9/04/85	M	P	F	A	33.5	3238
10/02/85	M	P	F	A	33.5	3204
10/07/85	M	H	N	A	33.6	3263
10/18/85	M	H	N	A	33.6	3240

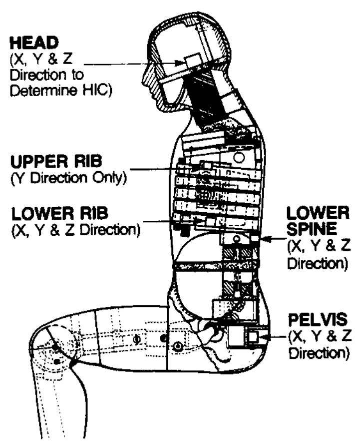
Structure	Door	Spacing
B-Baseline	H-Hardboard	N-Near (dummy next to door)
M-Modified	P-Padded	F-Far (dummy 5 " from door)

- Dummy pelvis and lower rib accelerations are more discriminating than other measures.
- High correlations exist among response parameters except for the head response.
- Standard deviations for acceleration measurements range from 3 to 10G's.
- TTI standard deviation point estimate is approximately 4.4 TTI units.
- Interactions between padding, structure and spacing were generally small for dummy responses.
- There were only minor differences between dummy responses filtered at Class 1000 and Class 600.
- Dummy reproducibility is estimated to be of comparable or smaller magnitude than dummy repeatability.

Subsystem Tests

A series of 12 subsystem tests were undertaken utilizing the MVMA developed subsystem test

SIDE IMPACT DUMMY



NOMENCLATURE FOR ACCELEROMETER LOCATIONS

Head Res.	Head Resultant
Up Spine Res.	Upper Spine Resultant
Lo Spine Res.	Lower Spine Resultant
Pel Cent Res.	Pelvis Center of Gravity Resultant
T01YG2	Upper Spine Y-Axis
T01YGB	Upper Spine Y-Axis Redundant
T12YG2	Lower Spine Y-Axis
T12YGB	Lower Spine Y-Axis Redundant
RURYG2	Right Upper Rib Y-Axis
RURYGB	Right Upper Rib Y-Axis Redundant
RLRYG2	Right Lower Rib Y-Axis
RLRYG2	Right Lower Rib Y-Axis Redundant
PEVYG2	Pelvis Y-Axis
RRTYD2	Right Rib Displacement
RFDYG	Right Front Door Y-Axis

Figure 4

procedure(3) and the MVMA developed subsystem test device. The doors used for the tests were new 1985 Ford LTD doors identical to those used in the MVMA full vehicle test series. The modifications to the doors were identical to the modifications made to the full vehicle doors with baseline and padded doors.

Two levels of each variable were included in the experiment design. The door panel modifications consisted of, in one case, replacing the manufacturers door panel with a piece of 3.2 mm thick masonite hardboard. The second level consisted of the hardboard panel plus a piece of 127 mm thick foam. The foam, the same as used in the full scale tests, was Arcel 506 (32.1 kg./m). The level of padding was identical to that used in the 16 full scale crash tests.

The two velocities used for the subsystem testing were obtained from an analysis of the 16 full vehicle test series. In a side impact accident as the striking vehicle crushes the struck door, the struck vehicle moves relative to the struck vehicle occupant. The struck vehicle occupant slides until contact with the door is made. From the time the occupant contacts the inner door, the occupant velocity increases and approaches that of the door and the front of the striking vehicle. The occupant then rebounds away from the door.

This time of events is shown in Figure 5. For the 16 full vehicle tests, instrumentation to calculate velocity was installed in the dummy, door and the striking barrier. The door mounted accelerometers did not yield reliable data as it is believed the accelerometers rotated during the collision therefore yielding incorrect velocities. The average impact velocity of the baseline vehicle, dummy near the door, 22.3 mph (35.9 k/h). The average velocity of the padded door, for spacing and structurally modified vehicle was 16.4 m/h (26.4 k/h). These average values were then used to develop the subsystem test speeds of 25.7 k/h and 37 k/h.

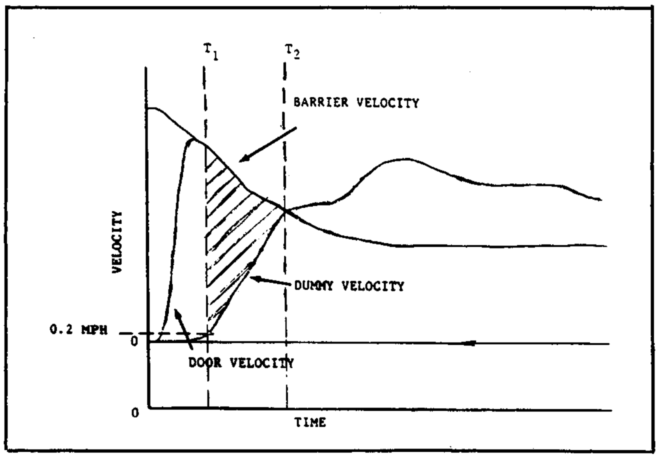


Figure 5

Experimental Design

As with the full vehicle crash tests, the test matrix was designed as a factorial experiment. There were two variables (padding and impact velocity), each at two levels. This results in a 2 x 2 factorial with a total of four treatments. Each test condition was tested three times, resulting in a total of twelve tests. This is different than the full vehicle test experiment which consisted of three treatments of two levels (2 x 2 x 2), and with each test being repeated once resulted in a total of 16 tests. Shown in Table 2 are the test conditions for the twelve subsystem tests.

In the full vehicle crash test, two different dummies were used. This allowed the results to be analyzed relative to two error components, i.e., within the dummy and between dummies. The between dummies error indicates the reproducibility of the test procedure. However, since only one subsystem test device was used for this testing program, then only "within" subsystem test device error can be measured. This error can be compared to the "within" dummy error measured during the full scale crash tests.

Test Procedure and Equipment

The test procedure that was defined by MVMA and titled: "A Proposed Procedure for Side Interior Testing with a Side Impact Device"(3) was used. The test procedure and characteristics of the test device are summarized in the following sections.

Side Impact Test Device

MVMA through MGA Research Corporation had designed and fabricated the Side Impact Test Device. The test device is designed to represent a human thorax in an automobile side impact collision. The device, illustrated in Figure 6, is a two mass impactor with a compliance between the two masses. The smaller frontal mass represents the rib mass of the

Table 2.

No.	Date	Door	Impact Velocity (kn/hr)
1	6/27/87	Hardboard	Low
2	6/30/87	Padded	High
3	7/01/87	Padded	Low
4	7/01/87	Hardboard	High
5	7/02/86	Hardboard	High
6	7/03/87	Hardboard	Low
7	7/07/87	Padded	Low
8	7/07/87	Padded	High
9	7/07/87	Padded	Low
10	7/08/87	Padded	High
11	7/09/87	Hardboard	Low
12	7/09/87	Hardboard	High

LOW SPEED TARGET = 25.7 km/hr
HIGH SPEED TARGET = 37.0 km/hr

thorax while the large mass represents the remaining mass of the thorax.

The test device is a pneumatically fired piston with speed capability up to 40 km/hr. The small mass moves relative to the large mass in a piston/tube arrangement as well. Two pieces of urethane placed between the small mass and the large mass provide the desired compliance to the system selected to result in a force-time history for the test device, at a particular impact velocity, that falls within a corridor derived from test data representing human thorax response. There is a piece of foam material on the surface of the small mass which is intended to contribute to the response of the test device as well. In Figure 7 are data illustrating the force time response of the device(6). Besides the requirement of being within both the one meter and two meter drop corridors, there are requirements on the peak relative deflection of the two masses.

There are various sensors on the test device and a data acquisition/computer system to record and process the data. The sensors include an accelerometer on the small mass and one on the large mass, a displacement potentiometer to measure the displacement of the small mass and another displacement transducer to measure the displacement of the large mass. There is also a contact switch and time zero channel to record the time that the impactor first contacts the test specimen.

The intent of the MVMA test procedure is to provide a repeatable standardized test that can be used to evaluate the energy absorption characteristics of vehicle interior door panels. The test procedure calls for a pre-crush of the exterior of the door to simulate the distortion that occurs in a side impact collision. Then, while the pre-crush device is holding the exterior of the door, the interior surface of the door is impacted with the side impact test device at a specified velocity.

The side impact test device was designed so it could be inserted into an automobile through a door opening to test the opposite side door. However, if the interior impact area is entirely within the periphery of the door, the door can be removed from the vehicle and fixtured in a frame that attaches at the door hinges and latch, and supports for the two sides and lower ledge of the door at least as rigid as the vehicle body. This approach was used for fixturing the doors for this testing program.

The interior impact location was determined using dimensions and practices outlined in SAE J826, April

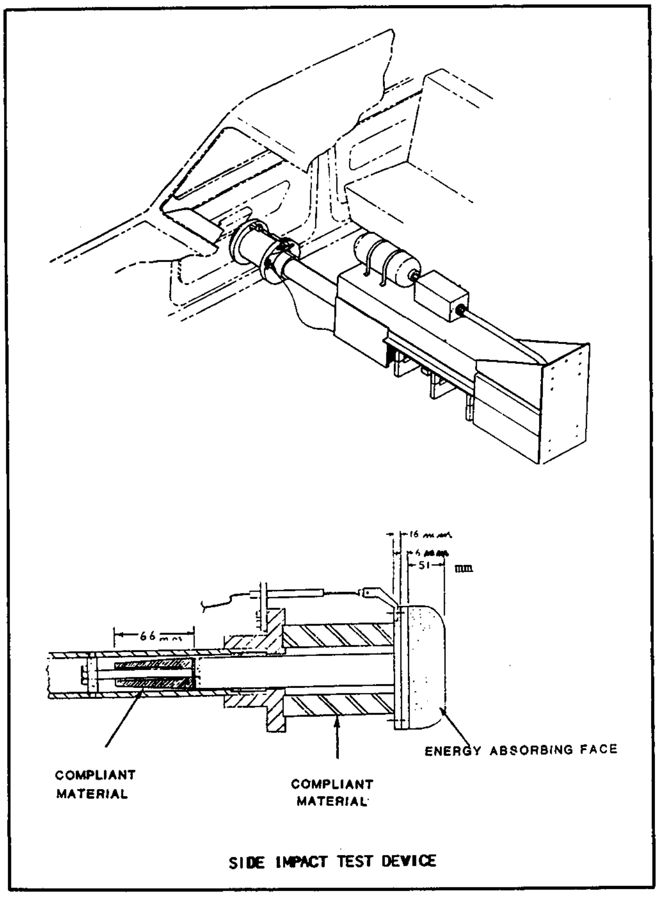


Figure 6

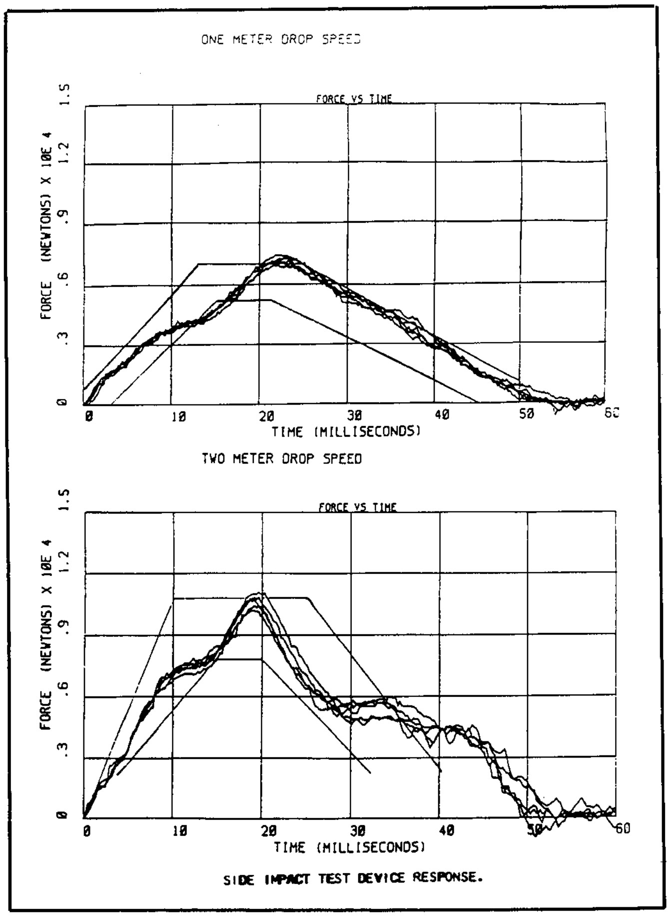


Figure 7

1980 and J1100, June 1984. The impact zone consists of an area that is created by placing the J826 H-point template in both the forward and rearward most seating position, the forward most position and the rear outline in the rearward most position. The upper boundary is the lower daylight opening of the window, and the lower boundary is a horizontal line 170 mm above the H-point with the seat in the rearward most position.

The exterior pre-crush procedure consists of displacing a ram of specifications illustrated in Figure 8 into the exterior of the door a total of 230 mm. The ram was placed on the end of two hydraulic cylinders which were controlled with a loop displacement feedback servo system. The rate of ram extension was approximately 2.0 mm per second. The ram is to be positioned such that the direction of travel is perpendicular to the longitudinal center line of the vehicle and the vertical center line of the ram passes through the H-point with the seat in mid-position. However, if the interior impact locations are located entirely within the body opening for the door, the ram will be positioned so that the periphery of the ram clears the body opening by a minimum of 75 mm. This was the case for the twelve tests conducted in this project. The bottom edge of the ram was positioned 400 mm above the ground surface reference plane for the vehicle under test.

Shown in Figure 9 is a view of the test door in the fixture. The pre crushing has taken place and the ram is held against the outer side of the door.

Prior to installing the door into the fixture, the door's inner panel was removed and replaced with a piece of 3.2 mm thick masonite hardboard. For six of the twelve tests, a piece of 127 mm thick foam was attached to the masonite in the impact area. The foam was not added until after the pre-crush.

The conclusions of the subsystem test series were:

- Impact velocity is a factor affecting discriminatory ability of the subsystem tests. Except

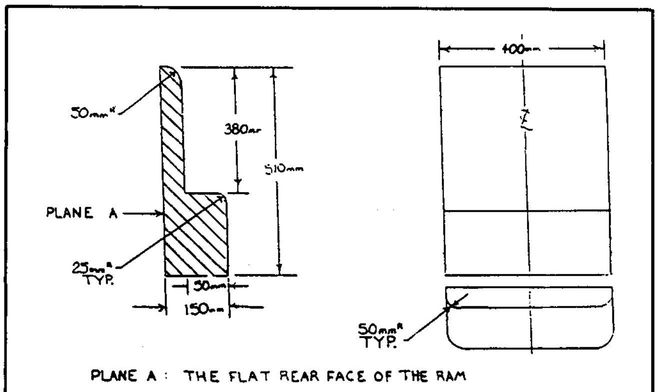


Figure 8

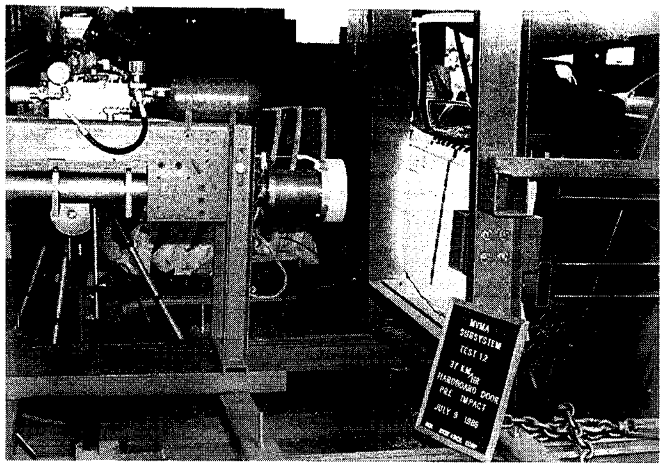


Figure 9. Subsystem test set-up

  for the peak displacement response, discrimination was better at the high velocity than at low velocity.
- Subsystem tests are highly repeatable, as evidenced by small error standard deviations and strong ability to discriminate between treatments. Repeatability and discriminatory ability are indicated to equal or exceed that of the full vehicle tests.
- Many of measured responses in the subsystem tests are highly correlated; in particular, viscous criterion (V*C) and TTI are essentially equivalent in their response to the treatment variable in the test sequence.
- Data filtering of Class 1000, 600 and 180 established high correlations (0.99 - 1.00) illustrating the insignificance of the degree of data filtering.

Comparison of Subsystem and Full Vehicle Testing

It is instructive to compare the results of the subsystem tests with those of the full-vehicle tests. For this comparison, it suffices to compare only the effects of padding, inasmuch as this is the dominant variable in the full-vehicle tests.

Comparison of the subsystem and full-vehicle tests is based on the following premises:

1) Padding was similar for the two series of tests.
2) Similar doors were used in the two test series.
3) The impact velocities for the subsystem tests were selected so as to bracket the range of severity for the full-scale tests.

In the comparison, attention will be directed to how closely the subsystem and full-vehicle tests agree and to the comparative repeatability of the two tests.

Inasmuch as the effects of structure and dummy spacing were small in the full-vehicle tests, it is considered logical to pool the results for the two structures and the two spacings in computing representative responses for the full-scale tests performed with and without padding. Mean responses with and without padding can then be compared with subsystem test results with and without padding.

How the pooled levels of structure and dummy spacing relate to the impact-velocity variable in the subsystem tests is unknown. Accordingly, it is appropriate to treat subsystem impact velocity parametrically. In Table 3, the pooled results for the full-vehicle tests are compared with:

1) Subsystem test results at low velocity
2) Subsystem test results at high velocity
3) Subsystem test results taken as the mean of low-velocity and high-velocity results

In the comparison, small mass accelerations are regarded as being analogous to upper or lower rib accelerations, and large mass accelerations as being analogous to lower spine accelerations. Also displayed are the ratios of the response *without* padding to the response *with* padding for both full-scale and subsystem tests.

The table suggests that it should be possible to "calibrate" the subsystem test velocity in such a way that the subsystem test would serve as an effective surrogate of the full-vehicle test. For example, for data filtered at Class 1000, the performance of the full-vehicle test appears quite comparable to the subsystem tests performed at the higher velocity.

An even more cogent example is provided by the comparison of full-vehicle and subsystem results for TTI, as shown in Table 4.

It is seen that the high velocity results essentially reproduce not only the TTI ratio for the full-scale tests but closely approximates the actual values of TTI for the hardboard and padded cases as well.

Subsystem tests may have a repeatability advantage in the sense that they tend to yield higher levels of significance for treatment effects than do the full-vehicle tests when the two tests are performed under what are indicated to be comparable conditions of exposure severity.

Further insight with regard to comparative repeatability of the subsystem and full-vehicle tests is afforded by Table 5.

Direct comparison can be made of the error (repeatability) standard deviations for the two tests and, more importantly, of F-Ratios and significance levels for the padding treatment variable for the two tests.

The values quoted for the full-vehicle tests are those obtained from a three-way Analysis of Variance, in which there were eight treatments, each with two replications. Thus the padding treatment main effect has one degree of freedom and the error estimate has eight degrees of freedom. The value quoted for the subsystem tests were derived from the two-way Analysis of Variance, in which there were four treatments and three replications for each treatment, also yielding a comparison involving one and eight degrees of freedom, respectively, for treatments and error. In this connection, it is fortunate that the number of degrees of freedom available for testing the significance of padding is the same in the full-vehicle and subsystem tests.

It is indicated that repeatability for the subsystem tests is, for the most part, comparable to that of the full-vehicle tests, whether viewed in the absolute sense (i.e., actual values of repeatability standard deviation) or in the discriminatory sense, as indicated by F-Ratios and significance levels. In interpreting this conclusion, keep in mind that the results for the

Table 3. Comparison of full-scale and subsystem test results.

COMPARISON OF FULL-SCALE AND SUBSYSTEM TEST RESULTS

Class 1000 Filtering

	Full Scale Tests		Subsystem Tests					
			------Large Mass------			------Small Mass------		
Padding	Lower Spine (T12YG2)	Lower Rib (RLRYGB)	Low Vel.	High Vel.	Ave. Vel.	Low Vel.	High Vel.	Ave. Vel.
			Accelerations in g's					
Hardboard	-123.1	-103.3	41.0	103.2	72.1	52.6	151.8	102.2
Padded	-71.0	-53.6	32.0	67.9	50.0	43.4	74.5	59.0
Ratio	1.73	1.93	1.28	1.52	1.44	1.21	2.04	1.73

Table 4. Comparison of full-scale and subsystem test results.

COMPARISON OF FULL-SCALE AND SUBSYSTEM TEST RESULTS

Thoracic Trauma Index TTI(40)

Padding	-----Full Scale Tests-----	-----Subsystem Tests----- Low Vel.	High Vel.
Hardboard	142.3	100.1	139.1
Padded	112.6	87.5	116.4
Ratio	1.26	1.14	1.20

subsystem tests are based on data pooled from two levels of impact velocity, one of which is too low to afford good discrimination. If all tests had been performed at the higher impact velocity, higher significance levels would have been realized. Furthermore, several potentially significant sources of variability in the full vehicle testing discussed earlier were not considered in the test sequence e.g., dummy interaction with the steering system.

For a complete analysis of the subsystem test conditions and statistical analysis the reader is referred to the final report in reference 5.

Future Full Vehicle Testing

Since the beginning of the original program a new anthropomorphic test device has been under development in Europe. In conjunction with the test device the European Community has been developing a side impact test procedure. The MVMA has expanded its original testing program to include the European test procedure and dummy (commonly called EUROSID) into the scope of the work for harmonization purposes. MVMA wishes to address the harmonization issue and to investigate all the potential test procedures and test devices to strive for meaningful tests.

In this regard MVMA has purchased 3 European Developed Side Impact Dummies (EUROSID's) from the Netherland Road Research Laboratories (TNO). These dummies are to be delivered within the next few days. The United Kingdom Department of Transport, Transportation and Road Research Laboratories has graciously provided the MVMA with 8 European side impact barrier faces meeting the specifications set forth by the government agency. In addition, eight barrier faces have been made available from the CCMC and can be used in the event the European Governments decide to adopt a more rigid barrier face.

At this time, MVMA has generated a factorial experiment to test a combination of identical vehicles as used for the completed 16 full vehicle test utilizing the NHTSA test procedure. Table 6 highlights the complete full vehicle testing program.

The test vehicles used for the full vehicle tests, either European or NHTSA, will be 1985 model year Ford LTD's to allow comparison to previous test results. The treatments to the vehicles will be identical to those treatments used in the first series of vehicle tests.

All variables will be held as close as possible to conditions that the original 16 vehicle tests were conducted. The tests will be conducted at the Transportation Research Center of Ohio utilizing the same instrumentation where possible, same type of vehicle, same test site and same personnel.

The second series will provide a direct comparison of the EUROSID repeatability and reproducibility as well as a comparison of how the EUROSID relates to the NHTSA SID in the ability to discriminate between changes in occupant protection in side impacts.

The third series of tests will include full vehicle test using the EUROSID, European test procedure and European barrier. This test series will permit an analysis to be made of the ability of the European test

Table 5. Comparison of full-scale and subsystem test repeatability.

COMPARISON OF FULL-SCALE AND SUBSYSTEM TEST REPEATABILITY

	----------Full Scale Tests---------- Lower Spine Accel. (Class 1000)	Lower Rib Accel. (Class 1000)	TTI	------------Subsystem Tests------------ Large Mass Accel.	Small Mass Accel.	TTI
Error M.S.	29.81	35.93	18.88	30.0	18.4	29.2
Error Std. Dev.	5.46	5.99	4.34	5.48	4.29	5.40
F-Ratio	139.53	49.10	186.41	49.1	306.5	32.20
Signif. Level	2.50-06	1.12-04	<1.00-06	1.12-04	1.16-07	4.66-04

Table 6.

SERIES	TEST PROCEDURE	DUMMY	BARRIER FACE	STATUS
1	NHTSA	NHTSA	NHTSA	Complete
2	NHTSA	EUROSID	NHTSA	Start 5/87
3	European	EUROSID	European	Start 6/87
4	European	NHTSA	European	Start 8/87
5	NHTSA	NHTSA	European	Start 9/87

and EUROSID to discriminate between potential countermeasures that were investigated in the series of sixteen original tests. An understanding of the repeatability and reproducibility of the proposed European test procedure will also be gained.

The fourth series of full vehicle tests will be run utilizing the European test procedure but substituting the NHTSA Side Impact Dummy for the EUROSID. As with all testing in this program, the vehicles, countermeasures, instrumentation, test site and personnel will be the same as used for the other test series to minimize variability. An analysis of the data from this series will permit a direct estimation of the sensitivity of responses of the EUROSID to the NHTSA Side Impact Dummy for identical test conditions.

The fifth sequence of tests planned at this time will be NHTSA type tests using the NHTSA Side Impact Dummy but substituting the European barrier face for the NHTSA barrier face. An analysis of these data should indicate if the test could be conducted with either barrier face or if a correction factor may be applied or if there is no correlation between tests when the barrier faces are substituted.

MVMA has purchased 3 EUROSID dummies for this test program and two of the dummies will be utilized in the test series. Two dummies will provide dummy variability. The tests will be run in the sequence shown in Table 6. The third dummy will be held in reserve to be used only if there is a catastrophic failure in either of the two dummies selected for the tests.

The full vehicle test portion of the test series is scheduled to be complete in the fall of 1987. A complete statistical analysis will be undertaken to determine and understand the interactions and variability of the test procedures and dummies when related to common occupant protection measures in nearly identical vehicles. It is hoped that harmonization can be achieved from a comprehensive program of this nature.

Future Subsystem Testing

One major aspect of the subsystem test is that consideration must be given for structure and padding features added for occupant protection. The series of 12 tests conducted using the subsystem test procedure used test velocities generated from the first series of 16 full vehicle tests. Accelerometers were placed in the door for the 16 full vehicle tests and an estimation was made of the door velocity when the dummy contacted the door during the full vehicle test. As detailed in the final report on the full vehicle testing the door velocity estimates are not closely bounded and are subject to error(2). One possible reason is that as the door crushed the orientation of the accelerometers changed, thus changing the apparent velocity that the door came in contact with the dummy.

MVMA will undertake a comprehensive analysis to analyze full vehicle test data, subsystem test data and cadaver data to establish a method to determine subsystem test velocities taking into account vehicle structure and other design consideration that may have a separate or synergistic effect on occupant protection. Full vehicle test data will be investigated to determine if a correlation exists between certain vehicle parameters and door velocity at the time the dummy contacts the door during a side impact collision. Subsystem test results will be evaluated to determine if crushing and holding the door in the fully crushed position with an unyielding device is representative of how a realistic test could be conducted and if there are relationships between full vehicle test velocities and subsystem test velocities for tests already conducted. An analysis of existing cadaver data will also be undertaken to determine factors in human anatomy and injury mechanisms and injury severity and determine if these can be related to full vehicle tests as well as subsystem tests.

The prime output of this analysis of existing data will be a definition of a relationship or relationships between vehicle structure, padding and resultant subsystem test velocities. The goal is to develop a method to determine the test velocity that the subsystem test should be conducted for various types of vehicles. The relationships between vehicle structure, mass and other as of yet undefined factors will be developed and presented.

This subsystem test data analysis and recommendations will start the summer of 1987 and be complete by the fall of 1987.

The MVMA side impact program is a comprehensive multi task program that is scheduled for completion this year. The results of the completed portion of the program have provided insight into side impact testing both for full vehicle tests as well as for subsystem tests. The remainder of the program will enhance the initial knowledge by investigating other newly developed test procedures and test dummies and relating this information to that already gained.

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Further Consideration of the European Side Impact Test Procedure

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Introduction

In 1982 the procedure proposed by EEVC for evaluation of protection in side impact was presented at the 9th ESV conference(1).

This procedure was based on the data available at that time concerning the side impact accident situation, but none had the experience of the proposed test methodology.

The three main topics defining the procedure are the tests conditions, the characteristics of the bullet device and the dummy to be used.

The main parameters of the EEVC procedure as proposed in 1982 were:

- The tested car must be stationary.
- The striking bullet would be a mobile deformable barrier.
- The trajectory of the MDB would be perpendicular to the sagittal plane of the tested car.
- The geometrical and the force deflection characteristics of the deformable front face of the barrier were defined in the procedure.
- The results would be analyzed on taking into account the values of injury parameters recorded on a new side impact dummy.

The procedure is summarized on Figure 1 and 2. During the last five years, many tests using the proposed procedure were conducted in several European laboratories(2) as well as a new dummy (called EUROSID) was developed and evaluated.

It is time now to take into account this gained experience and to reconsider the original EEVC specifications of the European side impact test procedure.

Test Conditions

The original procedure proposes a pure 90° trajectory of the bullet mobile deformable barrier relative to the struck car which is stationary during the test. This configuration does not represent exactly the most frequent accident configurations.

In real accidents, the struck car is also moving but at a speed which is generally lower than the striking car. To simulate such a configuration, it is necessary either to make a test against a moving car (which is complicated and has a poor repeatability) or to give a crabbed motion to the barrier.

EEVC has decided to 1982 not to keep a procedure with a crabbed motion of the barrier because it was considered that the axial components of the forces exerted to the occupants of the struck car would not increase the severity of injuries to be mitigated by side impact protection improvement. However, it is important to verify if this assumption is true.

In the frame of the validation program of EUROSID a series of tests were performed using the EEVC procedure (stationary struck car) and two other tests

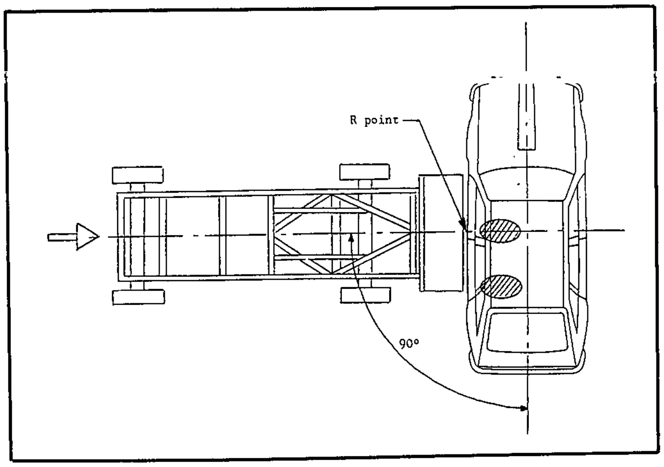

Figure 1. Configuration of the EEVC side impact test procedure

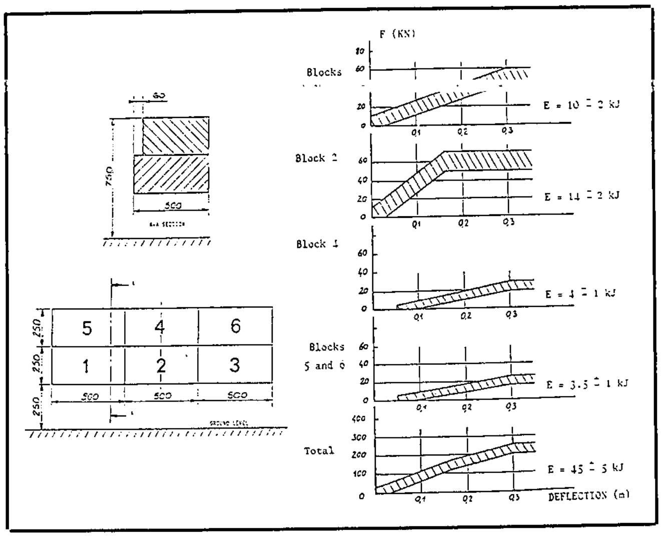

Figure 2. Main characteristics of the EEVC mobile deformable barrier

in the same conditions, but with a crabbed barrier(3). The tests with the crabbed barrier were performed at 54 km/h, which correspond to 50 km/h perpendicular component.

In terms of car deformations, no important difference was found between the tests in EEVC procedure and the crabbed barrier tests. All the tests were performed with a EUROSID dummy as driver of the struck car.

Analysis of injury parameters recorded during these tests shows that the tests with a crabbed barrier gives lower value, as indicated in Table 1.

These results confirm that there is no reason to modify the EEVC side impact test specifications.

The Mobile Deformable Barrier

When the EEVC side impact test procedure was set up, nobody had the experience of using a mobile deformable barrier corresponding to the specification.

During the last years, many tests using the EEVC mobile deformable barrier were performed in several European laboratories. Two characteristics can be reconsidered taking into account the experience gained with this test, the first one is the geometrical specification, especially the ground clearance, and the second one the force deflexion specifications.

Table 1. EUROSID measurement mean values for non crabbed and crabbed tests.

				Noncrabbed tests 50 km/h 90 degrees (Mean values of 3 tests)		Crabbed tests 54 km/h 63 degrees (Mean values of 2 tests)	
HEAD	Acc. 3ms (g)	RES.		39.9		34.7	
		HIC		265		160	
CHEST	Acc. 3ms (g)	RES		69.3		51.2	
		RIB	U	94.2	114.3	70.9	91.8
			M	139.9		107.3	
			L	108.7		91.1	
	Defl. (mm)	RIB	U	32.5	35.8	29.1	32.6
			M	36.5		31.3	
			L	38.5		37.4	
PELVIS	Acc. 3ms (g)	RES		79.9		70.3	
	Force (kN)	PUB		4.7		3.6	
		IL		1.0		0.7	

Geometrical characteristics

The EEVC specifications give a value of 250 mm to ground clearance of the front face of the barrier. This low value was kept in order to anticipate the evolution of car front structures design: it was expected a lowering of rigid front structures.

In fact even compared to the most recent cars, the EEVC mobile deformable barrier behaves differently: in car to car tests, the striking car overrides the door sill, even if it is in a diving braking position; in MDB to car tests, the door sill is clearly involved in the deformations, and then participates in the energy dissipation.

To give a more realistic behavior, the ground clearance of the MDB has to be increased to 300 mm.

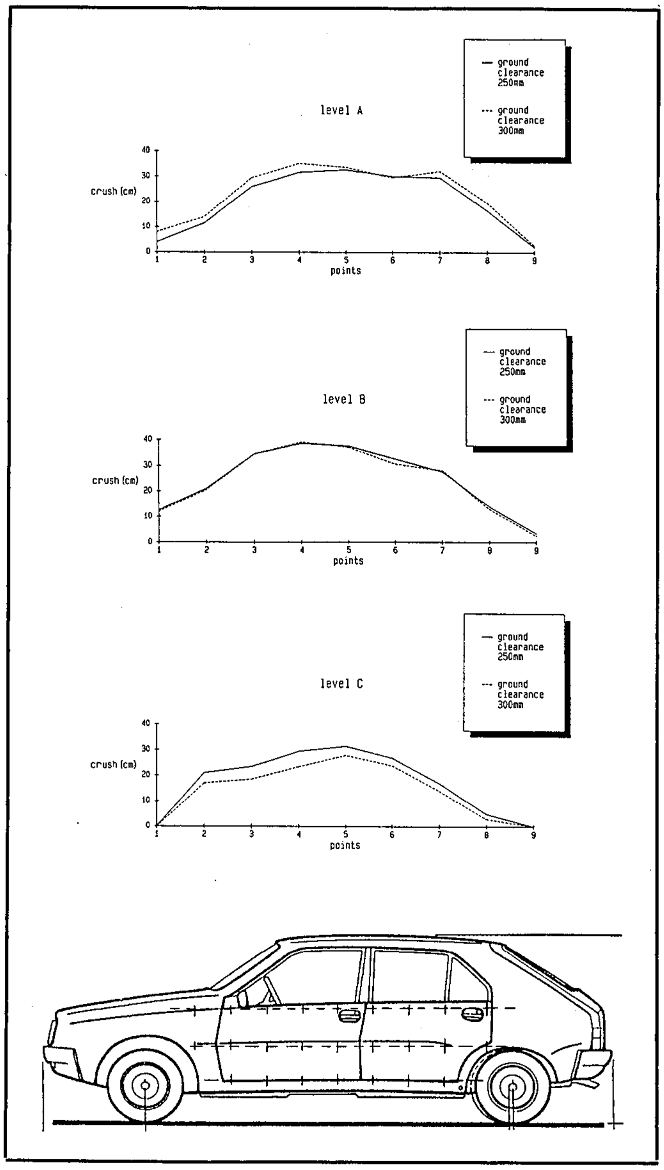

Figure 3. Side deformations in MDB tests with 250 mm, and 300 mm ground clearance and in car to car tests

However, increasing the ground clearance would also raise the upper edge and then load the side of the struck car at a level up to 800 mm, which may be clearly higher than what occurs in car to car accidents with the most recent cars having a low bonnet.

The value of 300 mm for the ground clearance was also the original proposal made by CCMC(3). Comparison of side deformations in MDB tests with 250 mm, and 300 mm ground clearance are indicated on Figure 3.

Stiffness specifications

The front face of the EEVC mobile deformable barrier is made from 6 blocks. The force deflection characteristics of each block and the total force deflection were specified as representative of a European car population. For each block, the force deflection variation must stay inside a specified corridor during a frontal impact test against a 90° rigid dynamometric wall at 35 km/h. impact speed.

In such a test made with the present designs of EEVC barrier front face, it has been found that it is impossible to stay inside the corridor during all the deformation.

A more or less important peak escaping from the corridor occurs at the beginning of the deformation. Figure 4 shows a typical total force deflection curve in a frontal impact test of the EEVC mobile deformable barrier. The values of injury parameters depends mainly from the behaviour of the barrier during its second half of deformation, and then it seems acceptable to consider that the corridor specifications would apply only to the deformations greater than 15 cm. It is then important to keep the specifications defining the energy to be dissipated by each block during the test.

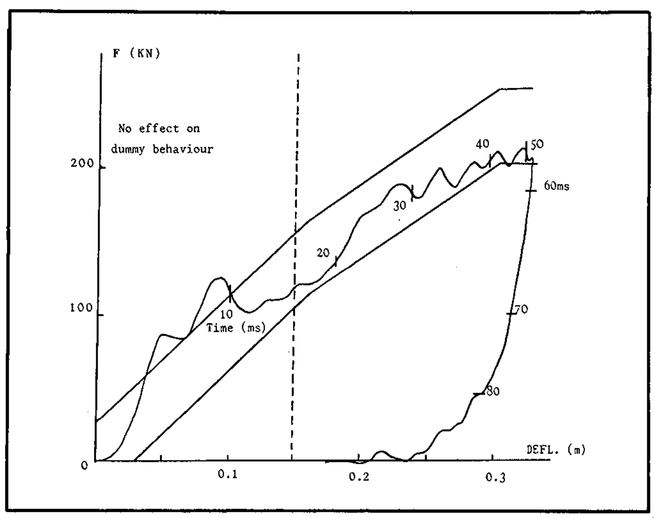

Figure 4. Typical total force deformation curve for EEVC mobile deformable barrier

The EUROSID

The second part of the procedure is the dummy to be used in the test. A comprehensive effort was made during the last years to design, manufacture and evaluate a new dummy, the EUROSID.

Eleven preproduction prototypes of this dummy have just been completed and will be tested soon in several laboratories around the world. This step is the last one, and the EUROSID prototypes were extensively tested in 1986 by five European laboratories and the results of this validation programme were reported during a seminar held in Brussels in December 1986.

Conclusions

The experience gained by several laboratories using the EEVC side impact test procedure allows to revise the specifications of this procedure.

Most of the parameters defining the procedure can be confirmed as they were originally described. The only possible modifications would concern the ground clearance which would be increased to get a more realistic interaction between the front of the barrier and side of the struck car, and the force deflection corridors which would not apply to the first 15 cm of barrier crush.

In the same time, a new dummy called EUROSID was developed especially to be used in side impact tests; the complete procedure is then available after being validated.

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A Study on Energy-Absorbing Units of MDB for Side Impact Test

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Abstract

Japan Automobile Research Institute, Inc. (JARI) and Japan Automobile Manufacturers Association, Inc. (JAMA) have been making continuous effort to study on the following themes since 1985 in order to examine the side impact test methods.

1. Analysis of side collision accidents in Japan:
   — Analyze the data collected on side collision accidents
2. Evaluation of energy-absorbing units of MDB:
   — Investigate average specifications of the Japanese domestic cars and the characteristics of energy-absorbing units made by JARI/JAMA
3. Examination of full scale test methods:
   — Review such factors for occupant injuries
4. Examination of component test methods:
   — Perform numerical analysis simulations
   — Study the various causes of occupant injuries
5. Comparison tests of side impact dummies:
   — Compare SID(NHTSA) dummy and EUROSID dummy

This report describes part of our research. It deals with the results of our development efforts of making energy-absorbing units of MDB based on the average dimensions and characteristics of the front end of Japanese domestic cars.

Specifically, this report explains in detail the development process, static and dynamic characteristics, dependency on impact speed and crab angle, etc., and also the results of full scale test using the energy-absorbing units.

Background

The problem of reduction of the death and injury of occupants in side collisions has become important internationally. Based on that recognition, JARI/JAMA commenced basic studies in 1985. However, many technical problems still remain unsolved and studies are continuing in 1987.

At present, JARI/JAMA is continuing its research into side impact test methods, both on full scale testing and component testing.

Table 1 shows items studied from 1985 through 1986.

This paper introduces the development of energy-absorbing units for MDB frontal structure, the testing of their characteristics, and a full scale test of MDB with the energy-absorbing units developed, which represent the completed portion of the 1985 and 1986 study items.

Results of a Survey to Determine MDB Energy-Absorbing Unit Specifications

To determine the various dimensions and characteristics of MDB energy-absorbing units, the various dimensions, weights, etc. of Japanese representative passenger cars were surveyed.

Dimensions and Weight

The average width and frontal height of Japanese cars are 1,550 mm and 450 mm, respectively, from the survey. The minimum ground clearance of Japanese car fronts is approximately 250 mm, which was calculated from the ground clearance of unloaded car, that of a loaded car and the degree of nose-dive during braking.

The median curb weight of Japanese cars, representing a cumulative percentage of 50%, derived from the number of cars in operation as of March 1984, is approximately 950 kg.

Front Stiffness

The average front stiffness was calculated from 16 representative passenger cars. As the result, it was found that the forces for deformations of 200 mm and 300 mm were 160 kN and 220 kN, respectively.

Table 1. Study items and schedule of side impact tests made by JARI/JAMA.

Item \ Year	1985	1986	1987
Analysis of side collision accident in Japan	Statistical analysis of side collision accidents that occurred in Japan		
Study of full scale test methods	Survey of average specifications of domestic cars.	Development of domestic MDB energy-absorbing units (Review of their characteristic testing and calibration methods.)	A full scale test to review factors that affect occupant injuries.
Study of component test methods		The extraction of parameters that affect occupant injuries and the study of the component tests based on the results of numerical analysis and full scale tests.	
Evaluations of side impact dummies			Performance comparison EUROSID dummy and SID(NHTSA) dummy.

Summary of MDB Specifications

Table 2 compares the results of a Japanese car survey conducted by JARI/JAMA with proposals by other nations. From the Table, it can be seen that the frontal dimensions of average Japanese cars are close to what is proposed by the EEVC[1] and the CCMC[2], and the front stiffness is close to the EEVC proposal.

From the above results, we were led to conclude that the values proposed by the EEVC would be effective Japanese MDB specifications in terms of both dimensions and front stiffness, and we developed MDB energy-absorbing unit according to these specifications. (The MDB energy-absorbing unit developed by JARI/JAMA is referred to as "J-absorber," hereafter.)

Development and Characteristics of J-Absorber

For the development of the J-absorber, we referred to those reports[3][4] of EEVC and CCMC energy-absorbing units in the 10th ESV International Conference (in 1985 in England) and we set the following goals.

(1) Fluctuations due to changes in the environmental conditions including temperature and humidity to be minimal. (2) Test fluctuations, that is, fluctuations in the deformation pattern be minimal. (3) The energy-absorbing unit itself is not to be torn to pieces and scattered by a impact as observed in the case of the energy-absorbing unit of the EEVC.

The developed J-absorber satisfied these conditions and was also inexpensive.

As shown in Figure 1, the J-absorber is primarily made of polyethylene foam, and the high rigidity block (Block 2) is coupled with rigid polyurethane foam (of different crush characteristics to the equivalent proposed by the EEVC) for reinforcement.

To join the members, an epoxy adhesive and a synthetic rubber adhesive were used.

Characteristics of J-absorber

(1) Static and Dynamic Characteristics

Figure 2 shows the static force-deformation curve of each J-absorber composite material. Each graph

Table 2. Comparison between MDB specifications and average japanese car.

Specifications \ Countries (Organizations)		Comparison of MDB specifications: NHTSA[5][13]	EEVC[1]	CCMC[2]	ISO SC10[7] draft	GRCS[8] draft	Average Japanese car
Weight (kg)		1,360	950	950	1,100 ±10	950	950 + occupant
Dimension	Width (mm)	1,679	1,500	1,550	1,550 ±20	1,500	1,550
	Height (mm)	559	500	490	500	500	450
	Ground clearance (mm)	280	300	300	260 ±10	250	250
Front stiffness (kN)	Up to 200 mm deformation	140	140∿190	200	190∿260	140∿190	160
	Up to 300 mm deformation	360	210∿260	480	480∿590	210∿260	220

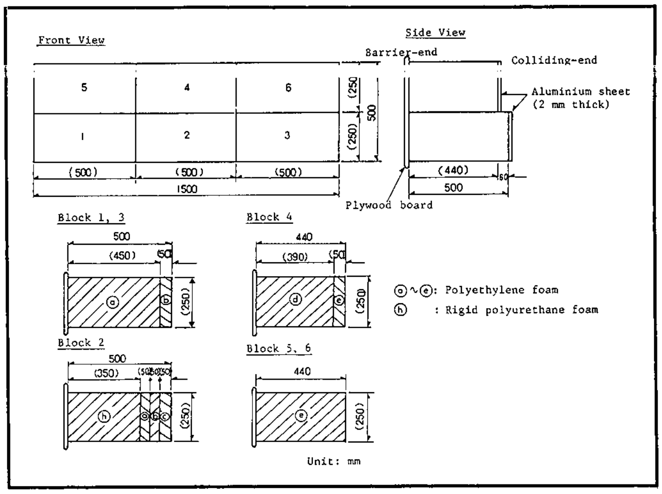


Figure 1. Composition and dimensions of J-absorber in each book

shows the result of static compression of a block measuring (L)500 x (W)500 x (H)250 mm made of each material.

Figure 3 shows static and dynamic characteristics of the J-absorber as a whole. The static characteristic was derived from the result of static compression at a static compression speed fo 50 mm/min, and the dynamic characteristic from the result of colliding an MDB with J-absorber weighing 1,100 kg against a fixed barrier at a speed of 35 km/h.

For reference, the characteristic zone proposed by the EEVC is also shown in Figure 3.

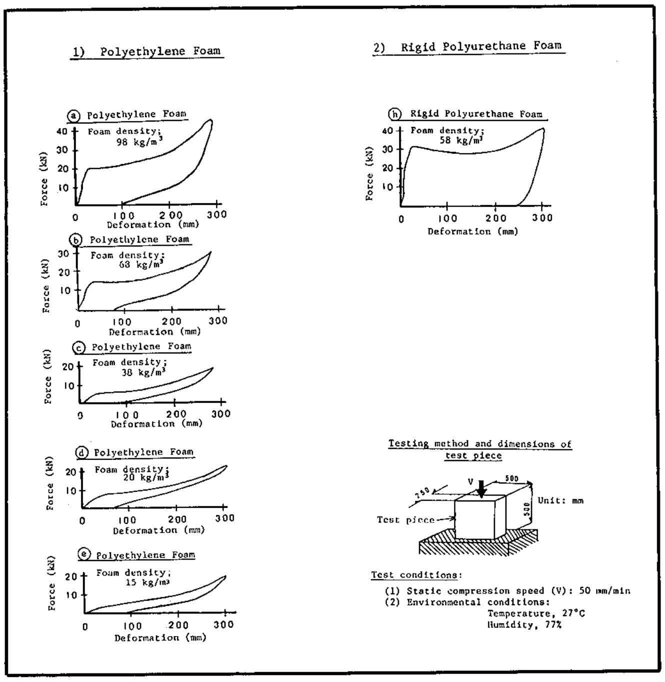


Figure 2. Static force-deformation curves of each material used for J-absorber

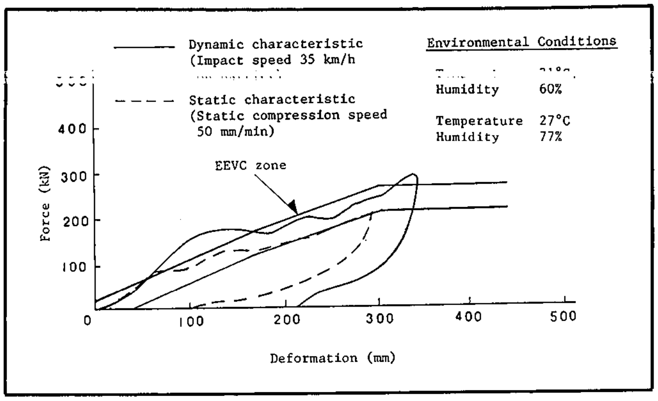


Figure 3. Dynamic and static characteristics of J-absorber

From Figure 3, it can be seen that the characteristics of the static force-deformation curve of the J-absorber as a whole are smaller than the dynamic one and that the J-absorber characteristics were nearly compatible with that of the EEVC, as intended.

(2) Influence of Impact Speed

Figure 4 shows the degree of the change in J-absorber characteristics caused by impact speed. To obtain the dynamic characteristics, 3 impact speeds of 35, 45 and 50 km/h were used.

From Figure 4, it is understood that the force-deformation curves of the J-absorber show the same characteristics within a range of impact speeds between 35 and 50 km/h, and that change due to speed is non-existent.

(3) Characteristics under with/without crab angle

Figure 5 shows the test method. As opposed to the impact speeds of 35 and 45 km/h used "without crab angle," respectively 39 and 50 km/h are used for tests with "27° crabbed" so that the speed component vertical to the fixed barrier can be equivalent.

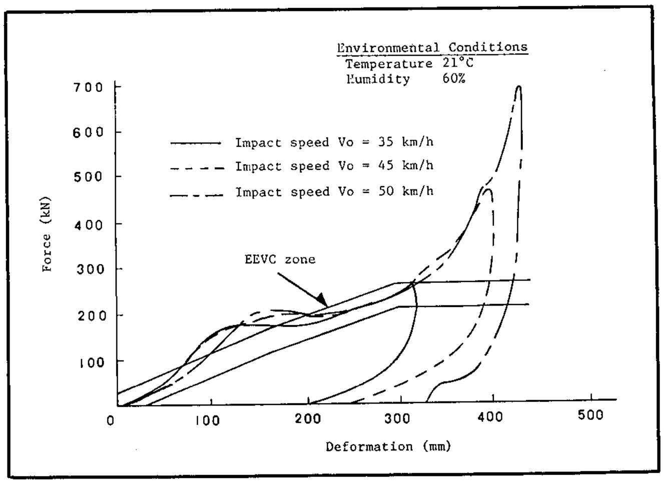
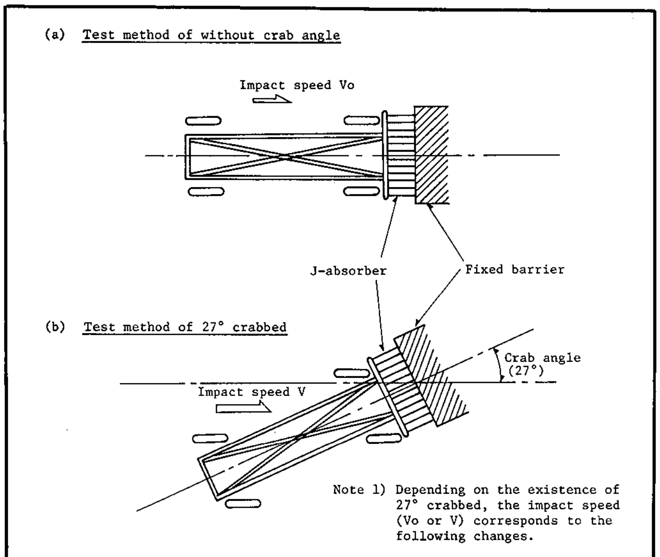


Figure 4. Comparison of characteristics under different impact speeds (J-absorber)



	Impact speed (Vo or V) km/h	
Without crab angle	Vo ±35	Vo = 45
27° crabbed	V = 39	V = 50

$V = \frac{Vo}{\cos 27°}$

Figure 5. Comparison of test methods with/without crab angle

Figure 6 shows the dynamic characteristics of J-absorber for "without" and "27° crabbed."

Figure 6-(a) shows the result of an impact speed 35 km/h of "without crab angle" and (b) shows 45 km/h of "without crab angle," respectively.

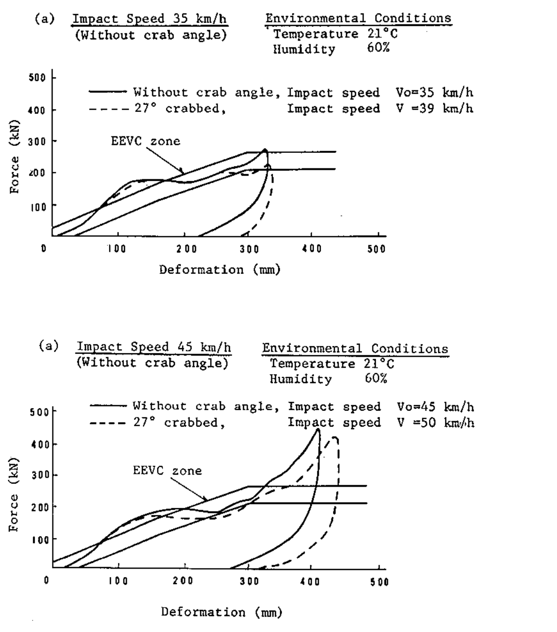


Figure 6. Comparison of J-absorber characteristics with/without crab angle

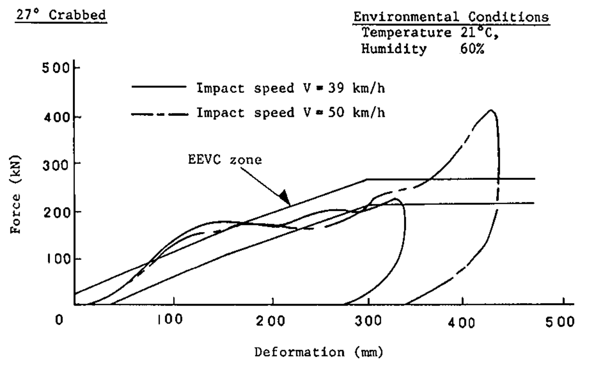


Figure 7. Comparison of (J-absorber) characteristics under 27° crabbed and under different impact speeds

Figure 6 shows that dynamic characteristics for "27° crabbed" are slightly lower than those for a "without crab angle" condition. However, the degree of change is nominal, and in the case of the J-absorber, characteristics remain the same without respect to the "without" or "27°" of crab angle. Figure 7 recapitulates the dynamic characteristics at 39 and 50 km/h speeds for "27° crabbed". From Figure 7, it was found that the characteristics for "27° crabbed" are changed little by speed and it was confirmed that the characteristics were unaffected by speed irrespective of the crab angle.

(4) Influence of Temperature and Humidity

Figure 8 shows the dynamic characteristics under different temperatures and relative humidity levels. The impact speed of the MDB is 35 km/h. Though the figure shows a slight difference in the characteristics of the J-absorber as a whole depending on environmental conditions, including temperature and humidity, the degree of the change is very small and it is understood that the dynamic characteristics of the J-absorber will have no problem in side impact testing within a temperature range of between 12 and 31° C, and humidity range of between 30 and 60%.

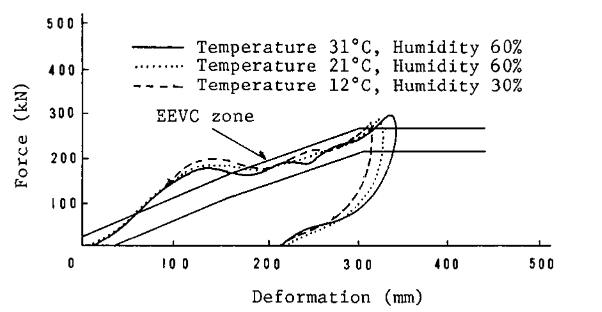


Figure 8. J-absorber characteristics (under different environmental conditions)

Table 3. Items and summaries of the basic tests implemented.

	Basic test item	Summary	Number of test for each test piece (N)	Environmental conditions
1	Stability tests (Tests to define standards)	Identification of data fluctuations in each test made under the same conditions; used as standards of data in the tests below.	10	Temperature, 18 to 22°C Humidity, 50 to 60%
2	Temperature influence tests	Identification of test piece data fluctuations due to different temperatures (5°C and 35°C).	3	Temperature, 20°C Humidity, 58%
3	Force direction influence tests	Identification of data fluctuations in different force directions (18°, 27°).	3	Temperature, 19°C Humidity, 52%
4	Foam direction influence tests	Identification of data fluctuations due to force directions differing from the foam direction.	3	Temperature, 21°C Humidity, 55%

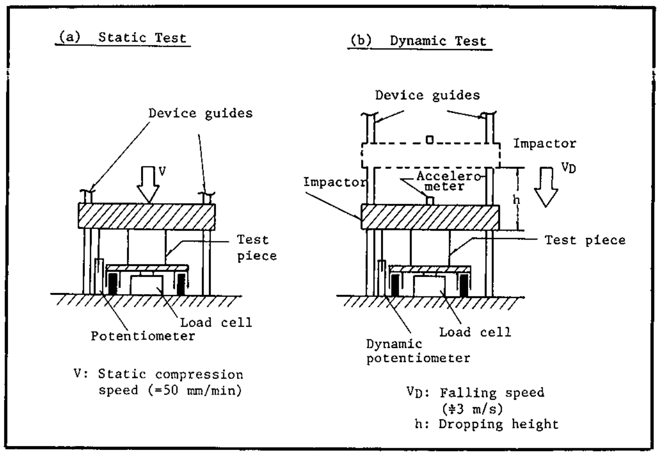


Figure 9. Test method (test-piece test)

Characteristic Test Using Test Pieces

As mentioned, the J-absorber as a whole features excellent characteristics. They are obtained as a result of an in-depth study of the materials for their reliability and the influence due to environmental changes through material tests made before its development. Here, the basic characteristics of the material tests are introduced. A basic test of characteristics, of the contents of which are listed in Table 3, is made by using each test piece ((L)100 x (W)100 x (H)100 mm) of the representative materials used to make the J-absorber.

Three different materials were used in this test; (A) polyethylene foam, (B) rigid polyurethane foam and (C) rigid polyurethane foam made according to the method introduced in an EEVC report[3] (EC-spec, rigid polyurethane foam, hereafter). Test conditions used as the standards for each test piece are as listed in Table 4.

Static and dynamic tests are made by the test methods shown in Figure 9 and force valued corresponding to test piece deformation of 20 mm, that is,

Table 4. Test conditions used as the standards for each test piece.

	A. Polyethylene foam	B. Rigid polyurethane foam	C. EC-spec. rigid polyurethane foam
Dimensions (W x H x L) mm	100 x 100 x 100	100 x 100 x 100	100 x 100 x 100
Features	One of the J-absorber materials shown in Figure 2 as a (a) polyethylene foam Foam density: 98 kg/m³	One of the J-absorber materials shown in Figure 2 as a (h) rigid polyurethane foam Foam density: 58 kg/m³	Made in accordance with the EEVC manufacturing method Foam density: 50 kg/m³
Range of test piece temperature change during test	18 ∿ 22°C	⟸	⟸
Direction of force to foam direction	Force applied at right angles to the foam direction	⟸	⟸
Number of days after manufacturing	3 to 7 days	⟸	⟸
Other conditions	Test pieces were not allowed to absorb moisture or left outdoors after manufacturing	⟸	⟸

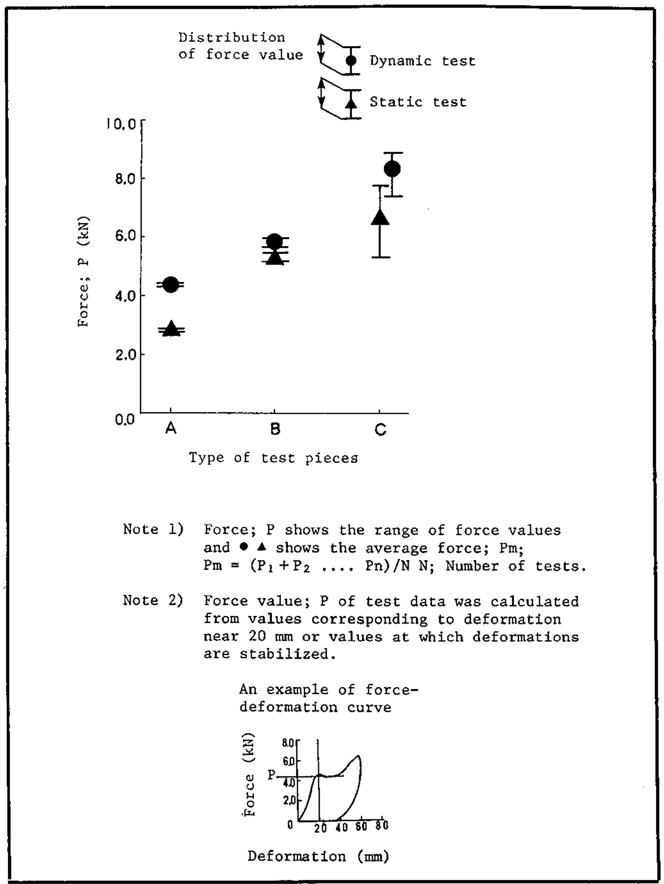


Figure 10. Results of stability tests [N = 10, (each test piece)]

force values at which forces become constant to deformations shown in Figure 10, are compared.

(1) Stability of Crushing Characteristics

Figure 10 shows the test results of the three test pieces tested under the same conditions. Figure 10 shows the distribution of force (P) with test piece deformation near of 20 mm, in static and dynamic tests made by using 10 test pieces each. According to this test, it was understood that the distribution of force value are small in (A) polyethylene foam and (B) rigid polyurethane foam, which are J-absorber materials.

(2) Temperature Characteristics

Figure 11 shows the degree of change of 3 different test pieces to different temperatures and is derived from the results of static test. The graph shows that the higher the temperature, the smaller the force tends to be. The degree of change is the smallest in the case of the rigid polyurethane foam used to make the J-absorber.

(3) Influence of Differences in Force Direction

Figure 12 compares force value changes to different load directions for three different test pieces. The test is made by the method shown in Figure 12 (angle of force direction θ is changed) for comparison with results obtained by the method shown in Figure 9 (angle of force direction θ is zero). The measured values of deformation and force were corrected by the angle of force direction for conversion into longitudinal direction components.

Figure 12 shows that the larger the angle of force direction ( ), the lower the test piece rigidity tends to be. As a result, it is found that the polyethylene foam and rigid polyurethane foam used to make the J-absorber featured little change in their characteristics to the changes of the force direction.

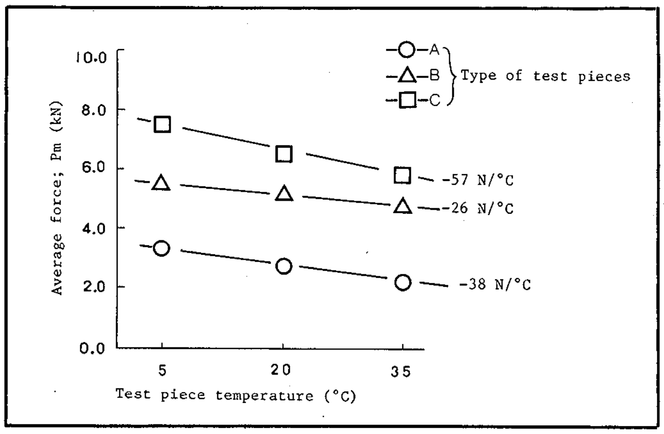


Figure 11. Results of temperature characteristic tests (Number of tests: Temperature of 20° C; N = 10 temperature of 5° C and 35° C; N = 3)

(4) Influence of Foam Direction

Figure 13 shows results of a static test to study the degrees of change in the force value by loading three different test pieces parallel and normal to the foam direction.

The figure shows that the degrees of change in the case of the force being applied parallel to the foam direction are approximately 10%, 20% and 14% larger for (A) polyethylene foam, (B) rigid polyurethane foam and (C) EC-spec. rigid polyurethane foam, respectively, than in the case of force application at right angles to the foam direction. Consequently, it was found necessary to pay sufficient attention to this foam direction when energy-absorbing materials are manufactured or used.

Summary of Development and Characteristics of J-absorber

The J-absorber developed by JARI/JAMA has the following features:

(1) Changes in the data of tests made under the same conditions are minimal.

(2) Changes in the data due to different speeds are minimal.

(3) Changes in the data due to different angles

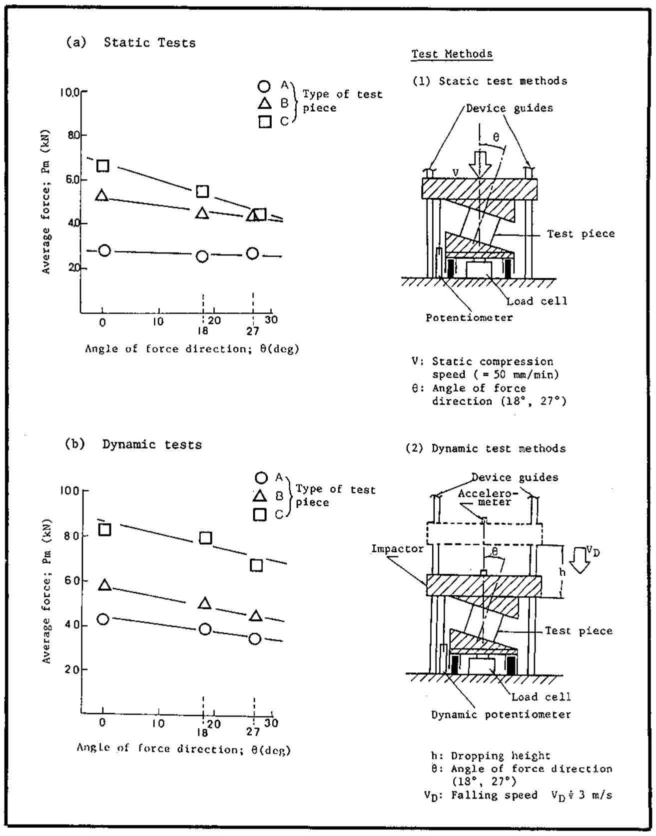


Figure 12. Results of force direction Influence test (Number of tests: θ = 0°; N = 10, = 18° and 27°; N = 3)

of force direction (with crabbed) are minimal.

(4) Its features vary slightly, depending on temperature changes and foam direction. Within a temperature range (of 12 to 31° C) and a humidity range of 30 to 60%), however, changes in the data are minimal.

Generally, the features of an energy-absorbing material are required to include low cost, easy handling, sufficient availability, good shaping performance, excellent stability of quality (as small fluctuations of characteristics as possible), resistance to environmental conditions and storability for a long time. It was confirmed that the J-absorber was inexpensive, easy to handle, readily available, easy to shape and showed excellent stability. As the J-absorber is made of plastic foam, it is, however possible that its features will be affected slightly, depending on the method of long-term storage and environmental conditions.

On the other hand, the energy-absorbing unit using aluminum honeycomb presents good features in connection with its storage method and environmental conditions including temperature, humidity and aging. But it is expensive and has some problems related to its handling, including the necessity of precrushing and shaping. Table 5 compares the features of energy-absorbing units developed abroad.

Energy-absorbing units developed by foreign countries have both merits and demerits, and the types of absorber required will have to be further discussed, along with calibration methods, in addition to conditions of side impact tests.

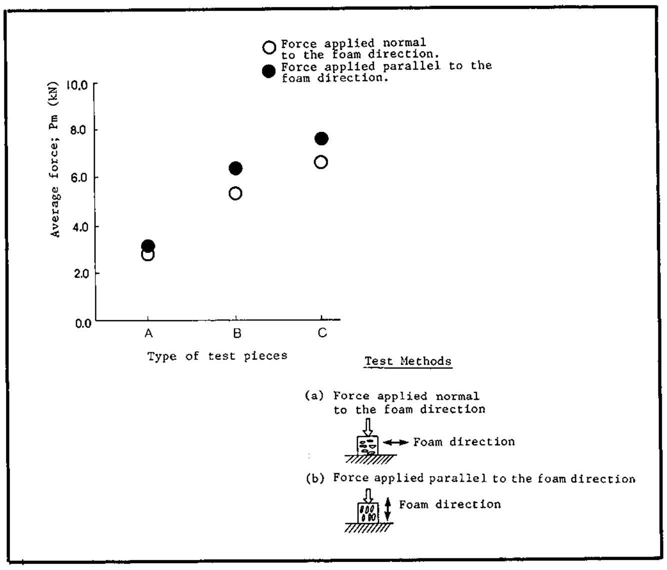


Figure 13. Results of the foam direction influence test (Number of tests: case of force applied normal to the foam direction; N = 10 case of force applied parallel to the foaming direction; N = 3)

Full-Scale Test

Using the J-absorber developed by JARI/JAMA, a total of 5 full-scale tests were studied;

(1) behaviour of the J-absorber used in full-scale tests,

(2) degrees of deformation of test vehicles.

Test Conditions

Impact Speed and Angle

Typical impact speeds and angles were assumed as follows from 60 cases of passenger car side collision accidents in Japan in which the occupants of the struck cars were injured, and based on this information, impact patterns were determined.

Striking car impact speed: 40 km/h

Struck car impact speed: 20 km/h

Impact angle: 90 degrees (angle between positional direction of both cars)

Test Vehicles

(1) Striking Car (MDB was used in the test)

The MDB weighed 1,100 kg, which is a total of 950 kg, the curb weight representing 50% of the cumulative percentage derived from the number of cars in the operation mentioned already, the body weight of occupant and some cargos.

(2) Struck Car

A subcompact passenger car (4-door sedan) in the weight class 1 ton found in Japan was used as the struck car.

Impact Position

The impact position was designed to be such that the left front end of the MDB hits the struck car on its left front side 610 mm from the wheelbase center,

Table 5. Comparison of characteristics of energy-absorbing units developed abroad.

Type of energy-absorber / Characteristics	Plastic material		Alumi-honeycomb material
	J-absorber	EEVC(3) CCMC(4) absorber	NHTSA(5) UTAC(6) absorber
Handling	◎	○	○
Shapeability	◎	◎	△
Stability of quality	◎	○	◎
Storability	△	△	◎
Sensitivity to environment	○	○	◎
Cost	◎	◎	△
Availability	◎	△	△

◎	Best
○	Better
△	Good

Note) Ratings of ◎,○,△. of availability and cost were made by JARI/JAMA.

in reference to ISO/TC22/SC10/GT1[7] document N95 (1985), as typical road accident analysis results were not available in Japan. The minimum ground clearance of the MDB front was designed to be 250 mm.

Impact Patterns

The impact patterns tested are variations of a 90° side impact on a car at rest as shown in Figure 14. The differences in the impact condition in each impact pattern are shown below.

(1) Pattern A (A pattern based on road accident analysis results; test 27° crabbed)

This takes the speed factor of the struck car (1/2 of the striking car's speed) into consideration. In pattern A the impact speed of the MDB is 45 km/h, the 27° crabbed, the minimum frontal ground clearance 250 mm and the J-absorber is used; and 2 tests were made under the same conditions. Pattern A is based on road accident analysis results, and is used as the standard for the other impact patterns.

(2) Pattern B (Test without crab angle—No.1)

For Pattern B, the struck car was stationary and the impact speed of the MDB was assumed to be 40 km/h and without crab angle, and all other conditions remained the same as in Pattern A. Two tests were made under the same conditions.

(3) Pattern C (Test without crab angle—No.2)

For Pattern C, the minimum frontal ground clearance of the MDB was changed to 300 mm, 50 mm higher than in the others, and the other conditions left the same as in Pattern B. One test was made under this set of conditions.

Test Results

The results of a full-scale test made at this time were assessed from the viewpoints of; (1) behavior of the J-absorber in full-scale tests, (2) deformation degrees of vehicles used in the test, etc.

Fluctuations of impact speed and position to targeted impact conditions in the full-scale test is shown in Table 6.

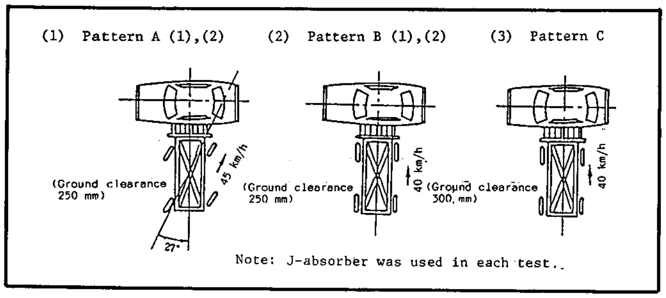


Figure 14. Full-scale test impact patterns

(1) Impact Speed

Fluctuation of the actual impact speed was within 1% of the targeted impact speed.

(2) Impact Position

The deviation range of the actual impact position from the targeted impact position was ±30 mm in the longitudinal direction and ±10 mm in the lateral direction.

Degree of Deformation of the Test Vehicles

(1) Behavior of J-absorber During Impact

As a result of the analysis of a high-speed film, it was discovered that such phenomena as separation and peel-off of the J-absorber were not observed in any block or in the absorber as a whole during the impact, and it worked effectively.

Here, the deformation of the J-absorber upper surface was derived from an analysis of a high-speed film and the deformation in each impact pattern was compared with each other for display in Figure 15. Figure 15 (a) compares Pattern A (1) with (2), (b) Pattern B (1) with (2) and (c) Pattern A with B and C.

From Figure 15, it was confirmed that the repeatability of the J-absorber under the same test conditions was excellent and it could be used effectively as an energy-absorbing material in a full-scale test.

(2) Degrees of Deformation and Intrusion into the Struck Car

Figure 16 compares deformation patterns of outside the struck car by impact patterns. Figure (a) compares deformation patterns within Pattern A, Figure (b)

Table 6. On the accuracy of test conditions in full-scale test.

Test No.	Impact pattern	Type of vehicle	Weight of vehicle (kg)	Target impact speed (km/h)	Actual impact speed (km/h)	Deviation of test position (mm)	
						Front-rear	Up-down
1	Pattern A (1)	MDB	1,100	45	44.7	30	0
		Struck car	1,200	0	0		
2	Pattern A (2)	MDB	1,100	45	44.6	30	5
		Struck car	1,195	0	0		
3	Pattern B (1)	MDB	1,100	40	40.2	-10	5
		Struck car	1,220	0	0		
4	Pattern B (2)	MDB	1,100	40	39.9	-30	0
		Struck car	1,235	0	0		
5	Pattern C	MDB	1,100	40	40.1	15	-10
		Struck car	1,215	0	0		

Note 1) (+) deviation of impact position in the vertical direction shows an upward deviation from the minimum ground clearance of MDB of 250 mm and a (-) downward deviation.

Note 2) (+) deviation of impact position in the longitudinal direction shows a forward deviation of the struck car and a (-) backward deviation.

within Pattern B, and Figure (c) between Patterns A, B and C. From Figure 16, it was understood, as in the case of deformation patterns of the J absorber, that the deformation patterns of the struck car observed under the same conditions demonstrated excellent repeatability. Table 7 compares the maximum permanent deformations and intrusion into the car and the positions of their occurrence, etc. by impact pattern. This indicated that the occurrence position of the maximum and permanent deformation and intrusion into the car varied a little depending on the pattern of the impact.

Next, Figure 17 compares the maximum permanent deformations of the car, the maximum permanent intrusion into the car interior and the damage length of the struck car (maximum longitudinal length of a deformed portion on the side of the struck car; see

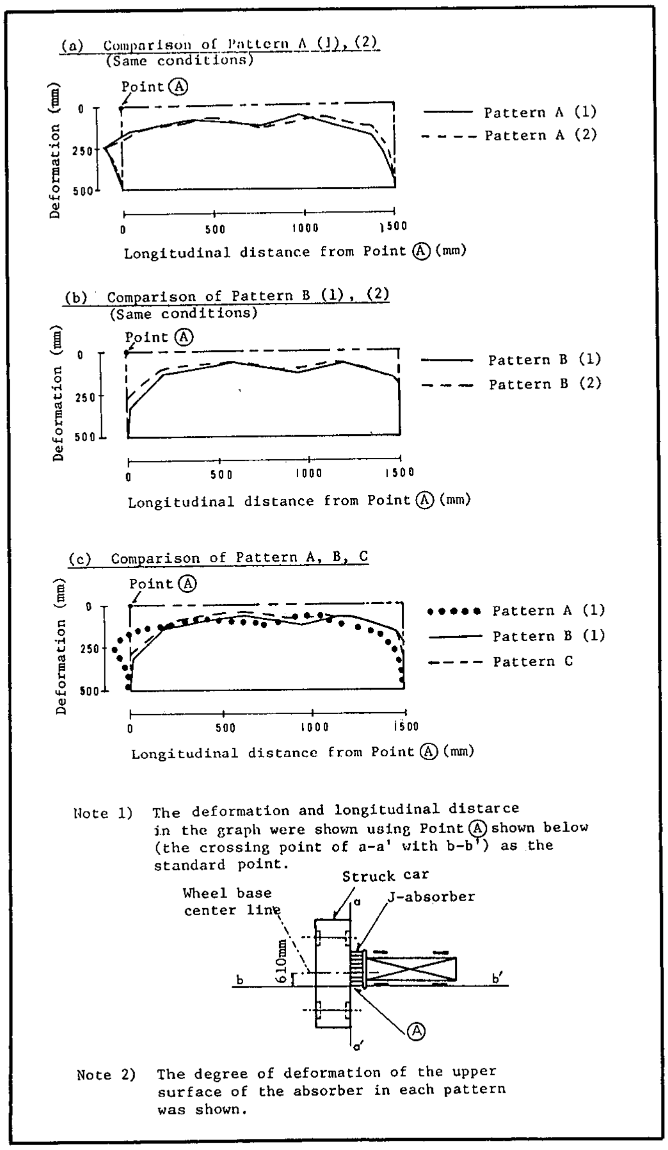


Figure 15. Degree of deformation of the J-absorber in a collision (In the maximum deformation)

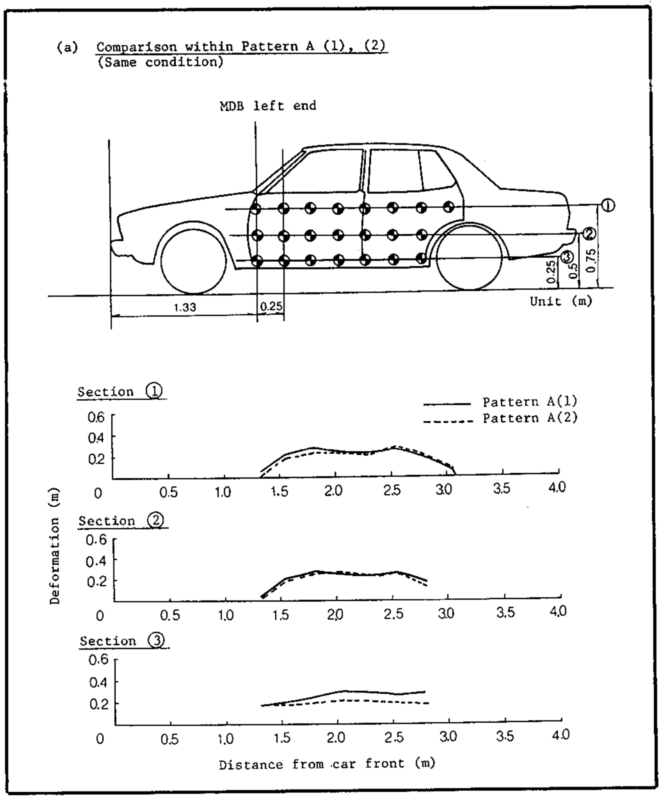


Figure 16 (1). Deformation of the outside of the car in each impact pattern

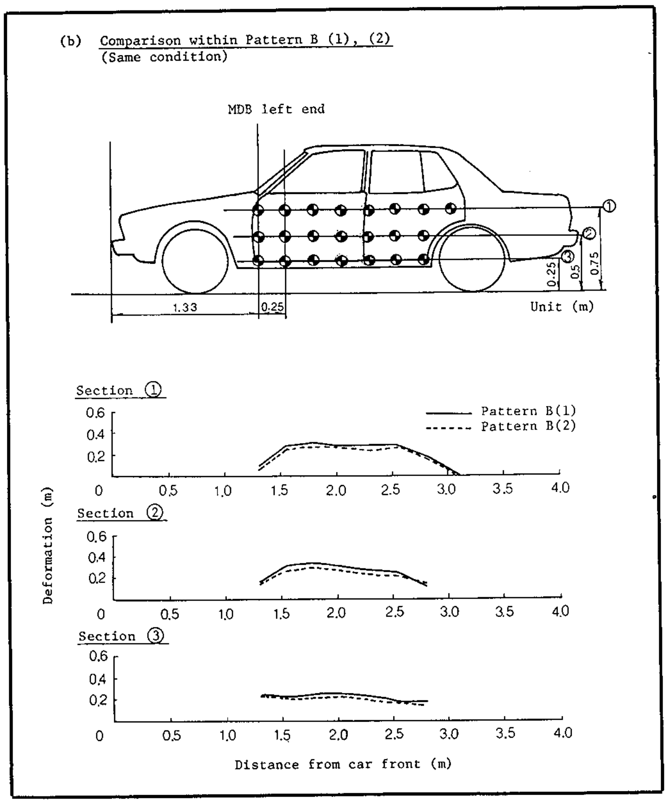


Figure 16 (2). Deformation of the outside of the car in each impact pattern

Table 7) in each impact pattern, in relative figures proportional to value in Pattern A (1) as a unit value. (see Figure 14)

The following conclusions can be derived from Figure 17 through the comparison of each impact pattern, although an in-depth review cannot be made because of limited number of tests.

1) Degree of Maximum Permanent Deformation and Intrusion of the Struck Car
   a) The maximum permanent deformation and intrusion of the struck car without crab angle was approximately 5% larger than that with 27° crabbed.
   b) The degree of deformation with a larger ground clearance was approximately 10% larger than that with a smaller ground clearance.
   c) The difference is deformation between the two tests made under the same conditions was approximately 5%.
2) Damage Length on the Side of the Struck Car

The damage length on the side of the struck car (see Table 7) was similar in each impact pattern.

It was anticipated that the front end of the MDB would slide along the side of the struck car during tests with 27° crabbed, but test results revealed no difference among any impact pattern.

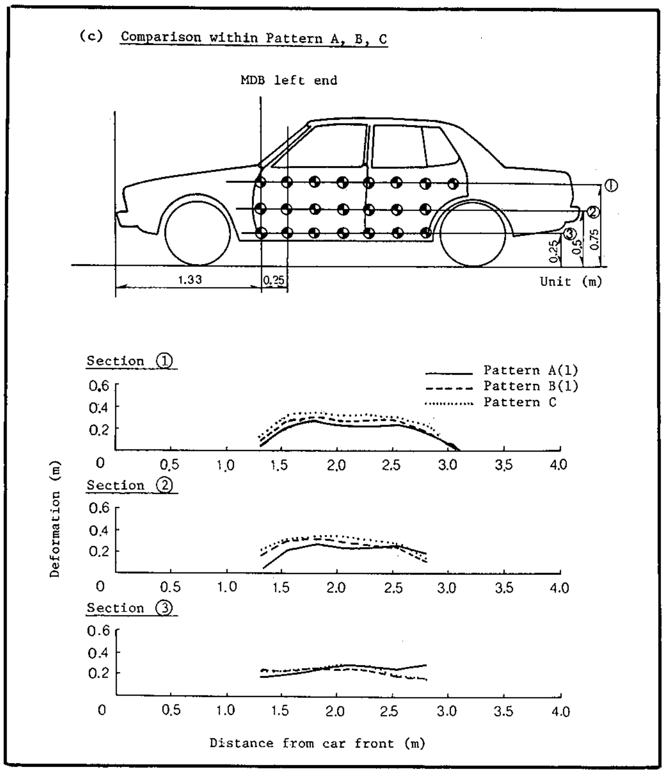


Figure 16 (3). Deformation of the outside of the car in each impact pattern

Table 7. Permanent deformation and intrusion of the struck car and its occurrence position.

Test Number	Impact pattern	Maximum permanent deformation (m)	Occurrence position of maximum permanent deformation (m) (Note 1)	Occurrence place of maximum permanent deformation	Maximum permanent intrusion (m)	Occurrence position of maximum permanent intrusion (m) (Note 1)	Occurrence place of maximum permanent intrusion	Length of damage (Note 2)
1	Pattern A (1)	0.31	2.40	Near the front of rear door	0.20	2.14	Near the under of B pillar	1.76
2	Pattern A (2)	0.30	2.40	Near the front of rear door	0.21	2.14	Near the under of B piller	1.75
3	Pattern B (1)	0.32	1.84	Near the center of front door	0.22	2.14	Near the under of B pillar	1.75
4	Pattern B (2)	0.31	1.94	Near the rear of front door	0.21	2.14	Near the under of B piller	1.78
5	Pattern C	0.35	1.86	Near the center of front door	0.24	2.16	Near the center of B piller	1.74

Note 1) The occurrence positions of the maximum permanent deformation and intrusion are longitudinal distance from the front end of the struck car.

Note 2) The length of damage refers to the maximum length of a deformed section on the side of the struck car. (Shown in terms of the maximum displacements in vehicle longitudinal direction of deformed sections measured at a height of 750 mm in Figure 16.)

(3) Depth of Intrusion at the Position of a Seated Occupant

Figure 18 shows the depth of permanent intrusion into the car at the chest and the pelvis level of an occupant on the front seat (measurement values at points C and P in Figure 18) in each impact pattern relative to the values of Pattern A (1) for comparison.

1) At Chest Position (Point C in Figure 18)
   a) The intrusion without crab angle was approximately 60% larger than with 27° crabbed.
   b) The intrusion with larger ground clearance was approximately 100% larger than that with a smaller clearance.

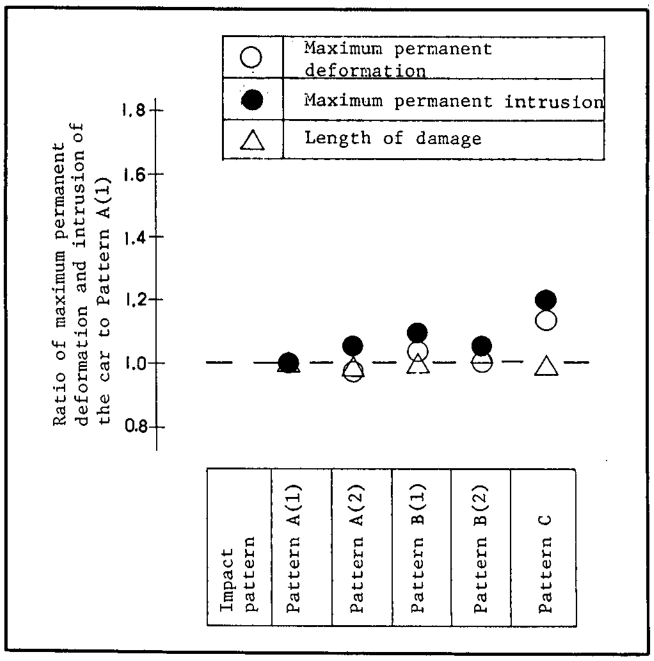


Figure 17. Comparison of maximum permanent deformation and intrusion of the struck car

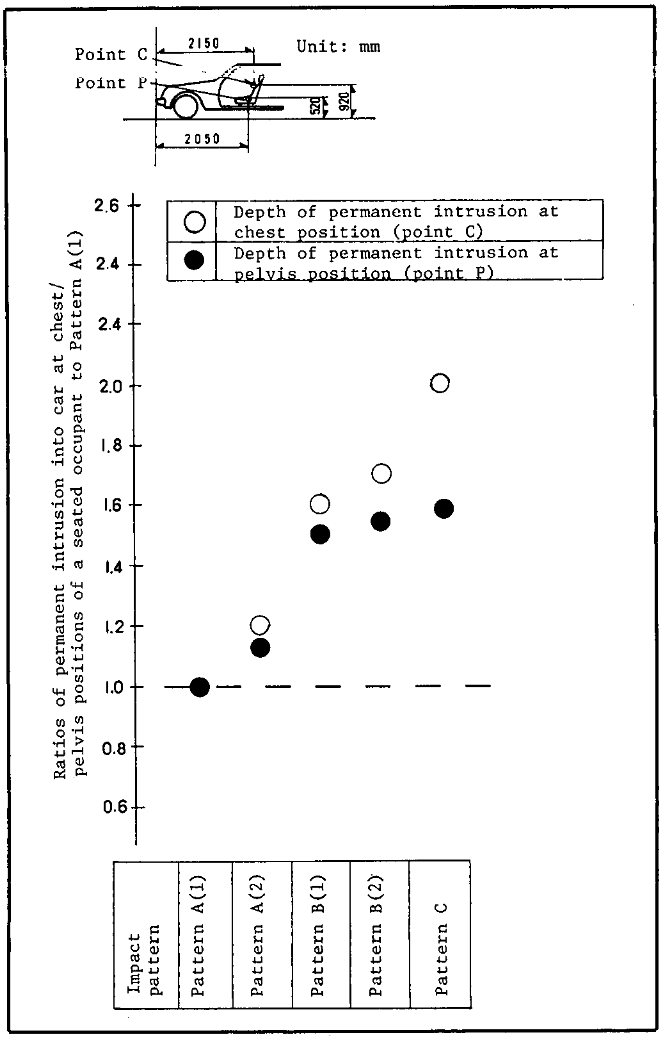


Figure 18. Comparison of permanent intrusion into car at chest and pelvis positions of a front seated occupant

c) The difference in intrusion depths in two tests made under the same conditions was approximately 20%.

2) At Pelvis Position (Point P in Figure 18)

a) The intrusion in the test without crab angle was approximately 50% larger than that with 27° crabbed.

b) The depth with larger ground clearance was approximately 60% larger than that with a smaller clearance.

c) The difference in intrusion depths in the two tests made under the same conditions was approximately 15%.

Summary of Full-Scale Test Results

We conducted 5 full-scale tests in all by using the J-absorber developed by JARI/JAMA to study (1) behavior of the J-absorber when it is used in the full-scale tests, and (2) the degrees of deformation on the test vehicles. The following can be concluded from the results of this study, though detailed investigations were unavailable from the limited number of tests made at this time.

(1) Behavior of the J-absorber Used in Full-Scale Tests

Separation and peel-off of the J-absorber were not observed and its repeatability of deformation under the same test conditions were also favorable.

(2) Degrees of Deformation on the Test Vehicles

1 It was discovered that the degree of maximum permanent deformation and intrusion into the struck car fluctuated little if tests were made under the same conditions. It was also found that the values varied widely depending on differences in impact patterns (changes in the minimum ground clearance, for example).

2 It was revealed that the depth of deformation and intrusion into the car without crab angle was larger than that with 27° crabbed.

3 It was found that the fluctuation margin of the degree of intrusion into the car on the level of an occupant, grew wider as the deformation pattern in the car was changed by slight differences in the test conditions, not to speak of differences in the test pattern. This is considered to mean that it is likely to affect the degree of occupant injury as well. This will have to be further studied in parallel with the analysis of factors that affect the degree of occupant injury, that is, the injury criterion, etc.

Conclusions

This paper is intended to report on the development of an MDB energy-absorbing material which was included in the subjects of studies that have been implemented by JARI/JAMA since 1985, and results of the full-scale MDB tests made using the material. This paper represents the first stage of an approach to establish side impact test method.

As far as JARI/JAMA is concerned, it has also started reviewing "component test methods" at this point of time as mentioned earlier in addition to "full-scale test methods," that is, side impact test methods using MDB.

As for component test methods, further study will be conducted as they are considered highly effective in independently evaluating causes of injury if their advantages of amply repeatability and possibilities of simplification can be utilized, while it is very difficult to reproduce a full-scale test.

As for full-scale test methods, fluctuations of the depth of intrusion into the car interior on the level of occupant, etc., are larger, as is known in this study, from tests made under the same conditions, and their

influence on the degree of occupant injury is anticipated. Therefore, conclusions will have to be drawn after gaining a full understanding of the various factors conceivable in an actual road accident, such as
(1) Impact pattern,
(2) Vehicle weight,
(3) Frontal stiffness and shape,
(4) Relative speed and collision angle,
(5) Impact positions,
(6) Bumper heights, etc.

At present, (1) the development of a side impact dummy as a tool for evaluation and (2) the development of a simplified dummy used in the car interior impact test are being promoted by research institutions of various countries to establish a test method for side impact. Because of this, the study of passenger injury criteria, as part of evaluation standard, will have to be furthered, and constitutes one of the most important items for study.

Although the method of side impact testing is being discussed and reviewed independently by various government and research organizations in various nations, it is important that efforts be made to develop an internationally harmonized test method, and as far as JARI/JAMA is concerned, we will continue to make technical exchanges over this test method by disclosing our study findings.

JARI/JAMA is prepared to continue with its study in the future in accordance with our study schedule and in consideration of the above.

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Movable Deformable EEVC Barrier for Side Impact

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Abstract

Since 1982 the Federal Highway Research Institute was engaged to give considerable effort in the development and validation of the EEVC barrier face and has conducted about 80 full-scale tests according to the EEVC proposal for side impact upon which is given this synopsis. The tests include parameters like:

- a non-deformable face according to SAE J 972a versus deformable faces according to EEVC and CCMC
- barrier/car tests versus car/car tests (including dived braking position of striking car), car stationary

- mass variation of barrier (ranging from 950 up to 1100 kg)
- tests with two moving cars (collision angle 90 degrees)
- crabbed mode of barrier (collision angle = impact angle = 90° versus collision angle 90°/impact angle 63°)
- all tests with subcompact cars (VW Golf I, Type 17), intermediate cars (Daimler Benz W 123 series) and Hybrid II dummies because no side impact dummies were available at that time
- calibration tests against a dynamometric wall.

The main conclusions which can be drawn from this test experience are as follows if one takes car/car tests as basis of comparison:

- Rigid barrier faces have proved to be unrealistic for several aspects. The force/deflection characteristic of the EEVC face is suitable for European (and regarding other publications, also for Japanese) passenger cars and can be met, e.g., by foam elements.
- The collision configuration of the EEVC proposal is valid; that means collision angle = impact angle = 90°. This opinion is based on the following experiences:
  - The crabbed barrier does not result in an improved simulation of deformation.
  - The load of the driver dummy ranks from high to low in the following way:
    a) two moving cars, 90°
    b) car/car, 90°, struck car stationary
    c) deformable CCMC barrier, 90°
    d) deformable EEVC barrier, 90°
    e) crabbed barrier, EEVC face
    That means the crabbed mode represents (for these tested car types) the lowest and probably unrealistic loading. For d) and e) it is the same sequence with EUROSID dummy[10].
  - The crabbed mode is slightly less reproducible and needs greater effort.
- Mass variation of the barrier ranging from 950 up to 1100 kg has a negligible influence; therefore any mass in this range can be accepted at least for European cars.
- The only parameter which had to be changed is the ground clearance (250 mm) of the deformable face. In most of our car/car tests we observed an override of the door sill even in the dived braking position of the striking car. This could not be reconstructed by the EEVC barrier with the ground clearance of 250 mm. That means for the barrier test an unrealistically high energy dissipation for the door sill and insufficient deformation in the upper weaker door region. Therefore it was inevitable to increase the ground clearance to a value of 300 mm. This value was also the former specification of the European CCMC-barrier face and it is taken into account within a regulation draft for testing today's cars.
- Calibration problems are thought to be solved.

In conclusion the EEVC barrier and test configuration seems to be a suitable test arrangement for type approval procedure.

Introduction

In the field of passive car safety, the development of a regulation for Europe concerning the protection during a side collision has made progress.

Since 1982 the Federal Highway Research Institute (BASt) has been involved in this work and has conducted more than 80 full-scale tests on the protection of private car passengers in these lateral collisions. The work was based on the output of Working Group 6 of the Governmental European Experimental Vehicles Committee (EEVC). In addition to private cars, various barrier forms have been used as striking vehicles in a number of test series with various objectives.

The BASt was also engaged to give a considerable effort in the development and validation of the EEVC barrier face, and tests with the new European side impact dummy EUROSID were also conducted. The synopsis given here reports the main results upon the most important test parameters from the European point of view. Results upon tests with the EUROSID are reported separately in this conference[10].

Efforts in Europe and the USA to improve the protection of private car passengers in lateral collisions have led to the separate development of lateral collision test procedures. The European solutions EEVC (from European governments) and also CCMC (from the European car manufacturers) are very similar with respect to the test tool, test parameters and evaluation. In EEC/ERGA safety draft S 65 the proposal of EEVC is integrated. The American proposal not only differs from the European proposal with respect to the design of the test tool (mass, stiffness, ground clearance, impact point and width of the barrier front), but it also considers the velocity on the struck vehicle during the collision. In an effort to keep down the number of tests in the type approval test procedure for new private cars, a proposal for harmonisation which contains elements of both the European and American solutions has been elaborated and put up for discussion by the European car industry[1,2]. This is the crabbed CCMC barrier, that means a barrier with impact point and impact angle (crabbed) from the US proposal but with the CCMC deformable face and a mass between the US and European value (in the same sense the later, so-called EEVC crabbed barrier should be understood).

Barriers and test range

Calibration tests, comparative tests with a rigid barrier, two deformable barriers and also car/car side collisions (including tests with two moving cars) were carried out with the object of allowing the assessment of a suitable barrier form as a side impact standard (see Fig. 1). Comparisons concentrated on vehicle deformations and the most significant, or most frequent, occupant loading values measured with different barrier forms and masses as well as in the car/car crashes. The tests were carried out at the crash test facility of the Federal Highway Research Institute[3]. For each test configuration at least two tests were carried out under the same conditions.

The mobile barriers which were used in these tests were all based on the same chassis according to SAE J 972a as well as ISO/DIN standards. They were modified in the following way (in the sequence as they were available):

- rigid face according to SAE
- deformable face according to CCMC[7]
- deformable face according to EEVC (Fig. 2[8,9])
- crabbed mode with EEVC face.

Data on crashes with a *rigid mobile barrier* could be taken from earlier projects dealing with the problems of the interaction of car passengers in a side collision[4,5]. Some of the results of the tests with the EEVC- and CCMC-barrier have been published in[6] and[12].

For the *vehicle tests*, models representing a high proportion of the car population were selected. As in the tests described in[6] the two-door VW Golf/Rabbit (German version), with the highest market representation in the Federal Republic of Germany in this vehicle class, was the basic vehicle used. The W123 model from Daimler-Benz was chosen as the larger vehicle type, again having the highest market representation among heavier (intermediate) vehicles.

To allow discussion of a wide range of impact conditions, tests were additionally conducted in which the impacting vehicle was brought into a dived braking position, as well as two further tests exhibiting the intrusion of a lighter vehicle into the side of a heavier vehicle. The dived braking position of the VW Golf was simulated in these tests by fixation of the front axle suspension (lowered 50 mm, corresponding to a braking deceleration of 0.6 g). Also tests against a 4-door Golf were conducted as well as barrier tests with a modified ground clearance of the deformable face.

Furthermore, tests with a mass reduced from 1100 kg to 950 kg in order to allow determination of the influence of just the mass in conjunction with this deformation element were conducted.

The main tests (as reported in chapters 5 and 6) were carried out with stationary cars and 50 km/h; the tests with two moving cars with 50/25 km/h and tests with the crabbed barrier with 54 km/h. The tests with two moving cars were conducted on a new part of our crash test facility and they could be run with a good repeatability. They are thought to have the best fidelity to real life accidents and they were used to judge a proposal with the crabbed mode and European deformable element.

Standard measuring equipment and high speed cameras were used. *Tables 1 and 2* show the test set-ups. Hybrid II dummies were used to allow comparability with earlier test series and because no side impact dummies were available at that time. Results from our tests with EEVC barrier (90° and crabbed) and the new European side impact dummy EUROSID are reported separately on this conference[10] as already mentioned above.

Comparison rigid/deformable face

Only for seeking completeness and for alignment with the historical development of moving barriers

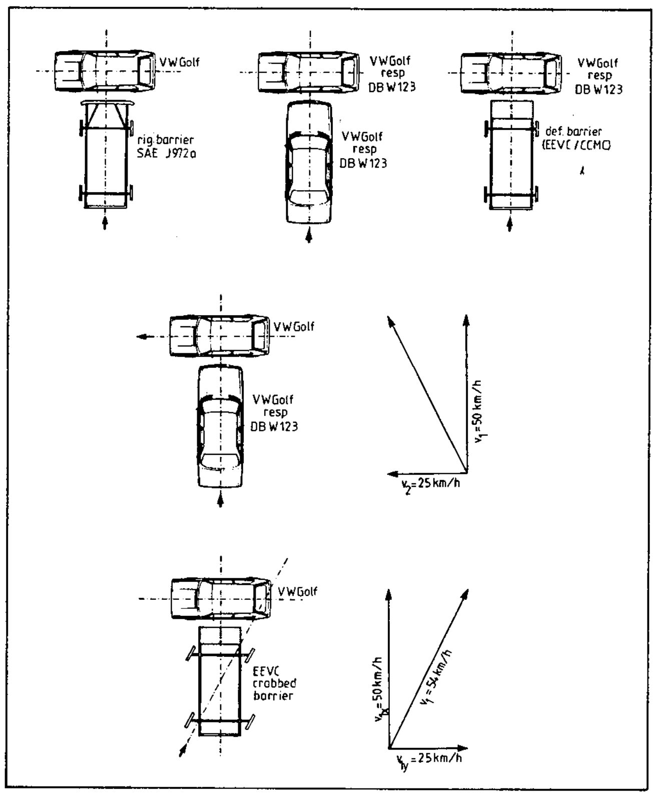


Figure 1. Test range

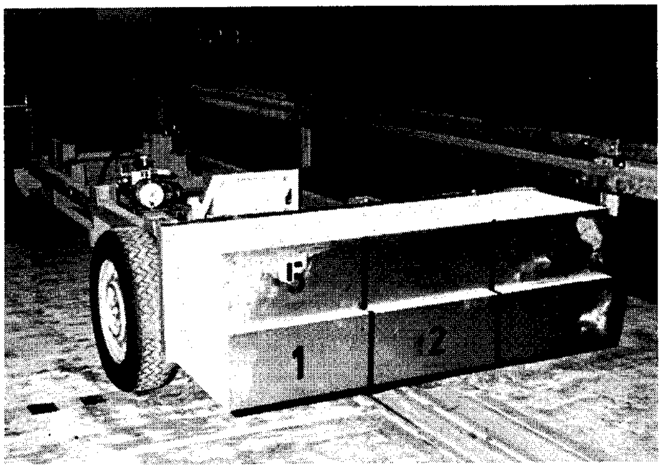


Figure 2. Mobile deformable EEVC-Barrier

some main facts concerning rigid/deformable faces are shortly reported again. For these tests the CCMC element was used because the EEVC one was still not available at that time.

Neglecting the advantages of deformable faces, the rigid barriers have some basic advantages too:

- they give very good reproducibility,
- they need only small effort (calibration etc.) for test equipment, and
- the tests are cheap.

For these reasons they were used for a long period of time. But as stated in[6], a mean level of fidelity compared to real car behavior should be realized. So the rigid barrier gave considerable lower and less fractured deformations (particularly at waist level) and in the softer regions but, at the same time, greater load transfer to the A- and B-pillars and shorter pulse durations. Concerning the dummy kinematics the dummy of the impacted side is accelerated first via the thorax and then via the pelvis (by the practically uniformly intruding side wall when struck by the rigid barrier). The sequence is reversed in car/car collisions and with the deformable barrier. In tests with the VW Golf, the dummy thorax loading is too high with the rigid barrier.

In Fig. 3 the shaded area with the left limit curve (lower limit values from the Golf-against-Golf collisions) and the right limit curve (upper limit values represented in each case by the highest values from the DB 2123 against Golf collisions) represents the dummy loading field which should be reproduced by a suitable test barrier.

Influence of barrier parameters on the struck car

Mass of the barrier as well as stiffness and ground clearance of the deformable face are important parameters for a movable barrier. Therefore a barrier

Table 1. Test parameters for the 90° impacts.

collision partners / test parameter veh. 2	rigid barrier vs. VW Golf	CCMC + EEVC vs. VW Golf		EEVC-barrier vs. DB W 123	VW Golf* vs. VW Golf	DB W 123 vs. VW Golf	DB W 123 vs. DB W 123	VW Golf vs. DB W 123
collision speed [km h]	45	45	45 + 50	50	50	45	50	45
impact angle	90° right	90° left		90° left	90° left	90° left	90° left	90° left
car masses test weight** [kg] veh. 1	1100	1100	950	950	800	1460	1460	800
veh. 2	800	800	800	1460	800	800	1460	1460
mass ratio m 1 / m 2	1,38	1,38	1,19	0,65	1	1,83	1	0,55

* also dived braking position

** test weight = curb weight + measuring equipment (without dummies)

test should take care for the engagement of the door sill, dived braking position of the striking car and the design of the struck car (2 or 4 doors).

The mass should reflect from our point of view the mean European car and a range from 950 to 1100 kg is interesting. *Fig. 4* shows the influence on lateral intrusion.

The influence of only the mass of the deformable barrier (all other parameters unchanged) on the deformations produced and dummy loads (*Fig. 3*) is only very small, at least in this range. By the way, the sensitivity on the test speed (the second parameter influencing the kinetic energy of the barrier) on the deformations produced with the deformable barrier is, as expected, high, see *Fig. 4*.

Load distribution between the softer door regions and the stronger regions at A-, B- and C-pillars as well as at the door sill is another important point. *Fig. 5* gives a typical impression from a small car/larger car test. The deformation on the struck larger

Table 2. Test parameters for the 63° impacts.

Collision-partner / Test-parameter		VW Golf vs. VW Golf	DB W 123 vs. VW Golf	EEVC-"crabbed" barrier vs. VW Golf
	Veh1 / Veh2			
Collision velocity	Veh1	50	50	54
[km h]	Veh2	25	25	0
Collision angle (Angle of the central longitudinal planes)		90° left	90° left	90° left
Impact angle (Direction of impact)		63° left	63° left	63° left
Impact location		Outer left edge of Veh1 impacts 37'' in front of the centre of wheel base of Veh2		
Test weight*	Veh1	800	1460	1100
	Veh2	800	800	800

* unloaden weight plus measuring equipment without dummies

car was unexpectedly high. On the other hand, tests against a 2 door Golf showed no differences versus tests against a 4 door Golf. This depends on the constant and relative small distance between A and C pillar.

In most of our car/car tests we have seen an override over the door sill, even in a dived braking position. This behavior of cars delivers the main determinant for the ground clearance of the deformable barrier face. The former specification of CCMC was 300 mm and that of EEVC 250 mm. Especially EEVC, taking this decision, awaited changes in compatibility which had rarely existed up until now with intrusion at waist level (250 to 400 mm) being almost double the intrusion at door sill level (150 to 250 mm). Consequently, measures to solve this problem should not only consider reinforcement in the door area but perhaps also higher deformation load transfer to the door sill. This could perhaps be accomplished by raising the door sill and/or lowering the strong longitudinal members of the striking vehicle.

As *Fig. 6* shows, the *braking position* in the Golf/Golf combination has practically no influence

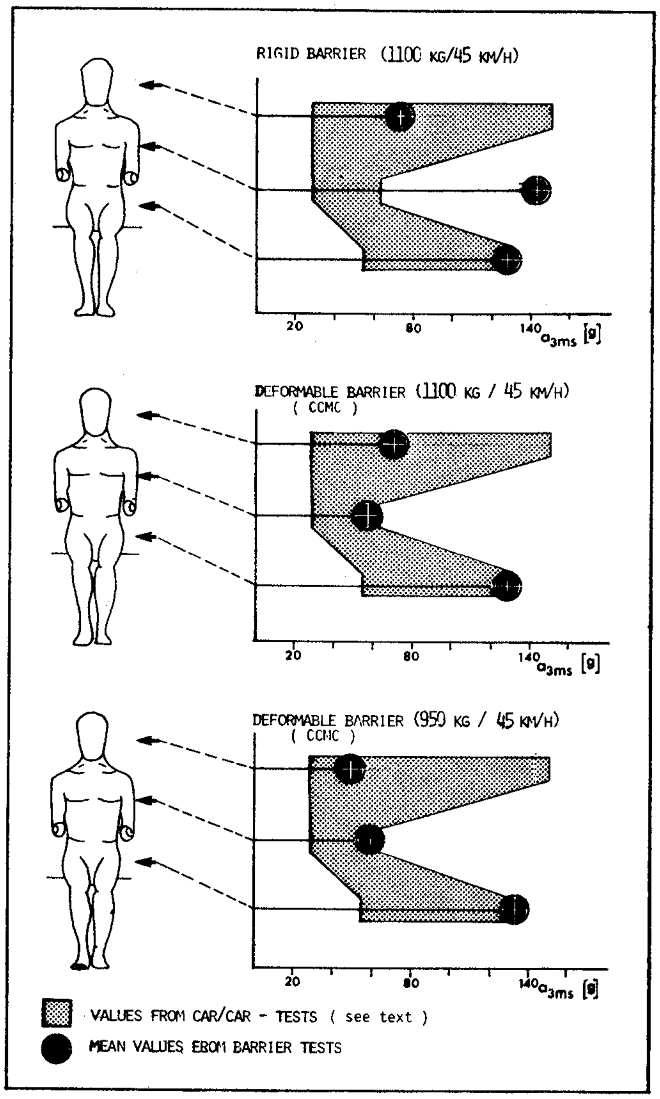


Figure 3. Comparison of the acceleration values for the near-side dummy in the struck car

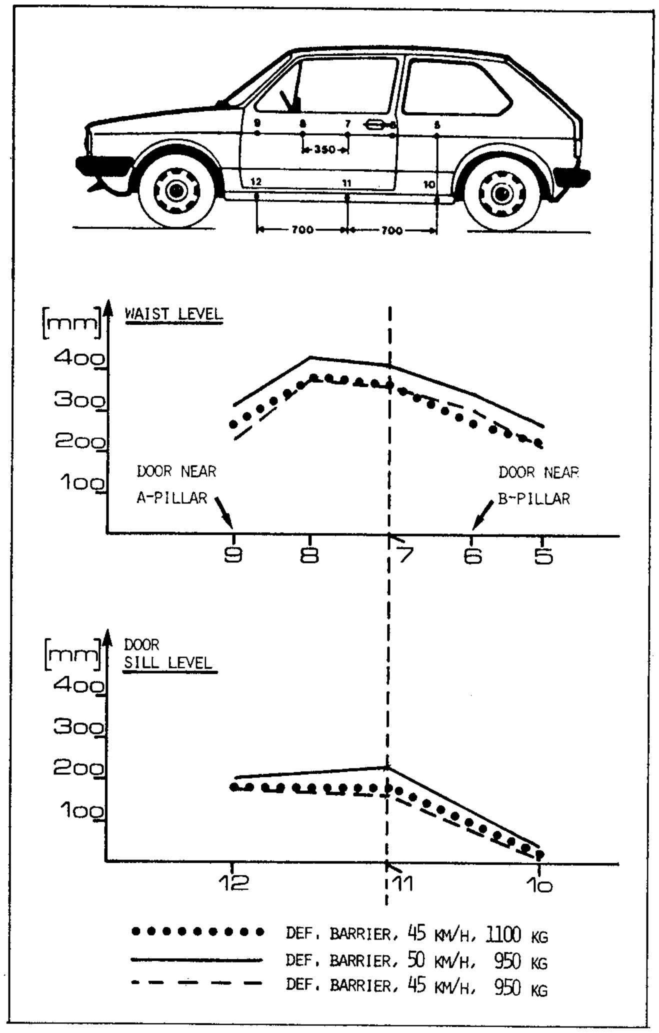


Figure 4. Influence of barrier mass and test speed on intrusion

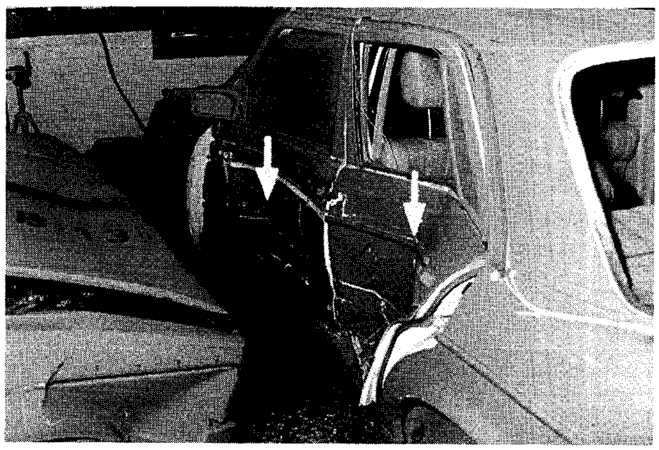


Figure 5. Lateral intrusion VW Golf/DB W 123 (45 km/h)

on intrusion characteristics even at the door sill level (including override of door sill). As, on the one hand, it can be assumed that in a real accident scene there is generally a braking reaction by the impacting car[15] and, on the other hand, it is desirable for compatibility reasons that the stiff vehicle parts contact in each case (door sill against strong longitudinal members), determination of the ground clearance of the barrier must take these factors into account.

For today's cars now also EEVC reconfirmed a ground clearance of 300 mm, which we also state as the right value because only with this ground clearance we see in all our barrier tests the realistic override of the door sill (see Annex A).

Barrier/car tests versus car/car tests

In this series, tests were carried out with the EEVC barrier against the VW Golf. The results of the vehicle and dummy loads were compared with those from another test series, VW Golf versus VW Golf. In addition, a similar series of tests was carried out with the DB W 123, results of which were used to demonstrate differences in vehicle and dummy behavior. All these tests were run with 50 km/h and the most important results from these tests are as already stated in[9].

Golf/Golf Tests

During the tests, the deepest deformation was measured in the front door area. The deformation of the door sill on the struck vehicle was low due to the relatively high ground clearance of the striking Golf. The A-pillar was ruptured at the transition of the sill. The dummy loads were comparatively low. The protection criteria were exceeded in only one test for the dummy (driver) pelvis (*Fig. 7*).

EEVC Barrier/VW Golf Tests

The barrier produced a very deep intrusion in the doorsill of the struck Golf; the profile of the deformation in the upper door area was similar but less deep. The maximum deformation was also in the front door area. The dummy loads were also greater than in tests with the VW Golf. The pelvis protection criterion was exceeded in all tests (*Fig. 7*).

DB W 123/DB W 123 Tests

In the tests, deformations in the upper door area were large and had approximately the same depth as in the tests with the VW Golf. The maximum side deformation in this case, however, was in the area of the B-pillar. The door sill deformation was significantly higher than in the Golf tests. The protection criteria were exceeded in all tests for the chest (60g over 3ms) and the pelvis (80g over 3ms), *Fig. 7*.

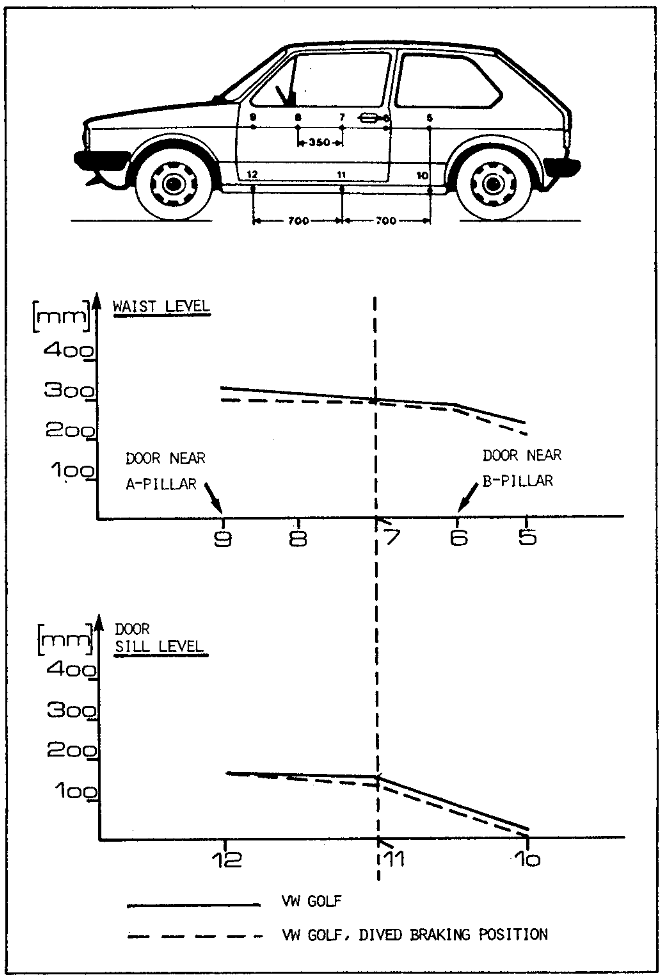


Figure 6. Influence of dived braking position on intrusion

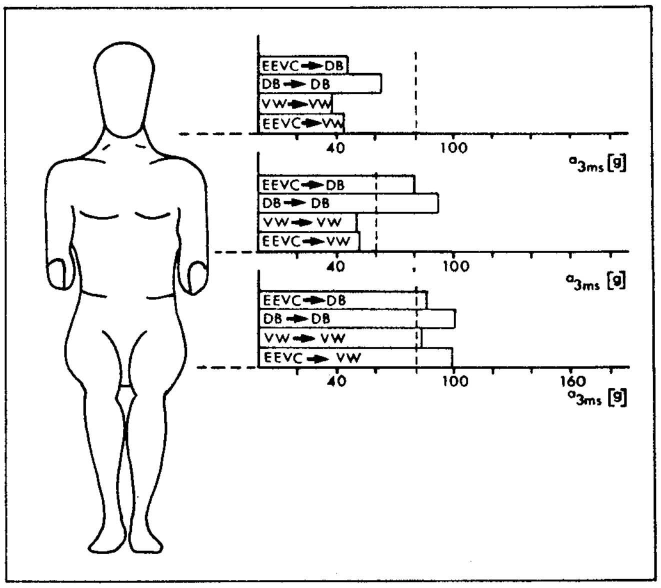


Figure 7. Comparison of acceleration mean values (a3ms) for various regions of body of driver dummy (Hybrid II)

EEVC Barrier/DB W 123 Tests

The deformations in the area of the doorsill were similar—the areas in the middle and upper side door were deformed considerably less than in the DB W 123/DB W 123 tests. The area of maximum deformation was again in the area of the B-pillar.

The similarity of the loading by the EEVC barrier compared with the DB W 123 is also indicated by the dummy loads; they are in a similar proportion for parts of the body but, as a result of the lower vehicle acceleration and deformation depth, are lower overall. The protection criteria for chest and pelvis were also exceeded during the barrier tests.

For both test series the following comments can be made regarding the suitability of the EEVC barrier:

- The distribution of the rigidity zones in the horizontal plane of the EEVC barrier conforms well with the demands of such a wide spectrum as that of the VW Golf and DB W 123 as shown by comparison of the deformation profiles. This is also valid for Japanese cars[15].
- The ground clearance of the EEVC barrier elements is too low to produce deformations similar to those produced by cars currently on the road in small vehicles, e.g., the VW Golf tested here. If the position of the doorsill, e.g., on the DB W 123, is so high that even vehicles currently on the road cause a deep deformation on the doorsill during lateral collisions, the area of the barrier that is superfluous for sill deformation appears to be lacking in the upper door area: i.e., for a better simulation for this type of vehicle, the ground clearance of the deformation elements had to be raised.
- The dummy loads in the EEVC barrier tests were, in all cases, similar to the loads caused by real vehicles and demonstrated only higher or lower values through the differences in impact energy. During the EEVC barrier/VW Golf tests, the pelvis protection criterion (80g/3ms) was exceeded. In the tests between the EEVC barrier and the DB W 123, chest (60g/3ms) and pelvis (80g/3ms) criteria were exceeded.
- The EEVC barrier is capable of simulating well the specific deformation characteristics even with the widely differing vehicles. The rigidity of the hard foam elements combined with the aluminum panels is a good approximation of the average vehicle structure. In view of the test results shown here, a correction of the ground clearance of the deformation elements appeared to be necessary for it to conform with vehicles currently on the road.

Tests with two moving cars

The advantage of this test arrangement lies in its close resemblance to a real accident, when both vehicles are moving. As normal series cars were used in the test as opposed to passenger car models (mobile deformable barrier), the results can be used as authoritative limit values when assessing the test procedure using a "crabbed barrier." In each case, three tests were conducted with a typical smaller vehicle (VW Golf) and a larger vehicle (DB W 123) as the striking vehicle (Veh. 1). This made it possible to determine an upper and a lower limit for the load on the struck vehicle, an important factor since test procedures for assessing passive safety must always consider vehicles of various construction types and masses for the colliding vehicle.

Despite the complexity of the tests, it was possible to conduct them well within the self-defined framework of the requirements for impact point accuracy (50 mm) and velocity tolerance (± %)[11]; *Fig. 8*.

Whereas the damage to the struck vehicle did nearly not differ from that sustained in the tests with the stationary vehicle, the damage to the striking vehicle clearly showed the influence of the transversal motion. The differences in frontal deformation could be clearly determined from the lateral displacement of the front structure of the vehicle to approximately the height of the front axle suspension, as shown in *Fig. 9*.

When assessing the lateral deformation of the struck VW Golf, note should be taken of the fact that the point of impact for tests with a stationary vehicle differs from that for tests with a moving vehicle. This difference is approximately 200 mm towards the front of the struck vehicle when it is in motion. However, the deformations do not differ to a great extent. For both types of tests, the door sill deformation and the deformation in the region of the R point (Point 10,20) which is important for the load on the occupants, are approximately equal (*Fig. 10*).

A comparison of the dummy measurement values for tests with a stationary and moving vehicle 2 reveals that the values for the front occupants are higher in the case of the moving vehicle (see *Fig. 12*).

From this it can be established that the vehicle and dummy loads for the tests with two moving vehicles of *the same mass and type* differ from tests in which one vehicle is stationary in the following ways:

- A lower resultant vehicle acceleration and a considerably longer impact time produce a greater velocity change due to the influence of the high x-component (Veh. 1) of the impact force.

- The deformation of the struck vehicle is similar to that in tests where vehicle 2 is stationary. This is particularly true for the R point region.
- The measured thorax and pelvis accelerations are considerably higher for the driver dummy and are not the result of unfavorable passenger cell loads but of the different dummy kinematics produced by the transversal motion. The protection criteria for the chest and pelvis of the driver dummy are exceeded in the tests with two moving vehicles (*Fig. 12*).

The results of the tests with two moving vehicles of *different masses and types* are consistent—with regard to execution and measurement values—with those involving vehicles of equal mass. In these tests it was also possible to clearly establish the influence of transversal motion on the striking vehicle.

The lateral deformation of the moving vehicle (VW Golf) in the door region was of approximately the same magnitude as that caused in tests with a DB W

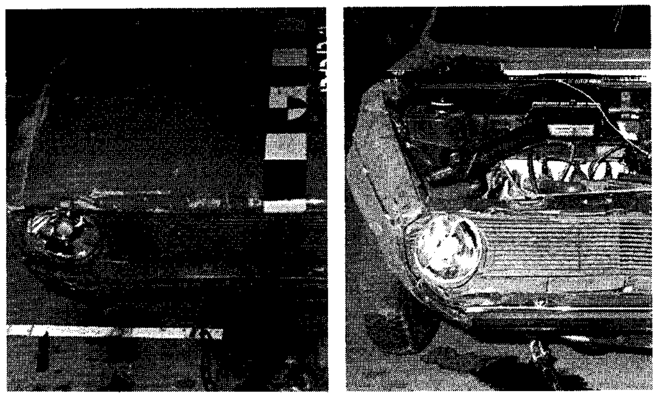


Figure 9. Comparison of the front deformation in tests with a stationary (left) and moving (right) vehicle

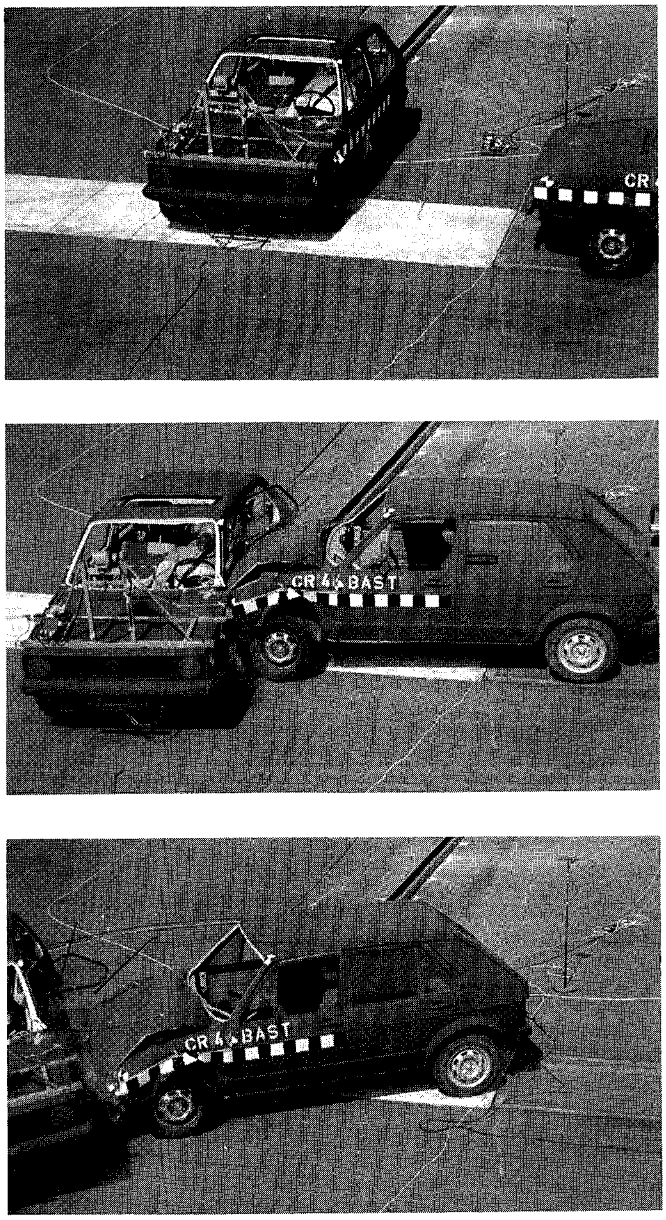


Figure 8. Test with two moving cars 50/25 km/h

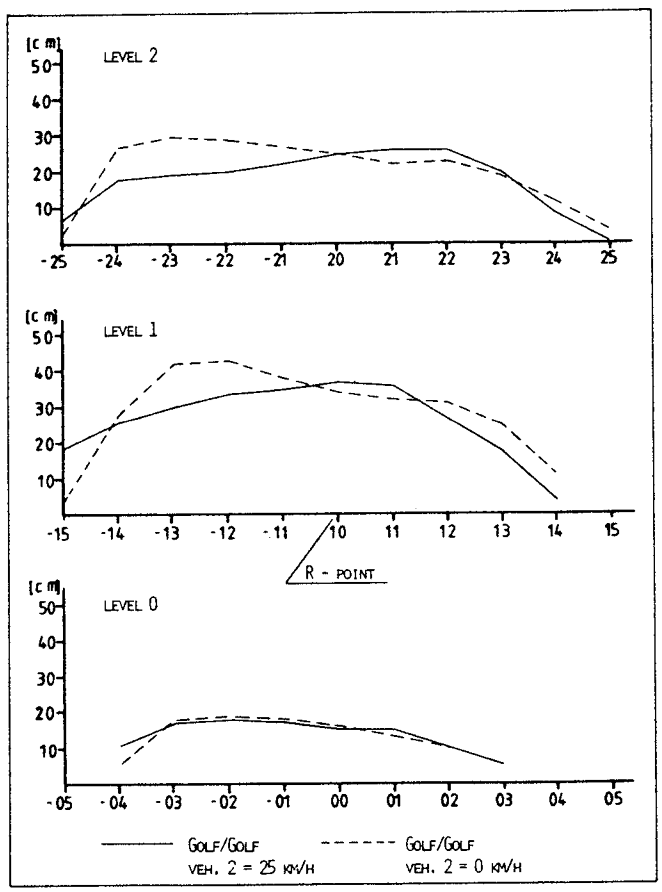


Figure 10. Comparison of the mean deformation values from Golf/Golf tests in which the struck vehicle was stationary in one test series and traveling at a velocity of 25 km/h in another

123 and a stationary VW Golf. However, when comparing the data, it must be noted that the earlier tests with a stationary VW Golf were only conducted with an impact velocity of 45 km/h for the DB W 123.

In addition it is possible to determine a decisive difference with regard to the sill deformation. In the case of the stationary VW Golf, the front structure of the striking DB W 123 deforms the sill of the VW Golf, particularly in the region of the side door. In the case of the moving VW Golf, the DB W 123 slides over the sill (in the door region), thereby causing lower deformation to the sill itself, but producing greater deformation to the lower door region. This sliding movement over the door sill of the VW Golf is made possible by a movement of the DB W 123 around its roll axis, caused by the proper motion of the VW Golf.

A comparison of vehicle and dummy loads for tests with a larger vehicle 1 and a stationary or moving vehicle 2 gives the following differences:

- As far as can be established with the different test velocities and deformation measurement points, the deformation—particularly in the sill region—is smaller in the case of two moving vehicles. Despite the fact that the striking vehicle slides over the sill, the deformation of the door region is less than in the comparative tests ($v_2 = 0$), particularly when one considers the somewhat lower velocity (45 km/h) of the comparative tests.
- In the tests with the moving vehicles, the protection criteria for the 3 ms acceleration values were clearly exceeded for the chest and pelvis (*Fig. 12*).

When comparing these tests with the tests involving moving vehicles of equal mass, mention must first be made of the change in impact direction caused by the different masses and second of the fact that the striking vehicle slides over the sill of vehicle 2 since the vehicle kinematics are affected.

Crabbed mode of barrier

The aim of the tests with the crabbed barrier is to simulate the loads during a lateral collision test more closely to those of a real accident than has previously been possible with earlier test procedures.

This was achieved by setting the vehicle at the beginning of the collision in such a way that their central longitudinal planes formed an angle of 90° (the collision angle), although by parallel angling of the wheels of the front and rear axles, the direction of impact of the barrier produced a direction of motion, and therefore the main direction of force of the barrier, which ran at an angle of 63° to the central longitudinal plane of the struck vehicle (the impact angle, Def. according to DIN/ISO 6813). The geometric dimensions of the barrier used for the tests corresponded to those of the EEVC barrier, the mass was increased to 1100 kg in accordance with[1]. The deformation element used was the EEVC element described in[8 and 9].

The behavior of the deformation elements during the impact corresponded to those of a right-angled impact. After the test, the front row of individual blocks had been broken into many fragments and separated from the rear row, which, in general, maintained its form, apart from undergoing the desired longitudinal deformation. *Fig. 11* shows the status of the deformation element after the crabbed collision.

A comparison of the tests with a normal EEVC barrier impacting at an angle of 90° and an EEVC barrier traveling at an angle and impacting at an angle of 63° thus reveals the following differences:

- The angled barrier produced an average change in longitudinal velocity of 6,5 km/h in the struck VW Golf, whereas this figure is almost zero for the impact with the normal EEVC barrier at the same point in time.
- The "crabbed" EEVC barrier produces a somewhat smaller deformation depth with a similar deformation profile (see Annex B).
- During a collision with the "crabbed" barrier, there was a considerably lower load on the front dummy (no protection criteria exceeded, *Fig. 2*) as a result of the lower deformation, the lower door penetration velocity, and the fact that contact with the side occurred earlier due to the changed vehicle kinematics, but film analysis showed no specific change in dummy kinematics. Lower dummy loads are conformed also with the EUROSID dummy[10] for the crabbed barrier.

The findings of the tests involving transversal movement (tests with two moving vehicles and with the

Figure 11. Final position after a collision with the "crabbed" EEVC barrier

"crabbed" barrier) can also be used to assess the simulation quality of the "crabbed" barrier. This requires a comparison between the factors which are clearly influenced by the transversal movement, e.g., the deformation, acceleration, change in velocity and the dummy load which is dependent on these factors.

The deformation caused in these tests in the different measurement planes show different degrees of correspondence. In the vehicle/vehicle tests, the sill deformation is small and results from the front of the vehicle sliding over the doorsill. The barrier causes a deep deformation of the sill without sliding over it. In the region of the R point, the deformation of the centre of the door shows a good degree of correspondence; however, the deformation caused by the crabbed EEVC barrier is a little too low. In addition, the deformation to the upper door area is not sufficiently deep in the barrier tests due to the fact that the deformation element does not slide over the doorsill. A comparison shows that the "crabbed" barrier produces similar deformation profiles to those produced with moving vehicles but that, because of the low ground clearance of only 250 mm, the behavior of the barrier is fundamentally different in the sill region. If the ground clearance were to be increased, thereby achieving the same sill deformation behavior as for tests with real vehicles, this would produce a deeper deformation in the upper measurement planes[12]. The ideal deformation, i.e. a course between the characteristics of the smaller passenger car (VW Golf) and the larger vehicle (DB W 123), was not achieved and the "crabbed" barrier does not result in an improved simulation of deformation—indeed, in the areas subject to loading, the simulation is not adequate.

In the tests with two moving vehicles, the acceleration values of the front dummies are always higher than those obtained with the "crabbed" barrier (*Fig. 12*). With the "crabbed" barrier collision, the front dummy load never exceeds the load criteria. In the tests with two moving vehicles, the load criteria for the chest (60 g over 3 ms) and the pelvis (80 g over 3 ms) are exceeded for both series of tests.

As already established in tests with a stationary vehicle[12], the most important reason for the dummy load deviations measured in barrier tests is the deformation behavior of the door sill as a result of the insufficient ground clearance of the deformation element. Comparative tests with a larger ground clearance (in tests without any transversal movement)[12] revealed that the load on the chest and pelvis is increased by an average of 10 - 15 g with a ground clearance of 300 mm (instead of 250 mm as here). As already mentioned, an increase in the sliding motion of the striking vehicle into the door sill of the struck vehicle was observed in tests with two moving vehicles as a result of the lateral sliding motion of the vehicles during the collision. This was particularly marked in tests with the DB W 123 as the striking vehicle. However, this influence was not noticeable in the tests with the "crabbed" barrier.

The tests conducted with two moving vehicles to assess the simulation quality of the "crabbed" EEVC barrier enable the following conclusions to be drawn:

- The degree of correspondence in the deformation behavior in the sill area is insufficient, but it is adequate in the door region.
- The dummy load—measured with Hybrid II dummies—is much too low for the front occupant.

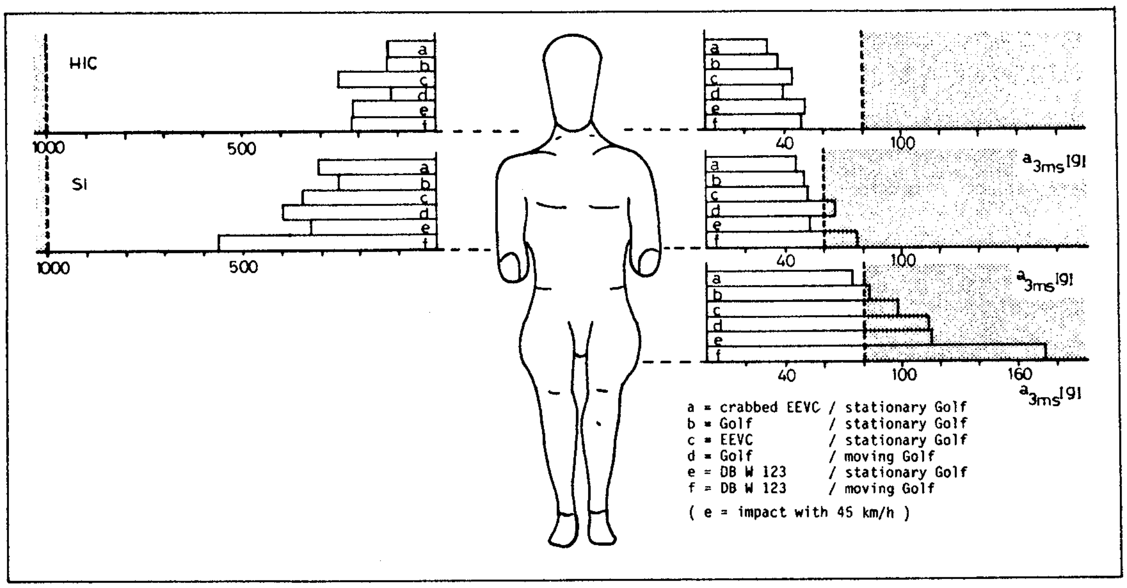


Figure 12. Measured loads on the front dummy, comparison of all test series used for the assessment (Hybrid II)

In particular, the dummy load exhibits a poor degree of correspondence when compared with the tests with two moving vehicles.

In addition, the "simple" simulation of a lateral collision (EEVC barrier test with an impact angle of 90° and with vehicle 2 stationary) already produces sufficiently good results with regard to vehicle load, and more realistic ones with regard to the load on the dummies. *Fig. 12* clearly illustrates that the loads with the "crabbed barrier" are the lowest with respect to acceleration values over 3 ms.

The action of the DB W 123 sliding into the sill of the Golf in tests with two moving vehicles can be considered a special further feature. Simulation of the transversal movement is not essential for taking into account the peculiarities in a lateral collision test, i.e., in order to adapt the test procedure; this can also be achieved by altering other parameters. A useful—and the easiest—measure involves increasing the ground clearance. [12] and [13] have already mentioned this as being important in achieving a closer resemblance to reality.

Performance of the EEVC barrier faces in calibration tests

In the previous chapters the principle concept with deformable elements on the movable barrier for side impact is found to be valid on the basis of comparison tests. With the integration of this proposal into a regulation it will be important to have a test tool meeting constantly the given specifications. Therefor, in parallel to the car tests, the so called calibration tests using a dynamometric wall were conducted. *Fig. 13* shows the results of those calibration tests with 35 km/h. These tests were described in[14], a paper from cooperating European laboratories.

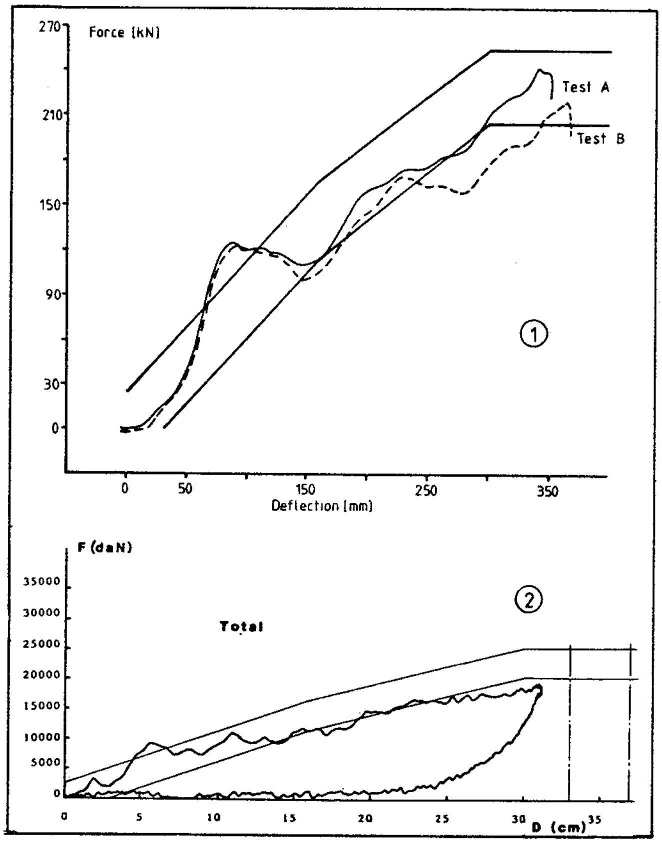
As can be seen from *Fig. 13* the prescribed force/deflection corridor could not be met over the total deformation. Improvements had to take place. From our point of view, the situation in Europe now is as follows:

- In France by Union Technique de L'Automobile, du Motocycle et du Cycle (UTAC) an EEVC barrier face made of aluminum honeycomb is preferred. It meets well the specifications
- In Great Britain, a barrier face is developed by Transport and Road Research Laboratory (TRRL) in close cooperation with a foam supplier. The deviations from the force/deflection corridor are of minor magnitude.
- In Germany where the first foam built EEVC barrier faces were developed and produced, three different modifications were additionally tested. Results from these tests are reported briefly.

The stepwise improvement led to the so called EEVC IV element. On this basis, three modified elements (A, B and C) were introduced to reduce the initial force peak and to increase the force level in the range of deformations greater than 200 mm. The modifications consisted of higher foam densities, mounting directions of the single blocks concerning their holes and changed bumper simulation concerning the aluminum cover.

The test results were not improved in all points. During the impact against the dynamometric wall, some greater undeformed foam parts were pushed away without having absorbed energy caused by a misfunctioning glewing. Greater deformation distances for the barrier were the output. The following *Figures 14 to 16* show the force deflection curves of the modified elements compared with the prescribed corridor.

On the one hand the initial peak could be reduced (by changed position of the front blocks), but on the other hand the force level from above 200 mm of deflection was decreased instead of increased. From film analysis, the modifications for reducing the initial peak seemed not to be interdependent with the bad



1. curve measured by TRRL [14]
2. curve measured by UTAC [14]

Figure 13. Force/deflection curves of early EEVC faces built of polyurethane foam[14]

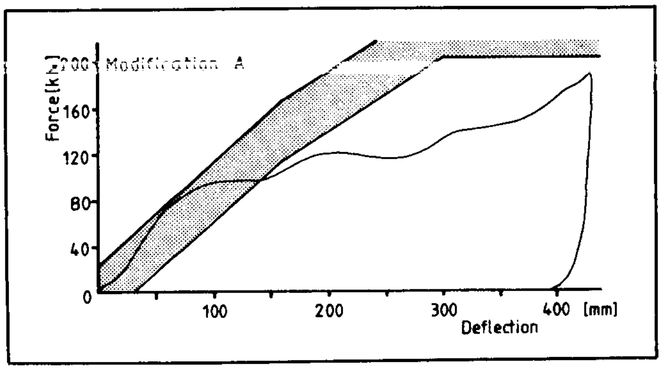


Figure 14. Force/deflection curve of the element with modified holes to reduce initial peak

glewing so that this problem seems to be nearly solved. However, before using the foam elements in the frame of regulations, a satisfactory production quality (especially glewing) and its superintention has to be guaranteed.

Summary and conclusions

Based on the output of EEVC Working Group 6 where the main parameters for a side collision procedure were defined, the Federal Highway Research Institute (BASt) has given considerable effort in the development of a foam built EEVC barrier face and its validation by conducting more than 80 full-scale tests.

The tests included (among others) main parameters like:

- rigid versus deformable faces
- barrier/car tests versus car/car tests (car stationary)
- tests with two moving cars (impact angle 90°)
- crabbed mode of barrier (impact angle 90°)
- tests with highly represented smaller (subcompact) cars (VW Golf I, type 17) and heavier (intermediate) cars (Daimler Benz W 123)
- calibration tests against a dynamometric wall.

The main results which can be drawn from this test experience are as follows if one takes car/car tests as basis of comparison:

- Rigid barrier faces have proved to be unrealistic for several aspects. The force/deflection characteristic of the EEVC face is suitable for European (and regarding other publications, also for Japanese) passenger cars and can be met, e.g. by foam elements (other solutions are awaited to be more expensive, aluminum honeycomb for instance at least twice).
- The collision configuration of the EEVC proposal is valid; that means collision angle = impact angle = 90°. This opinion is based on the following experiences:
  - The crabbed barrier does not result in an improved simulation of deformation.
  - The load of the driver dummy ranks from high to low in the following way:
    a) two moving cars, 90°
    b) car/car, 90°, struck car stationary
    c) deformable CCMC barrier, 90°
    d) deformable EEVC barrier, 90°
    e) crabbed barrier, EEVC face.
    That means the crabbed mode represents (for these tested car types) the lowest and probably unrealistic loading. For d) and e) it is the same sequence with the EUROSID dummy[10].
- The crabbed mode is slightly less reproducible and needs greater effort.
- Mass variation of the barrier ranging from 950 up to 1100 kg has a negligible influence; therefore any mass in this range can be accepted at least for European cars.
- The only parameter which had to be changed is the ground clearance (250 mm) of the

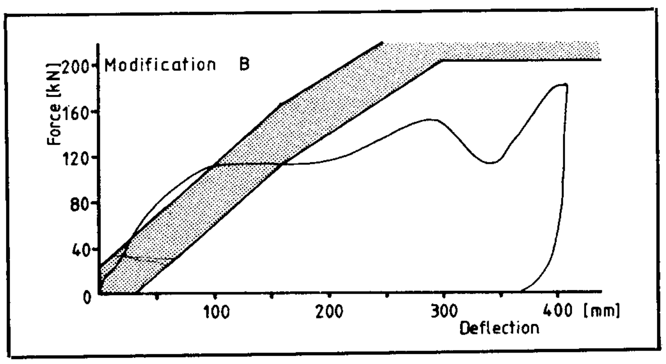


Figure 15. Force/deflection curve of the element with modified bumper simulation to reduce initial peak

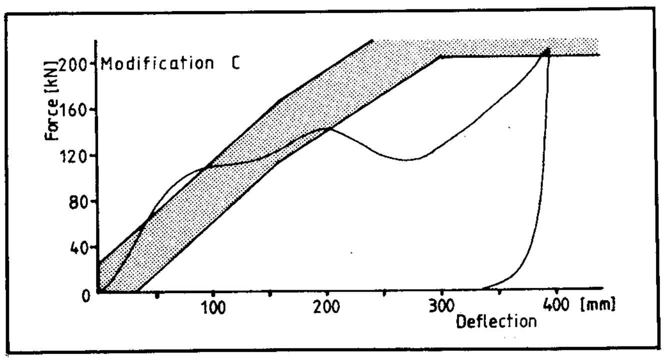


Figure 16. Force/deflection curve of the element with partly higher foam density to increase force level

deformable face. In most of our car/car tests we observed an override of the door sill even in the dived braking position of the striking car. This could not be reconstructed by the EEVC barrier with a ground clearance of 250 mm. That means for the barrier test an unrealistically high energy dissipation for the door sill and insufficient deformation in the upper weaker door region. Therefore it was inevitable to increase the ground clearance to a value of 300 mm. This value was also the former specification of the European CCMC barrier face and it is taken into account within a regulation draft for testing today's cars.

- Calibration problems for the 90° impact are thought to be solved but care has to be taken of the production quality of the foam elements because the glewing has an important influence in calibration tests (it is less important in car tests because of the smaller energy dissipation). There is up to now no idea how the calibration problems can be solved for a crabbed barrier because no data for the lateral stiffness of car front structures are available and nearly no dynamometric wall exists capable of taking into account the necessary measurement of lateral force components.

If the initial peak forces for foam elements can not be equalized, there is the possibility to consider them by allowing a defined deviation from the force/deflection corridor during the first 150 mm of deflection where the dummy is still not loaded. In the meantime, this is proposed already.

Acknowledgement

We wish to thank the European Community (EEC) for its fund support and the Committee of Common Market Automobile Constructors (CCMC) for permission to use the CCMC deformable elements for these tests.

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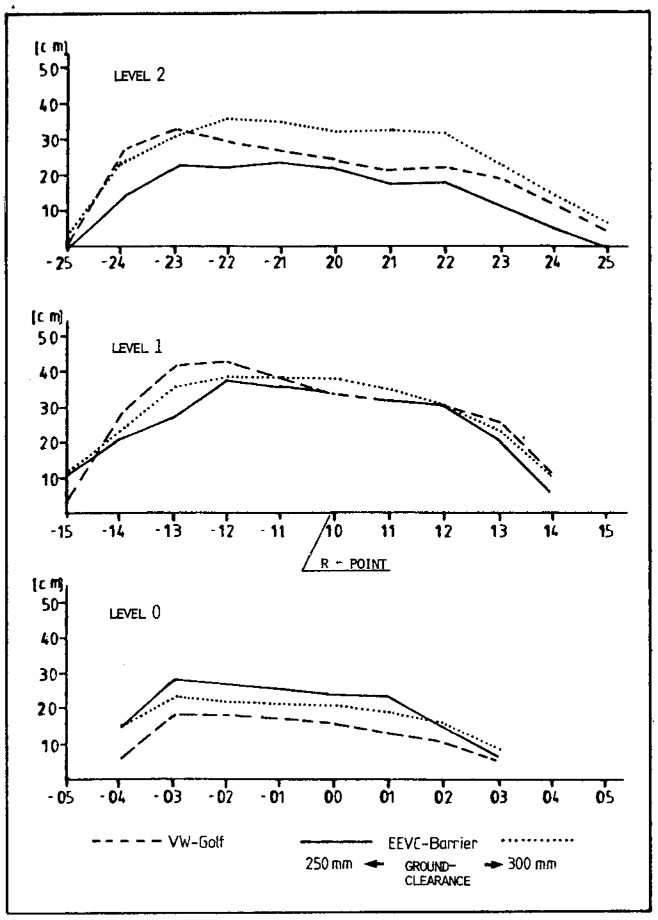
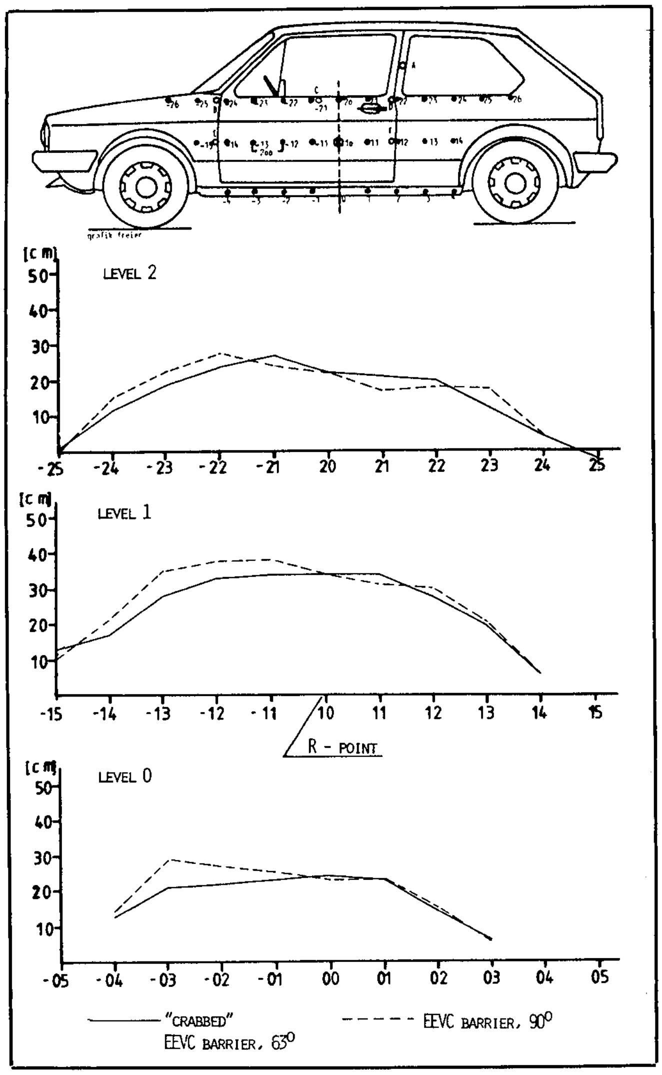
Appendix A:

Influence of the ground clearance of barrier face on the lateral deformation in relation to the Golf/Golf collision



Appendix B:

Comparison of lateral deformation in collisions with 90° and crabbed barrier



Evaluation of European and USA Draft Regulations for Side-Impact Collisions

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Abstract

Regulations on side-impact safety are being prepared in Europe as well as in the USA. Though on both continents regulations aim at the protection of occupants in the event of a side-impact collision, the drafts prescribe different test methods in which different types of side-impact dummies are used.

In the past, several studies were published on the subject. Some even proposed alternative test procedures that harmonized on certain fundamental test parameters, such as impactor weight and dimensions, impact speed, test configuration, dummy design and injury criteria (Xth ESV conference, Oxford, 1985).

This paper presents the results of five full-scale side-impact tests, applied to identical vehicles. One pair of tests was carried out in accordance with the

Draft EEC [1] TRANS/SCI/WP29/GRCS/R.58 document including the 1987 EUROSID production prototype with soft arm design and one pair in accordance with the NHTSA [2] proposal. The latter pair of tests included the 1986 SID. The fifth test was carried out with the EUROSID in the proposed NHTSA test setup in order to compare the performance and injury assessment of the two side-impact dummies.

It is concluded that the NHTSA test represents a more severe test condition than the EEC test. Both dummies sustain the tests without damage. The injury prediction potential of the EUROSID exceeds the SID. Definitive injury tolerance levels for both dummies are urgently needed.

Introduction

Vehicle collision directions can be divided into 12 main groups [1,2]. The rectangular side collision ranks third behind frontal and rear impacts. In the past several test methods for both frontal and rear impacts were developed. In more recent years European and US research efforts were directed particularly towards the realization of side-impact test methods [3,4]. In this field, however, rather deviating test procedures have been proposed due to lack of international coordination. In the proposed regulations use is made of different impact barriers and different side-impact dummies.

Several research institutes and car manufacturers have been performing tests in accordance with the proposals. Even harmonizing procedures have been developed and implemented to study the behavior of vehicle and dummy during side-impact. In none of these studies was use made of the EUROSID dummy as the dummy was not available yet.

TNO [3] has established a research program, in which the differences between the two regulations and the two dummies should be evaluated. The test program consists of the tests summarized in Table 1.

The most recent versions of the documents and the dummies were applied. As a consequence, due to adaption of the EUROSID in a late stage which influenced the availability, test 6 could not be conducted in time to include results in this paper. The tests were conducted with identical standard European manufactured sub-compact vehicles.

The objectives of this study were:

- to evaluate the differences between the two regulations,
- to analyse the differences in behaviour of the SID and EUROSID under identical circumstances,

Table 1. Test program.

test	procedure	dummy	configuration
1+2	NHTSA	SID	crabbed
3+4	EEC	EUROSID	non-crabbed
5+6*	NHTSA	EUROSID	crabbed

*) Due to late availability of EUROSID test 6 could not be conducted in time to include results in this paper.

- to study the effect of different barriers (in different crash configurations) on identical vehicles, and
- to compose a database for future occupant and vehicle structural mathematical simulations.

This paper starts with a brief summary of the side-impact test procedures. Subsequently, the specifications of vehicle, barrier and dummies are given. Then, the test results are presented for vehicle and dummies. In conclusion, the tests are discussed and conclusions are drawn.

Side-Impact Test Procedures

Three main elements are required for the development of a full systems test: the test conditions, the human surrogate and the standardized impactor. The test conditions should be relevant to actual real world accidents. The dummy must be similar to the human in dynamic response and should be based on the latest available biomechanical data. The side-impactor should have characteristics similar to actual vehicles. Each of these elements should be repeatable, reproducible and practical.

At the moment there are two government-supported, draft Side-Impact Collision Procedures. In the following these will be referred to as the NHTSA-procedure and the EEC-procedure. These two procedures differ strongly as to test set-up. Some of the differences relate to the differences in rolling-stock on the two continents. Others relate to the fundamental test parameters. Also, the regulations use different measuring tools and injury parameters.

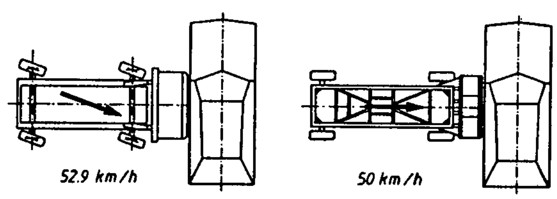

Figure 1. Test set-ups for (a) crabbed (NHTSA) configuration and (b) non-crabbed (EEC) configuration.

[1] European Economic Community
[2] National Highway Traffic Safety Administration, U.S. Department of Transportation
[3] Netherlands Organization for Applied Scientific Research

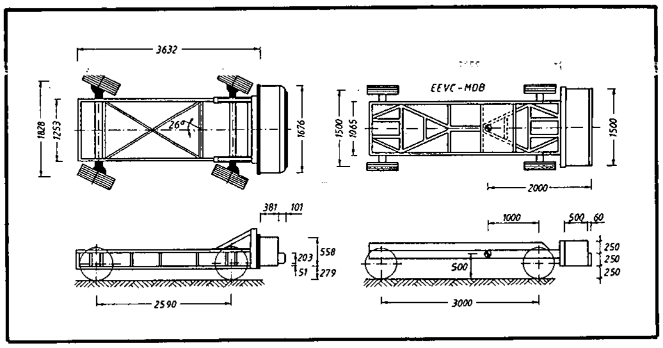

Figure 2. Moving deformable barriers according to (a) NHTSA procedure and (b) EEC procedure

The latest NHTSA procedure [3] proposes a test in which the struck vehicle remains stationary and the striking vehicle travels with a speed of 52.9 km/h (32.9 mph) in a crabbed configuration (see Figure 1a). The striking vehicle is represented by a moving deformable barrier (see Figure 2a) with an energy-absorbing impactor face which is based on the masses, dimensions and stiffnesses of vehicles registered in the USA in 1978. The left side of the barrier face impacts 940 mm (37″) left from mid wheelbase. As a human surrogate the so-called SID (Side-Impact Dummy) is prescribed[3]. This dummy is provided with a special thorax construction. No arms are used on this dummy. The head, neck and lower body are essentially the same as the Part 572 dummy.

The major features of the two test procedures are summarized in Table 2.

In the EEC test procedure [4] the vehicle to be tested is also stationary and a moving deformable barrier is used. This barrier travels with a speed of 50 km/h with a trajectory perpendicular to the struck vehicle (see Figure 1b). The symmetrical plane of the barrier passes through the R-point of the vehicle. In the document to EEVC [4] barrier has been adopted. The barrier (see Figure 2b) has a deformable front face the stiffness of which is derived from the characteristics of European vehicles. The front face consists of six deformable elements, divided into two rows of three. The dummy to be installed is the European Side-Impact Dummy [5] or EUROSID. This dummy has a specially designed thorax, shoulder, neck, abdomen and pelvis while, in addition, arms are included in this design. The head is identical to the Hybrid III and legs are standard Hybrid II design. The EUROSID dummy used in this test program is an updated version of the 1986 prototype reflecting the status of the 1987 EUROSID dummy. The major difference between the 1986 and 1987 version is in the arm design. In the 1987 EUROSID the metal arm skeleton is replaced by a flexible plastic structure.

[4] European Experimental Vehicles Committee

Table 2. Comparison of the NHTSA and EEC test procedures.

PROCEDURE	NHTSA	EEC
TEST CONFIGURATION		
impact speed	52.9 km/h(32.9mph)	50 km/h
impact centre	left side barrier front 940 mm (37") left mid wheelbase	symmetrical plane of barrier passes through R-point
impact angle	90°	90°
crabbed angle	26°	0°
barrier braking	after 200 ms	- *)
VEHICLE		
front seat position	midpoint	50 mm foreward of rearmost position
front seat back angle	manufacturer's stated nominal design position	25° rear or manufacturere's stated nominal design position
fuel tank	filled up to 93% capacity	filled up to 90% mass
BARRIER		
weight	1338 kg (2950 lbs)	950 kg
front ground clearance	275 mm (11")	250 mm
front height	550 mm (22")	500 mm
front width	1650 mm (66")	1500 mm
front material	aluminium honeycomb	free, according to specifications **)
DUMMY		
type	SID	EUROSID
number ***)	driver + rear left passenger	driver
configuration	50th percentile male	50th percentile male
detail	no arms; increased chest mass	arms

*) There is no definition in the EEC test procedure.
**) In the tests described in this paper the EEVC polyurethane foam front was used.
***)In spite of the procedure, in each vehicle a 75 kg replacement load was applied as to account for a left rear passenger.

Test Method

Vehicle and barrier preparation

Vehicles were prepared according to the specifications in [3,4]. Accelerometers were placed on the front and rear sills, the inside left door and the left B-pillar. To enable the inside car deformation and the dummy driver movements to be observed, the dummy rear passenger was replaced by a 75 kg load, secured to the floor. For better dummy observation from the rear vehicle camera the lower part of the covering of the seat back was removed. The hood was removed to supply fixation of the side vehicle camera. The vehicle's struck side was screened with a pitch of 100 mm in order to measure the deformation. The dimensions are based on the projection on a longitudinal vertical plane. The deformations will be presented in 4 levels, respectively denoted by LVL1, LVL2, LVL3, LVL4 in Figure 3.

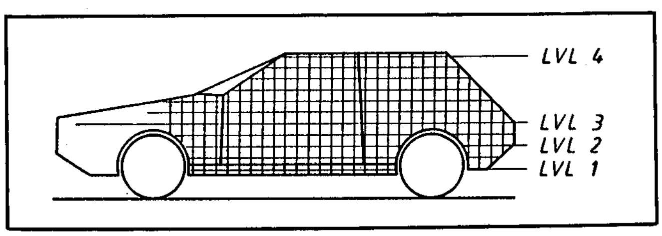

Figure 3. Test vehicle deformation levels

Table 3. SID instrumentation.

	transducer	CFC
Head	triax. accel.	1000
Upper rib	uniax. accel.	FIR*)
Lower rib	uniax. accel.	FIR*)
Rib deflection	lin. potmeter	1000
Lower spine T12	uniax. accel.	FIR*)
Pelvis	triax. accel.	FIR*)

*) Finite Impulse Response 100/187/-60dB

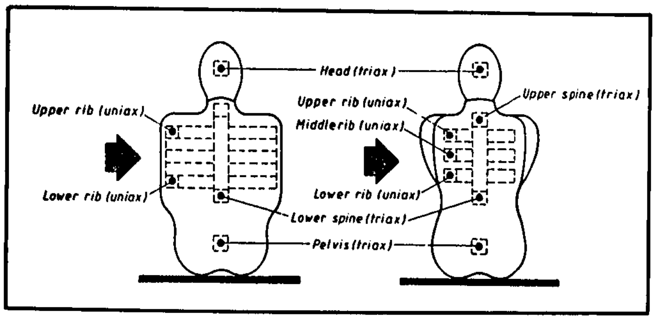

Figure 4. Rear views of a) SID and b) EUROSID and their respective accelerometer locations

Both the NHTSA and the EEC moving deformable barriers were equipped with three triaxial accelerometers: the NHTSA barrier on the front and rear members and in the centre of gravity, the EEC barrier on the left and right sides and in the centre of gravity. The NHTSA barrier was equipped with a HEXCEL energy-absorbing face assembly; the EEC barrier was provided with a FRITZMEIER polyurethane foam front, the so-called EEVC-IV element. To diminish the chance of this front disintegrating during the crash, a plastic foil was wrapped around the foam.

Dummy preparation and positioning

A side-impact dummy is used as a tool for determining occupant injury levels sustained during crash tests. The dummy (SID and EUROSID) must be calibrated prior to use. For both dummies special calibration procedures have been developed [3,5], consisting of a series of tests applied to dummy components and to the complete dummy.

The SID is being equipped with accelerometers (see Figure 4a) and a displacement transducer. Location, type of transducer and filter classification are summarized in Table 3.

The EUROSID was equipped with accelerometers, displacement and force transducers and level detecting switches (see Figure 4b) . Table 4 shows location, type of transducer and filter classification used.

Table 4. EUROSID instrumentation.

	transducer	CFC
Head	triax. accel.	1000
Upper spine T1	triax. accel.	180
Upper rib	uniax. accel.	180
	displ.transd.	180
Middle rib	uniax. accel.	180
	displ.transd.	180
Lower rib	uniax. accel.	180
	displ.transd.	180
Lower spine T12	uniax. accel.	180
Abdomen	3 switches	1000
Pelvis	triax. accel.	180
-pubic symph.	force transd	600
-iliac wings	straingauges	600

In all tests the dummy was set in the position as described in the seating procedure. The dummy was belted by a three point safety belt. The hands (of EUROSID) were taped on the steering wheel in a ten to two position. The right foot was placed on the accelerator pedal and the left foot on the floor left of the clutch pedal. To enable the dummy movements to be observed, the dummy was wearing just a pair of shoes and a pair of trousers. To detect dummy contacts, copper foil was fixed to the vehicle's inside padding and the head, chest and pelvis.

Test Results

Vehicle motions and deformations

Gross motion

The symmetrical planes of the two barriers show a shift of about 175 mm with respect to the vehicle, where the EEC barrier is located more backwards (see Figure 5). This is caused by the different definitions of impact points and different barrier front widths. As a result, different force origins and directions are introduced, causing different vehicle gross motion. During the NHTSA tests, the vehicle as well as the barrier rotated around their vertical axes. Moreover, the crabbed motion of the barrier caused a velocity in longitudinal vehicle direction. During the EEC tests,

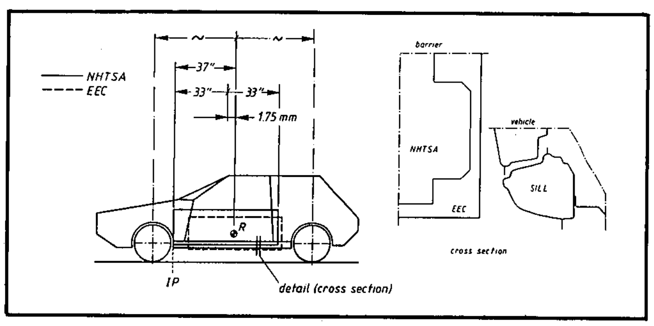

Figure 5. Initial barrier positions and barrier ground clearances

only the vehicle rotated around its vertical axis. This finally results in different post-test vehicle positions (see Figure 6).

Because of the larger ground clearance of the honeycomb front (especially regarding the bumper section) and the relatively low positioned sill of the vehicle, overriding of the sill occurred in the NHTSA tests (see detail Figure 5). The barrier remained in contact with the vehicle. This was in contrast with the EEC tests, where the vehicle was thrown away by the barrier.

Local deformations

As a consequence of the width of the honeycomb front and the location of the impact point in the NHTSA procedure, both the A-pillar and B-pillar are impacted, causing gross deformation just above floor level. However, the honeycomb front just showed some local deformation. In the EEC procedure the barrier impacts the B-pillar and the stiff floor (see Figure 6a). The impact load is transmitted through the sill, causing the foam to disintegrate. As a result, the crash-affected zone moves up to the window part of the side structure.

In the NHTSA test the top camera shows considerable bending of the centre line of the vehicle during the crash. Afterwards a permanent bend is observed. Bending appears to be less in the EEC tests. This difference in behaviour may be attributed to the bigger mass of the crabbed barrier, as well as to its direct impact on the A-pillar.

The side deformations of the vehicles show good reproducibility. The results of the tests 1,2,5 and those of the tests 3,4 are very much alike; a typical example of both tests is given in Figure 7.

Figure 8 shows the average side deformation of the B-pillar. The difference in aggressiveness of the honeycomb front and the foam front is clearly illustrated.

Dummy response

In spite of the relatively high load level during the NHTSA as well as the EEC tests, no damage of SID and EUROSID was observed after the tests.

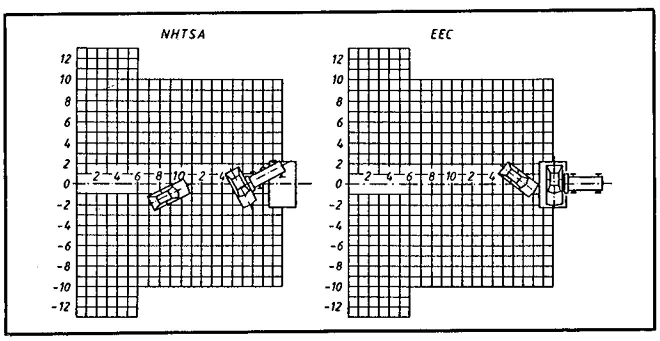

Figure 6. Pre-test vehicle and barrier positions and post-test vehicle positions (post-test barrier positions have been omitted)

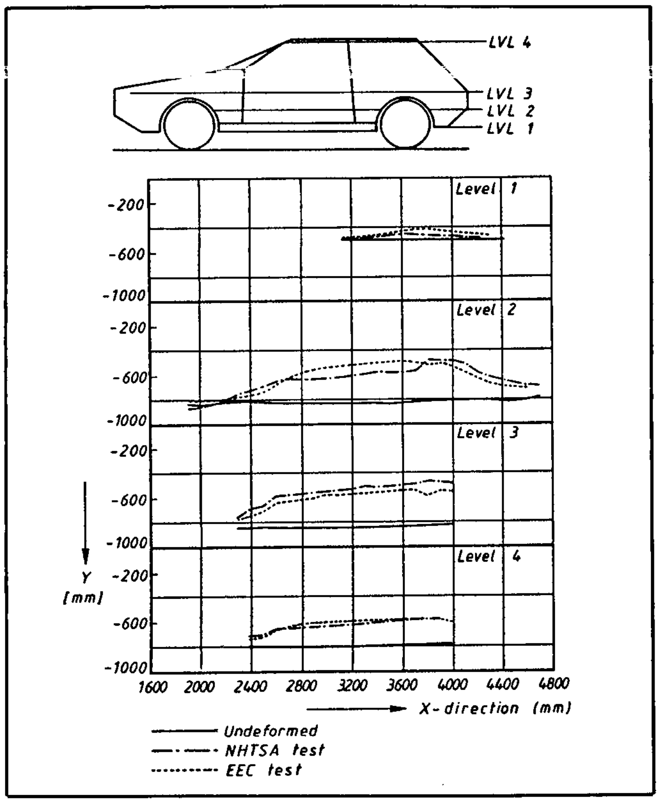

Figure 7. Vehicle (outside) side structure deformation, caused by NHTSA test and EEC test

Gross motion

During tests 1,2 and 5 (crabbed configuration) the dummy was in a free seating position. This was due to the forward dummy movement (relative to the chair), caused by the component of the velocity in longitudinal vehicle direction and which was possible due to the slack in the retractor system. This facilitated the

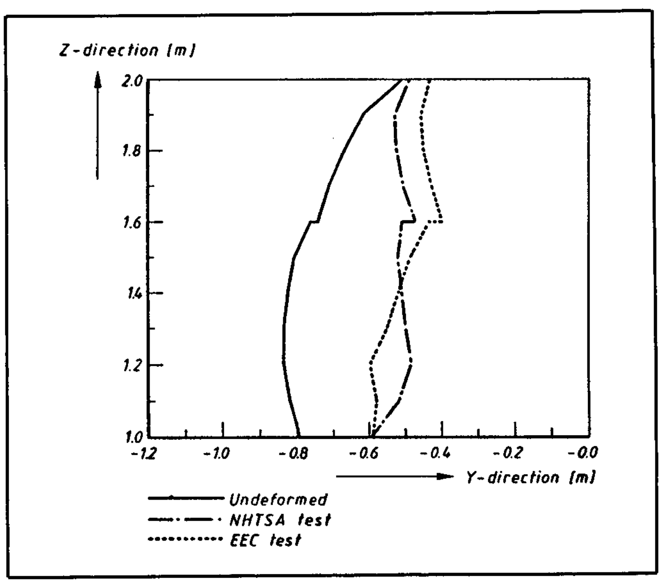

Figure 8. B-pillar (outside) deformation

thorax motion. Film analysis showed that when the safety belt restrained, the chest of the dummy started rotating around the diagonal part of the safety belt system. The SID rotated more than the EUROSID, where the arm assembly opposed this motion.

In test 1 the head of the SID came out of the side window and hit the front of the barrier. In test 2 the head hit the edge of the roof. In the tests 3,4 (EEC tests) the neck of the EUROSID was restrained by the diagonal part of the safety belt. Test 5 (EUROSID in crabbed configuration) showed rather a severe head impact on the edge of the roof.

Dummy measurements

The dummy test results are summarized in Table 5. Besides the accelerations and deflections, the Thoracic Trauma Index (TTI) and Viscous Injury Response (VxC) have been calculated.

The TTI is defined by the following equation [6]:

$$TTI = 0.5\ (ribg + T12g)$$

where: ribg = maximum absolute value of rib acceleration (g)

T12g = maximum absolute value of lower spine acceleration (g)

Both acceleration signals should be filtered according to the specifications (see Table 3 and 4).

The Viscous Injury Response is calculated as [7]:

$$V \times C = \max \left[\frac{d(D(t))}{dt} \times \frac{D(t)}{0.5 \times \text{torso width}} \right]$$

where: D = rib deflection, filtered (cfc 60) before and after differentiation.

Table 5. Dummy test results (values between parentheses represent the time in ms)

Test number	1	2	5	3	4
Test procedure	NHTSA	NHTSA	NHTSA	EEC	EEC
Dummy	SID	SID	EUROSID	EUROSID	EUROSID
Head accel. res. (g)	129.5(77)	97.3(47)	121.9(45)	40.3(51)	53.9(51)
HIC	1067	778	611	115	240
Upper spine lat (g)	-	-	67.3(39)	39.5(45)	45.8(37)
Upper rib lat (g)	78.2(37)	90.4(40)	173.6(31)	144.9(28)	112.6(30)
Middle rib lat (g)	-	-	95.5(35)	174.7(33)	190.8(34)
Lower rib lat (g)	68.6(37)	68.6(31)	92.9(27)	56.5(32)	64.5(33)
Lower spine lat (g)	179.5(30)	196.0(31)	98.7(32)	64.9(29)	60.1(31)
Thoracic Trauma Index	129*)	143*)	136	121	128
Chest deflection (mm)	23.3(59)	33.4(61)	-	-	-
Upper rib defl. (mm)	-	-	27.6(46)	40.2(56)	39.6(52)
Middle rib defl. (mm)	-	-	5.0(47)	20.0(51)	21.0(51)
Lower rib defl. (mm)	-	-	0.6(25)	1.5(39)	0.5(41)
V x C (m/s)	0.23	0.33	0.44	0.53	0.48
Abdomen switch cont.	-	-	yes***)	yes**)	no***)
Pelvis accel. lat (g)	121.7(23)	121.8(22)	133.2(24)	53.0(34)	57.1(31)
Ilium Force left (N)	-	-	1010(21)	868(28)	1260(32)
Pubic Symphysis (N)	-	-	16620(23)	3230(33)	4210(31)

*) value much lower when no lower spine penetration would occur (see text
**) switch has been preset to a force of 4 KN (see text)
***) switch has standard setting (4.5 KN)

Discussion

The difference in response of the SID during tests 1 and 2 (NHTSA procedure) is clearly illustrated by the head accelerations. In test 1 the dummy's head impacts the barrier front (peak acceleration 129.5 g at 77 ms) while in test 2 a much earlier impact with the roof is observed (peak acceleration 97.3 g at 47 ms). Close analysis of these tests learned that these differences most likely are attributed to a slight variation in initial positioning of the dummy. This indicates the need of maintaining very accurate dummy positioning procedures.

Like the SID, the EUROSID also showed the head impact during the NHTSA test (test 5). This is in contrast with the EEC tests, where no head impact was detected. Due to this, large differences in peak head accelerations and HIC values in the crabbed and non-crabbed tests can be observed.

Under similar circumstances (i.e. in the crabbed configuration) the SID in tests 1 and 2 registrates lower values for the rib accelerations compared to the EUROSID in test 5. An explanation for this is the significant difference in design of the thorax of both dummies: in the SID the ribs are mutually connected while moreover the total mass of the rib cage is increased in order to account for the missing arms; the EUROSID has three separate ribs. In contrast to this, the lower spine accelerations appear to be much higher for SID than for EUROSID. These high values for SID were caused by an adjustment mechanism in the seat back, which appears to penetrate the lower spine area. This penetration (which could be caused due to partly removing the covering of the seat back), was not detected by EUROSID, probably because of the difference of seating procedure and position of both dummies.

All tests show the same order of magnitude for TTI. In case of SID these relatively high values are due to the contributions of the high lower spine accelerations while in case of EUROSID (tests 3,4 and 5) the high rib accelerations appear to cause the large TTI values. It should be noted that the EUROSID is not designed to determine TTI.

In the SID an average chest deflection is determined, while the EUROSID measures chest deflections at three different locations. In the EUROSID the largest rib deflections are endured by the upper ribs. The lower ribs show only small deflections, since the impact of the deforming vehicle's side structure (including armrest) is mainly directed to the abdomen/pelvis area of the dummy causing pelvis displacement and leaving the lower chest part unaffected.

VxC values calculated for both the SID and the EUROSID show significant lower values for the SID since the chest deflection determined by the SID is an average value while that by EUROSID presented values related to local (i.e. largest) deflections.

In contrast to the SID, the EUROSID has been designed to detect abdominal injury. For this purpose the dummy is equipped with switches whose contact force is adjustable. A suggested force value is 4.5 KN. Since in earlier full scale tests, performed by other laboratories, abdominal switch contact never had occurred, in the first EEC test (test 3) the switches were preset to a lower value (4 KN), which indeed caused contact. In test 4 the switches were reset to their original value, which did not cause contact. In test 5 (NHTSA procedure), where the original setting of 4.5 KN was used, also switch contact was obtained.

Impact to the dummy pelvis area appears to be much more severe during the NHTSA (crabbed) tests than during the EEC (non-crabbed) tests, illustrated by much higher pelvis accelerations in both the SID and the EUROSID. The EUROSID shows an extreme high pelvis force (pubic symphysis) as well. The explanation for this is the much larger impact velocity of the vehicle's inside structure on the pelvis during the NHTSA tests.

The preceding analysis may be summarized by comparing the injury predictions resulting from both dummies. However, for both dummies no definitive injury criteria are available yet. Table 6 summarizes a comparison of injury predictions on the basis of potential (but rather preliminary) injury tolerance levels. The selected criteria are included in this table. In general, it can be seen that the NHTSA (crabbed) test procedure is more severe (i.e. causes higher injury levels) than the EEC (non-crabbed) test procedure.

Conclusions

Five side-impact tests with one vehicle type have been conducted using SID and EUROSID dummies as well as NHTSA (crabbed) and EEC (non-crabbed) moving deformable barriers. The results are based on one vehicle type and do not necessarily have to apply for other vehicles. Vehicle gross motion was described. Differences in local deformation as a result of the differences in test set-ups were presented. Dummy results were summarized and analyzed. Injury predictions were made.

Table 6. Injury predictions.

Test number	1	2	5	3	4
Test procedure	NHTSA			EEC	
Dummy	SID	SID	EUROSID	EUROSID	EUROSID
Head (HIC = 1000)	1067	778	611	115	240
Chest (SID :TTI = 80-100 g) (EUROSID:Deflection = 40-45 mm)	129	143	27.6	40.2	39.6
Abdomen (Force = 4.5 KN)	-	-	>4.5	>4.0	<4.5
Pelvis (Pubic force = 10 KN)	-	-	16.6	3.2	4.2

The vehicle gross motion and side deformations show good reproducibility in both test set-ups. The NHTSA side-impact barrier causes larger deformations in the lower vehicle area and, in general, higher injury levels.

In spite of the relatively high load level during the NHTSA as well as the EEC tests, no damage of SID and EUROSID was observed after the tests.

To obtain a reproducible dummy response, correct and accurate initial positioning of the dummy is required. In addition, in case of EUROSID, attention must be paid to the positioning of arms and hands.

Though different phenomena are responsible for the calculated TTI values (SID registrates high lower spine accelerations whereas EUROSID detects high rib accelerations), SID and EUROSID show TTI values of the same order of magnitude.

VxC values, calculated for both dummies, are significantly lower for the SID than for the EUROSID.

The injury prediction potential of the EUROSID exceeds the SID injury prediction potential. As a consequence, for vehicle design engineers more data are available for optimization purposes.

Based on preliminary injury tolerance levels a comparison has been made between SID and EUROSID injury predictions. More definitive requirements are urgently needed.

Acknowledgments

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Technical Session Two

Accident Investigation and Data Analysis

Chairman: Dr. Bernd Friedel, Federal Republic of Germany

Effects of Mandatory Seatbelt Laws on Traffic Fatalities in the United States

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Abstract

We examined state-specific and aggregate effects of U.S. legislation requiring the use of seatbelts among front-seat motor vehicle occupants. Effects of compulsory seatbelt use on the number of occupants fatally injured in traffic crashes were examined in the first eight states adopting such laws.

Monthly data on crash fatalities between January 1976 and June 1986 were analyzed using Box-Jenkins intervention analysis time-series methods. Because the new laws only apply to front-seat occupants, front-seat occupant fatalities were compared with: (1) rear-seat fatalities; (2) nonoccupant fatalities (motorcyclists, pedalcyclists, pedestrians); and (3) fatalities among front-seat occupants in neighboring states without compulsory seatbelt use. Exposure to risk of crash involvement was controlled by analyzing fatality rates per vehicle mile traveled.

Results revealed a statistically significant decline in the rate of front-seat fatalities of 8.7% in the first eight states with seatbelt laws. The fatality rate declined 9.9% in states with primary enforcement laws and 6.8% in states limited to secondary enforcement only. Rates of rear-seat and nonoccupant fatalities have not changed since the belt laws were implemented.

Introduction

Laws requiring the use of seatbelts were first implemented in Australia in 1971, and spread to a number of European countries, Canadian provinces, and other jurisdictions in the subsequent decade. In the mid-1980s, selected states in the United States implemented compulsory belt use laws. The objective of this study was to evaluate the effects of belt laws on motor vehicle fatality rates in the first eight U.S. states implementing such laws.

Numerous studies have found increased belt use and reduced rates of traffic fatalities following implementation of compulsory belt use laws. Although effects varied, rates of seatbelt use have typically doubled or tripled immediately after belt laws took effect, both in the United States (Table 1), and other countries (Table 2). After immediate dramatic increases in belt use at the time belt laws first took effect, many jurisdictions experienced some decay in use over the subsequent months or years. Estimated fatality reductions following implementation of compulsory belt use vary widely from country to country (from 0 to 80%; Table 3). Within the United States, preliminary estimates of the effect of belt laws on fatalities cluster much more narrowly in the range of 1 to 20% (Table 4). Many of these studies, especially the earlier ones, used nonrandom samples, inadequate control groups, and unreported analytic methods.

Methods

We evaluated the experience in eight U.S. states that implemented mandatory seatbelt use laws prior to October, 1985 using monthly data on traffic fatalities from January 1976 through June 1986. We used a longitudinal or **time-series design** to ensure that observed changes in fatalities were not due to long-term cycles or trends, or were not a result of a regression to the mean effect. In the absence of random assignment, time-series research designs have the highest possible levels of internal validity (Cook and Campbell, 1979).

To further strengthen causal inferences concerning the relationship between compulsory seatbelt laws and traffic fatalities, we examined **two types of control groups** one would not expect to be affected by the new laws. First, we paired each "experimental" state that recently implemented a seatbelt law with a neighboring "control" state that did not implement a belt law during the period under study. States analyzed include: New York with a belt law versus Pennsylvania without, New Jersey versus Maryland, Michigan versus Ohio, Illinois versus Indiana, Texas versus Georgia, Nebraska versus Kansas, Missouri versus Tennessee, and North Carolina versus Virginia.[1] Secondly, within the experimental states we examined two categories of traffic fatalities not di-

[1] Two of the comparison states, Ohio and Tennessee, implemented compulsory belt use laws in the spring of 1986. Analyses involving these states were limited to the period in which no belt law was in effect.

Table 1. Effects of US seatbelt laws on restraint use.

Source	Jurisdiction	Effective Month	Month Observed	Use Rate
Rood & Kraichy (1985)	NY	12/84	10/84	16%
			4/85	57%
			9/85	46%
Williams & others (1986)	NY	12/84	1/85	69%
			4/85	60%
			2/86	44%
Pace & others (1986)	NY	12/84	4/85	63%
			4&7/86	37%
Brick & others cited in Williams & others (1986)	NJ	3/85		18%
			7/85	40%
Williams & others (1986)	NJ	3/85	11/84	16%
			3/85	51%
			7/85	44%
			4/86	38%
Mortimer (1986)	IL	7/85	4/85	16%
			7/85	40%
			12/85	35% (Drivers)
			1/86	29%
			3/86	32%
			6/86	34%
Wagenaar & others (1987a)	MI	7/85	12/84	18%
			4/85	25%
			7/85	61%
			12/85	44%
			4/86	44%
			7/86	47%
			12/86	44%
Hatfield & others (1985)	TX	9/85	1-6/85	15%
Bunch & others (1986)	TX	9/85	1-6/86	66%
Dept. of Highways (cited in Campbell & others, 1986)	TX	9/85	3/86	75%
Office of Highway Safety (cited in Campbell & others, 1986)	NE	9/85	11/85	26%
			11/85	44%
			2/86	38%
IIHS (1987)	NE	9/85	2/87	29%
Missouri Safety Center (cited in Campbell & others, 1986)	MO	10/85	7/85	12%
			10/85	19%
Campbell & others (1986)	NC	10/85	9/85	25%
			11/85	44%
			1/86	42%
			3/86	45%
			5/86	48%
IIHS (1987)	NC	10/85	2/87	78%

Table 2. Effects on non-U.S. seatbelt laws on restraint use[1].

Table 2. continued.

Source	Jurisdiction	Effective Month	Month/Year Observed	Use Rate	Comments
Vulcan (1977)	Australia-Victoria	12/70	5/71	32-48%	Front seat occupants
			2/72	47-60%	
			2/73	52-65%	
			2/74	67-79%	
			2/75	73-79%	
			2/76	73-88%	
Joubert (1979)	Victoria	12/70	pre-law	18%	Observation date not cited
			post-law	64%	Rural
			post-law	75%	Urban
Manders (1984)	Melbourne	12/70	11/82	95%	Drivers
			3/84	96%	Drivers
Johinke (1977)	Adelaide South Australia	12/71	10/71	23%	Occupants with belts available
			10/72	78%	
			10/75	66%	
			mid '76	84%	
Crinion & others (1975)	Adelaide S.A.	12/71	10/64	8%	All seating positions
			10/70	14%	
			10/71	23%	
			10/72	78%	
Road Traffic Bd. (1983)	Adelaide S.A.	12/71	11/82	91%	Drivers
			11/82	85%	Ft-seat pass
			11/82	61%	Rr-seat pass
Seeney (1977)	Queensland	1/72	12/72	90%	
Schnerring (1983)	New South Wales-Sydney Metro	10/71	4/71	30%	Drivers
			10/71	60%	Drivers
			11/71	76%	Drivers
			12/72	89%	Drivers
			12/73	91%	Drivers
			7/75	91%	Drivers
			7/77	91%	Drivers
			1979	89%	Drivers
			7/81	84%	Drivers
Marburger (1986)	Austria	7/76	pre-law	5-10%	Urban
			pre-law	20-25%	Rural
			post-law	10-15%	Urban
			post-law	40%	Urban
			9/84	81%	Urban
			9/84	82%	Rural
			8/85	81%	Urban
			8/85	82%	Rural
Fisher (1980)	Belgium	6/75	pre-law	17%	
			post-law	87%	
Marburger (1986)	Belgium	6/75	11/84	70%	Rural
			11/84	60%	Urban
Rockerbie (1983)	Canada-British Columbia	10/77	pre-law	20-24%	
			post-law	50%	
B.C. Research (1983)	British Columbia	10/77	4/83	55%	
			6/83	67%	
Arora (1985)	British Columbia	10/77	11/83	67%	
Arora (1985)	New Brunswick	9/83	11/82	4%	Drivers
			11/83	66%	Drivers
N.B. Dept Transp. (1984)	New Brunswick	9/83	8/84	73%	
Arora (1982)	Newfoundland	7/82	11/81	9%	
			11/82	68%	
Arora (1985)	Newfoundland	7/82	11/83	76%	Urban
			11/83	61%	Rural
Murray (1984)	Newfoundland	7/82	7/84	74%	
Arora (1985)	Manitoba	1/84	11/83	11%	
			11/84	62%	
DataCom (1984)	Manitoba	1/84	6/84	79%	Drivers
Snow (1979)	Ontario	1/76	12/75	21%	
			2/76	77%	
			6/76	50%	
Pierce (1979)	Ontario	1/76	5/77	50%	Increased enforcement mid-77
			5/78	65%	
Mathews (1982)	Ontario	1/76	9-12/80	49%	
Arora (1985)	Ontario	1/76	11/82	49%	
			11/83	60%	
Jonah & Lawson (1986)	Ontario	1/76	5/84	70%	
Stulginskas & Pless (1983)	Montreal	8/76	5-6/76	15%	Drivers
			5-6/77	33%	Drivers
			5-6/78	45%	Drivers
			5-6/81	56%	Drivers
Arora (1985)	Quebec	8/76	11/82	68%	Drivers
			11/83	60%	Drivers
Regie de l'assurance Automobile du Quebec in Jonah & Lawson (1986)	Quebec	8/76	6/83	60%	Urban Drivers
				74%	Freeway Drivers
Simpson & Warren (1981)	Saskatchewan	7/77	pre-law	26%	Drivers
			post-law	78%	Drivers
Sheils (1978)	Saskatchewan	7/77	5/77	24%	
			7/77	65%	
			10/77	73%	
			5/78	60%	
Bergen & others (1979)	Saskatchewan	7/77	7/77	52%	Drivers
			10/77	70%	Drivers
			5/78	55%	Drivers
			5/79	70%	Drivers
Arora (1982,1985)	Saskatchewan	7/77	11/80	56%	Drivers
			11/81	49%	Drivers
			11/82	48%	Drivers
			11/83	54%	Drivers
Marburger (1986)	Denmark	1/76	pre-law	19%	
			post-law	74%	
Ashton & others (1983)	England	2/83	1/83	43%	
			5/83	95%	
Mackay (1984a,1984b)	England	2/83	2/83	90%	
Oranen (1977)	Finland	7/75	pre-law	15-20%	
			6/75	30%	Highway
			6/75	9%	Urban
			8/75	68%	Highway
			8/75	53%	Urban
			7/76	64%	Highway
			8/76	37%	Urban
Berard-Andersen in Fisher (1980)	Finland	7/75	pre-law	8%	Urban
			pre-law	31%	Rural
			post-law	38%	Urban
			post-law	66%	Rural
Oranen and Koivurova (1977)	Finland	7/75	8-9/78	71%	Highways
			8-9/78	41%	Urban
Marburger (1986)	Finland	7/75	4/82	87%	Urban
			4/82	86%	Rural
			9/83	82%	Urban
			9/83	92%	Rural
Fisher (1980)	France	7/73	pre-law	20-25%	
			7/73	80%	
			10/73	50%	
Chodkiewicz & Dubarry (1977)	France	7/73	1972	20%	
			1973	26%	
			1974	67%	
			1975	80%	
Gerondeau (1979)	France	7/73	7/73	80%	Law applied to rural only
			11/73	50%	
			early '74	80%	
			1979	70-75%	
Gerondeau (1981)	France	7/73	1974	54%	Law applied to rural only
			1975	76%	
			1976	79%	
			1977	72%	
			1978	67%	Law expanded to all rds in 10/79
			1979	69%	
			1980	79%	
Hartemann & Others (1984)	France	7/73	1982	95%	Highways
			1982	75%	Other non-urban
Hearne (1981)	Ireland	2/79	Fall '78	19%	Drvs Nat'l rds
			Fall '78	9%	Drvs other roads
			Sum '79	46%	Drvs nat'l rds
			Sum '79	38%	Drvs nat'l rds

Table 2. continued.

Source	Jurisdiction	Effective Month	Month/Year Observed	Use Rate	Comments
Hakkert & others (1981)	Israel	7/75	pre-law	6%	
			8/75	77%	
			1976	83%	
			1977	70%	
Fisher (1980)	Netherlands	6/75	1974	11%	Urban
			1974	24%	Rural
			7/76	58%	Urban
			7/76	75%	Rural
Vaaje (1986)	Netherlands	6/75	1983	46%	Urban
			1983	65%	Rural
Toomath (1977)	New Zeland	6/72	5/72	40%	
			6/72	87%	
			1974	83%	
			1975	89%	
Fisher (1980)	Norway	9/75	pre-law	15%	Urban
			pre-law	37%	Rural
			1976	28%	Urban
			1976	59%	Rural
			1977	30%	Urban
			1977	63%	Rural
Oranen & Koivurova (1980)	Norway	9/75	3/80	74%	Urban
			3/80	90%	Rural
Fisher (1980)	Puerto Rico	1/74	7/73	5%	
			5/74	24%	
			9/74	7%	
			1/76	34%	
			5/77	14%	
Fernie (1980)	South Africa	12/77	11/77	18%	
			3/78	62%	
			9/79	70%	
Bohlin (1979)	Sweden	1/75	1974	35%	
			1975	84%	
Tingvall in Fisher (1980)	Sweden	1/75	1974	36%	
			1978	79%	Urban
			1978	87%	Rural
Norin & others (1984)	Sweden	1/75	1983	80%	
Fisher (1980)	Switzerland	1/76	pre-law	35%	
			2/76	95%	Expressway drivers
			2/76	92%	Rural drivers
			2/76	89%	Urban drivers
			9/78	64%	Expressway drivers
			9/78	46%	Rural drivers
			9/78	33%	Urban drivers
Andreasson (1983)	Switzerland	7/81[2]	1982	77%	Expressways
			1982	76%	Rural
			1982	62%	Urban
Federal Inst. Streets in Fisher (1980)	W. Germany	1/76	8/75	28%	
			11/75	32%	
			1/76	50%	
			3/77	46%	
			9/77	48%	
			9/78	58%	
Marburger (1986)	West Germany	1/76	9/84	92%	Fines begin
			3/86	94%	August 1985

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1. Drivers and front-seat passengers unless otherwise noted.
2. Switzerland's 1976 law declared invalid by by the Supreme Court in 9/77 and reinstated by the Government on 7/1/81.

rectly affected by the new laws—rear-seat occupants and nonoccupants (including pedestrians, motorcyclists, and pedalcyclists).

Data Collection

All fatality data were based on the Fatal Accident Reporting System maintained by the U.S. National Highway Traffic Safety Administration. Monthly counts of the number of fatalities were calculated separately within each state for front-seat occupants, rear-seat occupants, and nonoccupants. Occupant fatalities included only those traveling in passenger cars, vans, light trucks, and utility vehicles. Medium and heavy trucks, buses, and a variety of special vehicles were excluded because some are exempted from provisions of the seatbelt laws and others were covered by pre-existing regulations requiring seatbelt use. All analyses were limited to persons age 10 and over because compulsory restraint use laws for young children were implemented several years before the adult seatbelt laws took effect. Although most child restraint laws are limited to those under age 4,

Table 3. Effects of non-U.S. seatbelt laws on fatalities.

Investigators	Jurisdiction	Month Effective	Post-Law Months	Fatality Change	Significance Level
Foldvary & Lane (1974)	Australia-Victoria	1/71	9	-15%	≤0.05
Trinca & Dooley (1977)	Victoria	1/71	48	-37%	
Andreassand (1976)	Victoria	1/71	10	-15%	
Joubert (1979)	Victoria	1/71	12	-15%	
McDermott & Hough (1979)	Victoria	1/71	84		≤0.01
Trinca (1984)	Victoria	1/71	144	-60%	
Johinke (1977)	Queensland	1/72		-14%	
Bhattacharyya & Layton (1979)	Queensland	1/72		-46%	
Crinion (1975)	South Aust.	11/71	12	-8%	
Fisher (1980)	Australia			-20%	
Snow (1979)	Canada-Ontario	1/76	6	-13%	
Sheils (1978)	Saskatch.	7/77	12	-20%	
Jonah & Lawson (1984)	Brit. Col.	10/77	51	-30%	≤0.01
	Ontario	1/76	72	-26%	≤0.05
	Quebec	8/76	65	-17%	ns
	Saskatch.	7/77	52	-37%	≤0.10
				-35%	≤0.05[1]
Hedlund (1986)	Brit. Col.	10/77	60	-52%	
	Ontario	1/76	72	-12%	
	Quebec	8/76	65	-18%	
	Saskatch.	7/77	52	-29%	
Nordic Traffic Safety Council (1984)	Denmark	1/76	12	-1%	
Hedlund (1986)	Denmark	1/76	12	-13%	
Mackay (1984b)	England	2/83	11	-25%	
Pye & Waters (1984)	England	2/83	3	-80%	≤0.001
Durbin & Harvey (1985)	England	2/83	23	-18%[2]	
				-25%[3]	
Chodkiewicz & Dubarry (1977)	France	7/73	30	-21%	
Hartemann & others (1984)	France	7/73	114	-50%	
Hearne (1981)	Ireland	2/79	11	-0.7%	
Hakkert & others (1981)	Isreal	7/75	30	-42%[2]	
				-44%[3]	
Hedlund (1986)	Isreal	7/75	30	-41%	
Toomath (1977)	New Zealand	6/72	24	+3%[4]	
Hedlund (1986)	New Zealand	6/72	24	-43%	
McCarthy & others (1984)	Norway	9/79		-21%	
Nordic Traffic Safety Council (1984)	Norway	9/79	12	-10%	
Hedlund (1986)	Norway	9/75	12	-29%	
McCarthy & others (1984)	South Africa	12/77		-8%	
McCarthy & others (1984)	Sweden	1/75		-14%	
Bohlin (1979)	Sweden	1/75	12	-12%	
Norin & others (1984)	Sweden	1/75	12	-12%	
Fisher (1980)	Switzerland	1/76	12	-12%	
Hedlund (1986)	Switzerland	7/81[5]	12	-15%	
	W. Germany	1/76	6[6]	-25%	

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1. Crash data for the period 7/1/77 to 12/31/82 were analyzed; actual fatality rates were significantly different from predicted rates only in 1980 ($p \leq 0.10$) and 1981 ($p \leq 0.05$)
2. Drivers only
3. Front-seat passengers
4. In contrast, nonoccupant fatalities increased almost 40% during this period
5. Switzerland's 1976 law declared invalid by by the Supreme Court in 9/77 and reinstated by the Government in 7/1/81
6. This study compared the pre- and post-fine period 1-6/84 to 1-6/85

spill-over effects on older children of those earlier laws have been reported (Wagenaar and Webster, 1986). The length of the resulting time series varied from 107 baseline months in New York to 117 baseline months in North Carolina, and from 9 post-law months in North Carolina to 19 post-law months in New York.

Exposure to risk of crash involvement was controlled by dividing all of the fatality frequency time-series by the number of vehicle miles traveled (VMT)

Table 4. Effects on U.S. seatbelt laws on fatalities.

Investigators	Jurisdiction	Effective Date	Post-Law Months	Fatality Change	Significance Level
Lund & others (1986a)	NY	12/84	9	-9%	≤0.05
Hedlund (1986)	NY	12/84	9	-15%	
Latimer & Lave (1986)	NY	12/84	6	-20%	
Pace & others (1986)	NY	12/84	3		-27%
Wagenaar & others (1987b)	MI	7/85	12	-10%	ns
Mortimer (1986)	IL	7/85	9	-3%	
Lund & others (1986b)	NY	12/84	13	-5%	≤0.01
	NJ	3/85	10	-4%	ns
	MI	7/85	6	-4%	ns
	IL	7/85	6	-7%	ns
Campbell & others (1986)	NY	12/84	13	-8%	≤0.10
	NJ	3/85	10	-6%	≤0.10
	IL	7/86	6	-9%	ns
	MI	7/85	6	-16%	≤0.10
	TX	9/85	4	-18%	≤0.01
	NB	9/85	4	-11%	ns
	MO	9/85	3	+5%	ns
	NC	10/85	3	-0.4%	ns
	All			-10%	ns
Hoxie & Skinner (1987)	NY	12/84	19	-7%	ns
	NJ	3/85	16	-2%	ns
	IL	7/85	12	-1%	ns
	MI	7/85	12	-14%	≤0.10
	TX	9/85	10	-18%	≤0.01
	MO	9/85	9	+18%	≤0.01
	NC	10/85	9	-5%	ns
	All			-6%	ns
Partyka (1987)	All[1]	Variable[2]	Variable[2]	-7%	

1. NY, NJ, IL, MI, TX, NB, MO, NC

2. Different for each state depending on date law enacted.

within each of the states under study. The resulting rates of fatalities per VMT were used in all subsequent analyses. State-specific VMT figures by month were obtained from the U.S. Federal Highway Administration and are based on traffic counter and motor fuel sales data.

Statistical Methods

Ordinary least squares regression and other commonly used statistical procedures were not appropriate for the present study because they assume independent observations. However, a series of observations over time, such as the fatality rate series analyzed here, are highly autocorrelated, violating the assumption of independence, and leading to biased standard error estimates using conventional methods. Therefore, we used the time-series intervention analysis methods of Box and Jenkins (1976). On a conceptual level, the analytic strategy involves explaining as much of the variance in fatality rates on the basis of the past history of those rates, before attributing any of the variance to an exogenous variable, such as implementation of a seatbelt law. This approach of intervention analysis was particularly appropriate for the current study, because the objective was to identify significant shifts in fatality rates associated with seatbelt laws, independent of observed regularities in the history of each series.

Baseline Auto-regressive Integrated Moving Average (ARIMA) models were iteratively developed for each time series, repeatedly going through cycles of specifying a model, estimating it, and evaluating its adequacy in terms of accounting for all significant autocorrelation patterns in the series. All of the time series were natural-logarithm transformed prior to parameter estimation to reduce heteroscedasticity. All of the final models met the multiple criteria for model adequacy identified by Box and Jenkins (1976), including significant noise model parameters, low correlations among parameters, and insignificant residual autocorrelations.

Transfer functions were added to the noise models to test for effects of seatbelt laws. Given the short post-law period for which data were available, simple shift transfer function models were used to represent potential effects of the belt laws. Additional transfer functions were added to the models for selected time series. The substantial decline in the fatality rate in 1982 in most of the states was controlled by including a simple shift transfer function. The 1982 decline was due to a variety of factors, including a major economic recession, campaigns to reduce alcohol-impaired driving, and changes in the age structure of the population (Hedlund and other, 1984). Our objective was not to fully elucidate the causal structure underlying those fatality reductions, but rather to statistically control for those reductions when estimating the effects of recent compulsory seatbelt laws.

Because the models are intrinsically nonlinear, the Gauss-Marquardt backcasting algorithm implemented in the software package BMDP2T was used to estimate the parameters (Dixon and others, 1983). All parameter estimates in the logarithm metric were converted to an estimated percent change in the series after the seatbelt law, from levels expected given baseline patterns, using: $(e^{\omega}$ -1)100. A plot of each time series analyzed, along with the final statistical model for each series are shown in the Appendix. Major findings are briefly reviewed here.

Results

Significant declines in the rate of front-seat occupant deaths per VMT occurred in three of the eight states with mandatory seatbelt laws (Figure 1). The fatality rate declined 8.3% in Michigan, 12.4% in New York, and 15.5% in Texas. Intervention parameter estimates were in the expected direction (though not significant) in New Jersey and Illinois. The fatality rate increased in Missouri, Nebraska, and North Carolina, but only in Missouri was the estimated increase larger than two standard errors.[2] While these analyses control for long-term trends and cycles within each state, and control for changes in exposure via rates per VMT, these state-specific

[2] Technically not statistically significant because we hypothesized a fatality reduction following implementation of belt laws, and consequently used one-tailed tests.

changes in fatalities may still simply reflect broader regional or national changes due to other factors. To ensure that observed fatality changes were associated with the seatbelt laws and not other factors, we analyzed the rate of fatalities per VMT in a state with a new belt law **relative to** the rate of fatalities per VMT in a neighboring state without a belt law during the period studied. In other words, the fatality rate in the target state was divided by the rate in the comparison state.

Analyses of the relative rates again indicated significant declines in fatalities associated with seatbelt laws in three of the eight states: New York, 7.1%; New Jersey, 24.5%; and Nebraska, 19.3%; (Figure 2). However, two of these three (New Jersey and Nebraska) showed no significant decrease when examining the state alone, without taking into account the experience in comparison states. In addition to significant reductions in the relative rates of fatalities in three states, time-series modeling produced estimates in the expected direction (although not significant) in an additional four states.

Clearly the small number of post-law data points available (9 to 19 months), and the substantial baseline variability in fatality rates over time result in moderately large standard errors and what appear to be inconsistent results across states. To reduce this background variation, we combined the eight belt-law states, and estimated the aggregate effect of the belt laws in these eight states. The state-specific time series were aligned on the month each state's belt law took effect and the number of fatalities and amount of vehicle mileage traveled were summed. The eight comparison states were similarly summed. The result was time series in which each month no longer represented a specific month in time, but rather represented the ordinal month from the point at which belt laws were implemented. Dividing the fatality rate per VMT for the belt-law states as a group by the fatality rate per VMT for the comparison states as a group resulted in an aggregate relative rate. Time-series modeling of the aggregate relative rate estimated a statistically significant 8.7% decline associated with belt laws in these states.

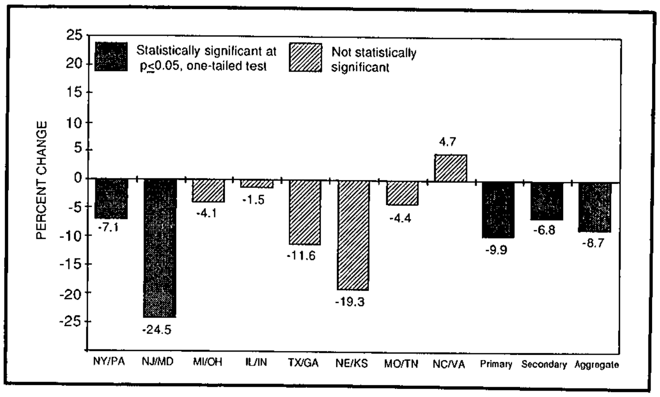

Figure 2. Percent change in rate of fatalities per VMT among front-seat occupants age 10 and over: belt law relative to comparison states

One obvious possible explanation for the differential effects of seatbelt laws across states is the size of the change in belt use caused by the law. Most states experienced an increase in belt use form about 16% before to a 45% a few months after laws took effect (Table 1). Texas had a larger than average increase in belt use (from 15 to 66%) and Illinois a smaller than average increase (from 16 to 30%). Given the different survey methods used in each state, and the standard errors of our estimates of belt law effects on fatalities, we are not in a position to argue that cross-state differences in our fatality reduction estimates reflect differences in belt use rates across states.

Nevertheless, specific provisions of the law, such as primary versus secondary enforcement,[3] and the intensity with which it is enforced are expected to influence belt use rates. To take into account these major differences in the laws across states, we conducted time-series analyses of two groups of states—states with primary enforcement versus states with secondary enforcement only. Because North Carolina and Missouri were not actively enforcing their laws during this period they were excluded from these analyses. Results indicated a significant 9.9% fatality reduction in the primary enforcement states, and a significant 6.7% fatality reduction in secondary enforcement states (Figure 2). As expected, states with primary enforcement experienced larger fatality reductions than states limited to secondary enforcement. However, it's worth noting that clear benefits also accrued

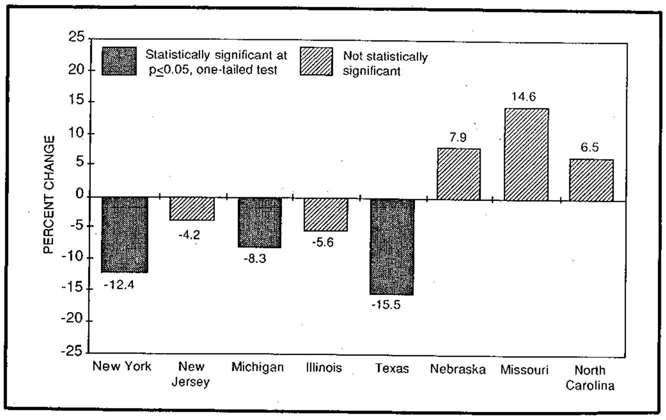

Figure 1. Percent change in rate of fatalities per VMT associated with seatbelt laws: front-seat occupants age 10 and over

[3] Secondary enforcement means that police officers may only issue citations for failure to use belts if the motorist is first stopped for some other offense. That is, a motorist may not be stopped solely for failure to use seatbelts.

from secondary enforcement belt laws, provided citations were actually issued to violators.

Finally, in addition to controlling for other plausible explanations for observed fatality declines by including comparison states, we conducted time-series analyses of rates of rear-seat occupant deaths and nonoccupant (motorcycle, pedalcycle, and pedestrian) deaths in the states with seatbelt laws. All of the laws examined here are limited to front-seat occupants; as a result, rear-seat occupants along with nonoccupants serve as useful comparison groups. Analyses of aggregate fatality rates for the eight belt-law states revealed no change in fatality rates among either rear-seat occupants or nonoccupants.

Discussion

Our results confirm that laws requiring seatbelt use can significantly reduce rates of motor vehicle fatalities. However, one cannot expect the fatality declines to be clearly demonstrable within single jurisdictions a short time after the laws are implemented. The nature of fatality trends over time, and the amount of unpredictable variation in number of deaths from month to month means that a minimum of a 6 to 10% reduction over a 6 to 12 month period is required before the reduction can be reliably identified. Despite the lack of statistical significance for the estimated effects of seatbelt laws in some jurisdictions, it is premature to conclude that laws in those states had **no** effect. As additional data become available, increasing the statistical power of analytic techniques used, some of the state-specific estimates obtained in the current study may become statistically significant. Results from our most powerful analyses, those involving aggregate effects across several states, clearly demonstrate significant fatality reductions. Moreover, use of comparison states and comparison groups not directly affected by the seatbelt laws increases our confidence in interpreting the observed declines as caused by the mandatory seatbelt use laws.

In terms of the magnitude of the effects of compulsory belt use laws, one can expect a U.S. law that permits primary enforcement and is actually enforced at moderate levels will result in about a 10% reduction in traffic fatalities. A law that permits secondary enforcement only or is enforced at very low levels will have smaller effects. Although some advocates of compulsory seatbelt use have indicated that substantially larger declines in traffic fatalities would result, a 10% decline in a leading cause of death for the entire population represents a resounding public policy success. How many major programs aimed at reducing disease and injury can document an immediate 10% decline in mortality due to that cause of death across an entire population of millions of people? Moreover, effective implementation of a compulsory seatbelt use policy requires minimal expenditure of resources when compared to efforts to reduce mortality attributable to other leading causes of death (e.g., cardiovascular disease, cancer).

Despite the clear success of compulsory seatbelt laws to date, much more remains to be done. As noted earlier, belt use in the U.S. typically peaks within a month or two of implementation of belt laws, partially decaying after that point. Special enforcement efforts not only can arrest that decline, but further increase belt use rates, at least temporarily (Williams and others, in press). Clearly, our results demonstrate that belt laws with primary rather than secondary enforcement provisions are needed. We believe that rigorous enforcement of a primary seatbelt law in the U.S. can achieve and maintain belt use rates of approximately 60%, in contrast to less than 20% under the most favorable conditions (extensive education and public information programs) without compulsory use.

Even if asymptotic belt use of 60% were achieved throughout the U.S., declines in traffic fatalities of more than 20% are extremely unlikely. This is because of the differential between belt users and nonusers; that is, those at highest risk of involvement in serious traffic crashes are least likely to use belts. Therefore, other avenues of reducing traffic crash-induced injury and death that do not require action on the part of each individual driver (such as airbags) must be pursued simultaneously with efforts to implement and enforce mandatory belt use laws.

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Appendix

Combination autoregressive integrated moving average/transfer function models used to assess the statistical significance and magnitude of hypothesized effects of seatbelt laws were of the general form: $(1 - \phi_1 B - \ldots \phi_p B^p)(1 - \Phi_1 B - \ldots \Phi_P B^{Ps})(1 - B)^d (1 - B^s)^D \, \mathrm{Ln} Y_t = \alpha + (1 - \theta_1 B - \ldots \theta_q B^q)(1 - \Theta_1 B^s - \ldots \Theta_Q B^{Qs}) u_t + \psi X_t + \omega I_t$, where B is the backshift operator such that $B(z_t)$ equals z_{t-1}, ϕ_1 to ϕ_p are the regular autoregressive parameters, Φ_1 to Φ_P are seasonal autoregressive parameters, d is the order of nonseasonal differencing, D^P is the order of seasonal differencing, s is the seasonal span, $\text{Ln}Y_t$ is the natural logarithm transformation of the dependent time series, α is a constant, q is the order of the moving average process, θ_1 to θ_q are regular moving average parameters, Θ_1 to Θ_Q are seasonal moving average parameters, and u_t is a random error component. The two intervention components added to the autoregressive integrated moving average model are ΨX_t and ωI_t, where Ψ and ω are parameters to be estimated. X_t is a step function with the value 1 beginning at month t and 0 otherwise. I_t is a step function with the value 1 beginning at month t and 0 otherwise.

A plot of each major variable and the final statistical model are included here. Standard errors are shown in parentheses below each parameter estimate.

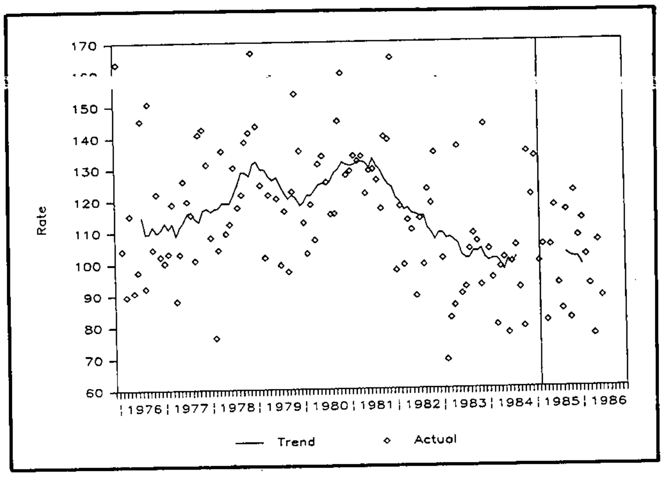

Figure A.2. Rate of New Jersey front-seat fatalities age 10 and over per 10 billion VMT

Final Model

$$(1\text{-}B^{12})\,LnY_t = \underset{(.028)}{(1\text{-}\ .892B^{12})u_t} - \underset{(.029)}{(1\text{-}B^{12})\ .155X_{73}} - \underset{(.039)}{(1\text{-}B^{12})\ .043I_{111}}$$

Adjusted $R^2 = .39$

Effect of New Jersey seat belt law, March 1985-June 1986: -4.2%

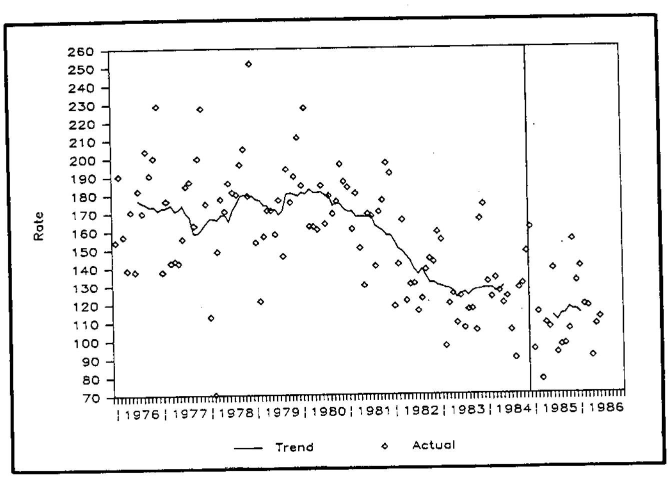

Figure A.1. Rate of New York State front-seat fatalities age 10 and over per 10 billion VMT

Final Model

$$(1\text{-}B^{12})\,LnY_t = \underset{(.028)}{(1\text{-}\ .876B^{12})u_t} - \underset{(.032)}{(1\text{-}B^{12})\ .283X_{73}} - \underset{(.041)}{(1\text{-}B^{12})\ .133I_{108}}$$

Adjusted $R^2 = .59$

Effect of New York seat belt law, December 1984-June 1986: -12.4%*

* $p<0.05$, one-tailed test.

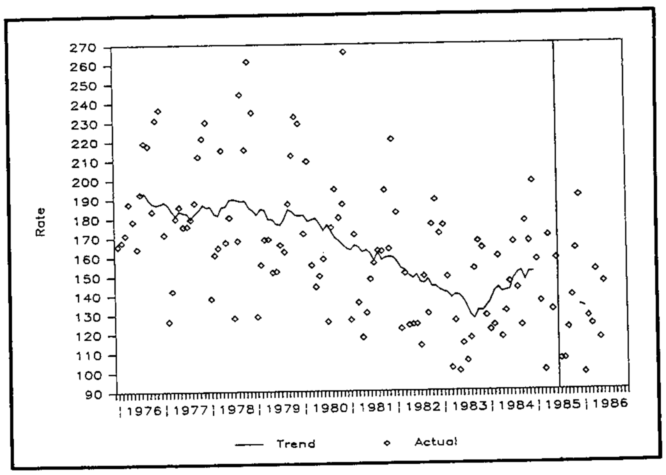

Figure A.3. Rate of Michigan front-seat fatalities age 10 and over per 10 billion VMT

Final Model

$$(1\text{-}B^{12})\,LnY_t = \underset{(.031)}{(1\text{-}\ .884B^{12})u_t} - \underset{(.030)}{(1\text{-}B^{12})\ .213X_{73}} - \underset{(.043)}{(1\text{-}B^{12})\ .086I_{115}}$$

Adjusted $R^2 = .54$

Effect of Michigan seat belt law, July 1985-June 1986: -8.3%*

* $p<0.05$, one-tailed test.

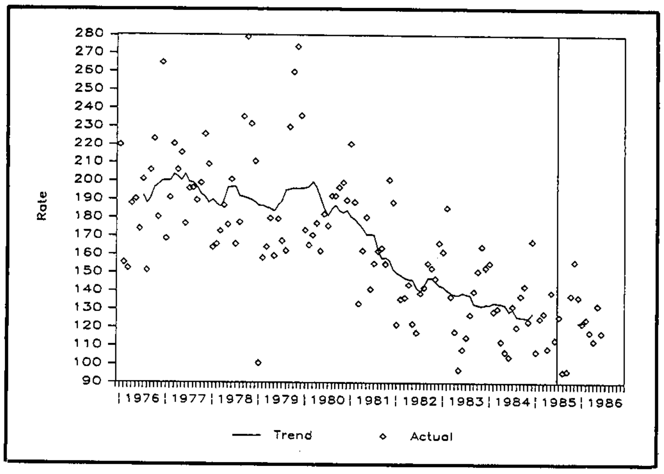

Figure A.4. Rate of Illinois front-seat fatalities age 10 and over per 10 billion VMT

Final Model

$$(1-B^{12})(1+\underset{(.081)}{.408}B^{12})\ (1-\underset{(.082)}{.207}B)\ LnY_t = u_t - (1-B^{12})\ \underset{(.051)}{.239}X_{73} - (1-B^{12})\ \underset{(.057)}{.057}I_{115}$$

Adjusted $R^2 = .55$

Effect of Illinois seat belt law, July 1985-June 1986: -5.6%

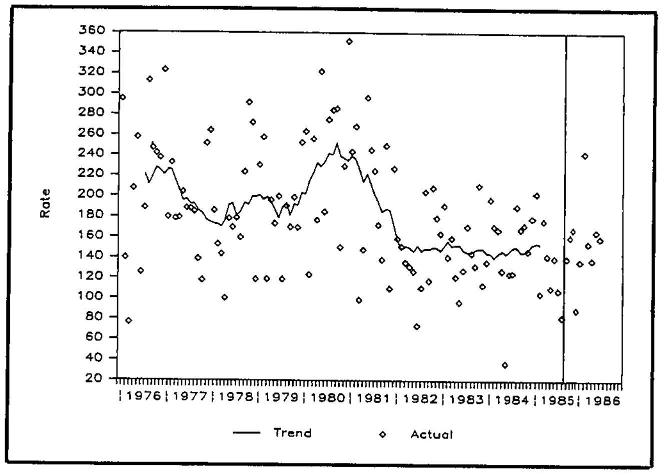

Figure A.6. Rate of Nebraska front-seat fatalities age 10 and over per 10 billion VMT

Final Model

$$(1-B^{12})\ LnY_t = (1-\underset{(.029)}{.869}B^{12})u_t - (1-B^{12})\ \underset{(.059)}{.337}X_{73} + (1-B^{12})\ \underset{(.098)}{.076}I_{117}$$

Adjusted $R^2 = .23$

Effect of Nebraska seat belt law, September 1985-June 1986: +7.9%

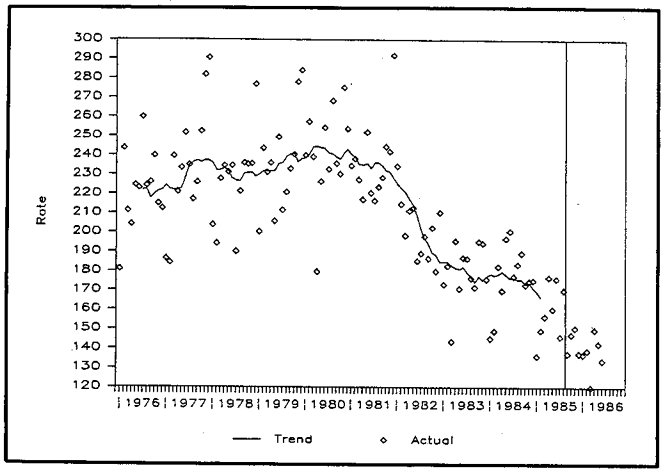

Figure A.5. Rate of Texas front-seat fatalities age 10 and over per 10 billion VMT

Final Model

$$(1-B)(1-B^{12})\ LnY_t = (1-\underset{(.028)}{.877}B^{12})\ (1-\underset{(.053)}{.800}B)u_t - (1-B)(1-B^{12})\ \underset{(.058)}{.088}X_{73}$$

$$-\ (1-B)(1-B^{12})\ \underset{(.053)}{.168}I_{117}$$

Adjusted $R^2 = .76$

Effect of Texas seat belt law, September 1985-June 1986: -15.5%*

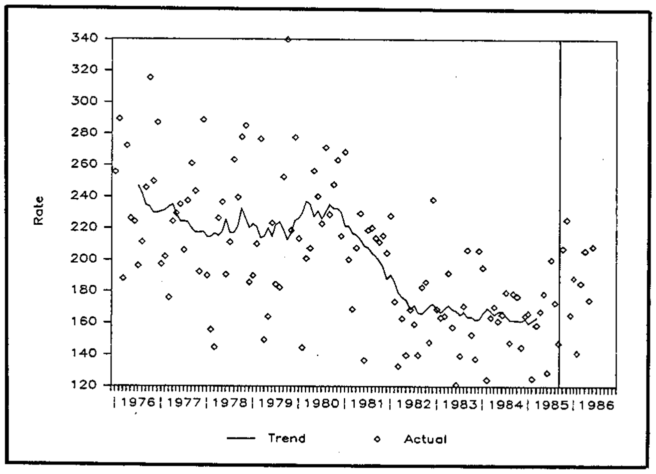

Figure A.7. Rate of Missouri front-seat fatalities age 10 and over per 10 billion VMT

Final Model

$$(1-B^{12})\ LnY_t = (1-\underset{(.029)}{.881}B^{12})u_t - (1-B^{12})\ \underset{(.030)}{.288}X_{73} + (1-B^{12})\ \underset{(.052)}{.137}I_{118}$$

Adjusted $R^2 = .47$

Effect of Missouri seat belt law, October 1985-June 1986: +14.6%

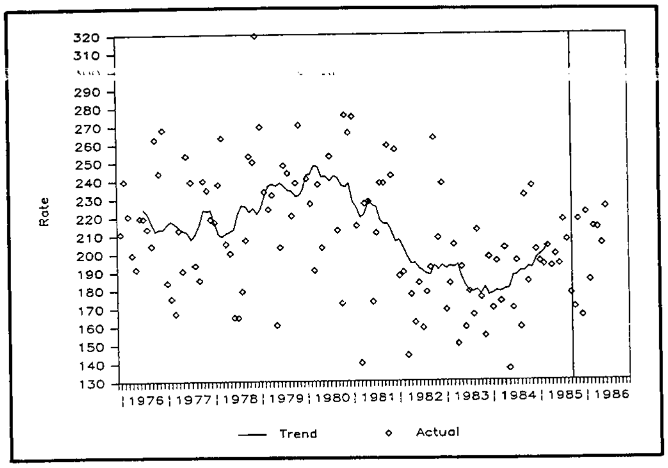

Figure A.8. Rate of North Carolina front-seat fatalities age 10 and over per 10 billion VMT

Final Model

$$(1\text{-}B^{12})\,LnY_t = (1\text{-}\ \underset{(.029)}{.873}B^{12})u_t - (1\text{-}B^{12})\,\underset{(.029)}{.178}X_{73} + (1\text{-}B^{12})\,\underset{(.050)}{.063}I_{118}$$

Adjusted R^2 = .23

Effect of North Carolina seat belt law, October 1985-June 1986: +6.5%

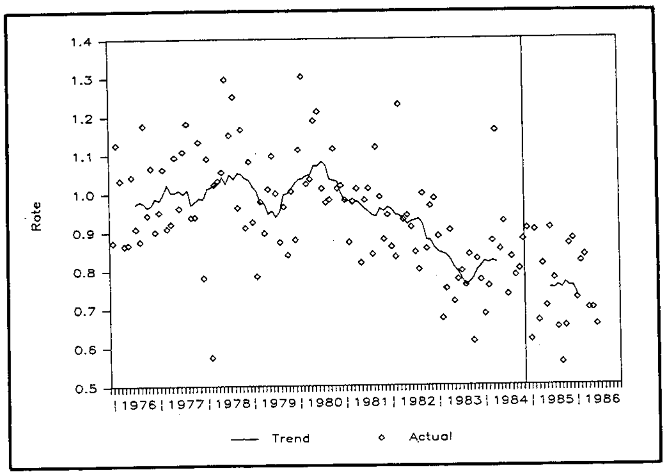

Figure A.9. Relative rate of front-seat fatalities age 10 and over per VMT: New York versus Pennsylvania

Final Model

$$(1\text{-}B)\,LnY_t = (1\text{-}\ \underset{(.006)}{.981}B)u_t - (1\text{-}B)\,\underset{(.035)}{.195}X_{85} - (1\text{-}B)\,\underset{(.042)}{.073}I_{108}$$

Adjusted R^2 = .40

Effect of New York Seat Belt Law, December 1984-June 1986: -7.1%*

* $p<0.05$, one-tailed test.

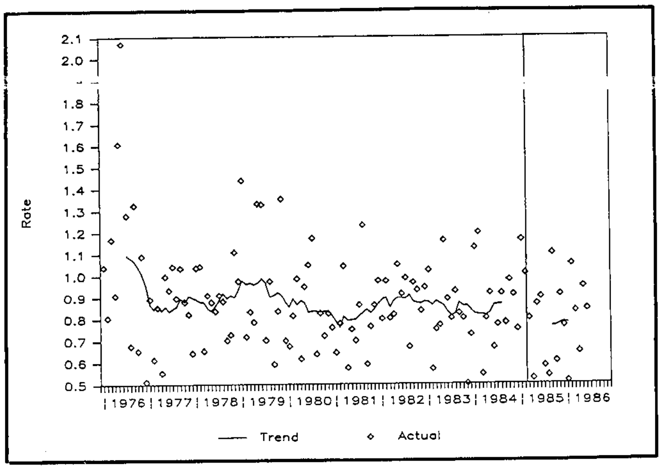

Figure A.10. Relative rate of front-seat fatalities age 10 and over per VMT: New Jersey versus Maryland

Final Model

$$LnY_t = u_t - \underset{(.070)}{.281}I_{111}$$

Adjusted R^2 = .25

Effect of New Jersey seat belt law, March 1985-June 1986: -24.5%*

* $p<0.05$, one-tailed test.

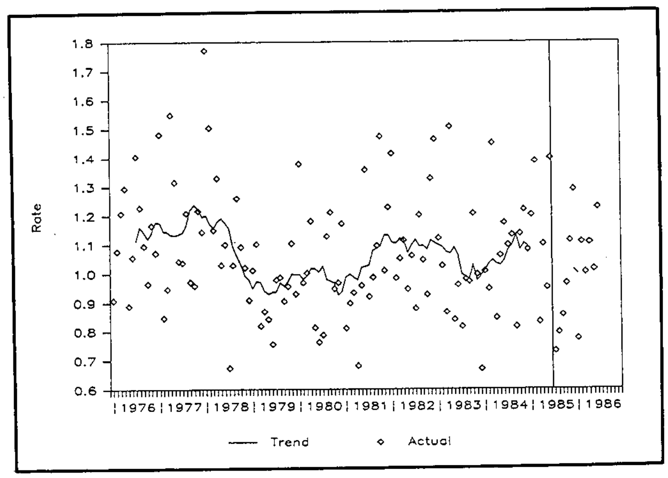

Figure A.11. Relative rate of front-seat fatalities age 10 and over per VMT: Michigan versus Ohio

Final Model

$$LnY_t = u_t - \underset{(.062)}{.042}I_{115}$$

Adjusted R^2 = .06

Effect of Michigan seat belt law, July 1985-June 1986: -4.1%

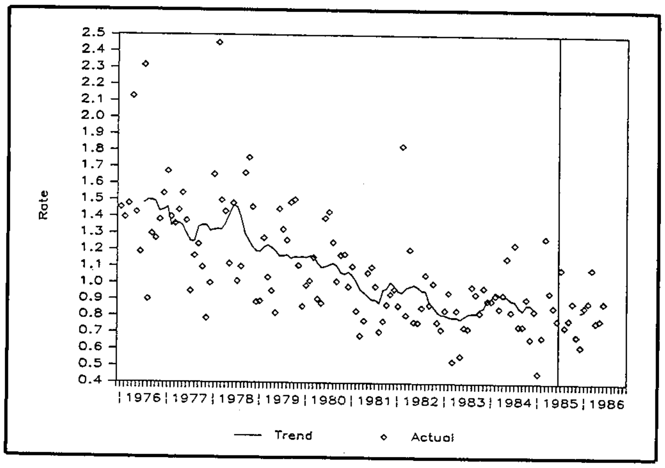

Figure A.12. Relative rate of front-seat fatalities age 10 and over per VMT: Illinois versus Indiana

Final Model

$$(1\text{-}B)\,LnY_t = 1 - \underset{(.029)}{.937}B)u_t - (1\text{-}B)\,\underset{(.088)}{.265}X_{61} - (1\text{-}B)\,\underset{(.092)}{.015}I_{115}$$

Adjusted $R^2 = .40$

Effect of Illinois seat belt law, July 1985-June 1986: -1.5%

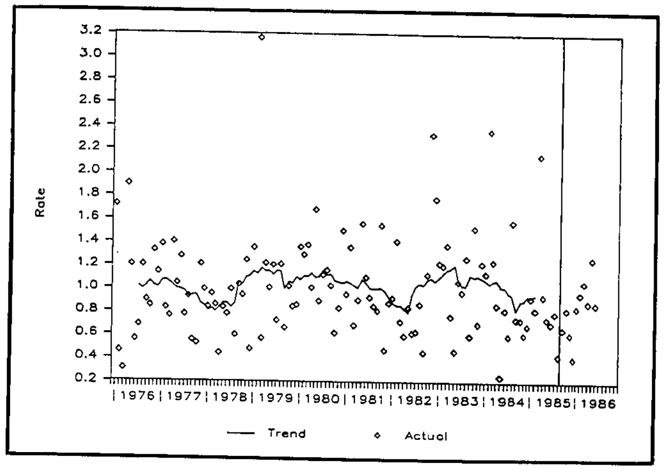

Figure A.14. Relative rate of front-seat fatalities age 10 and over per VMT: Nebraska versus Kansas

Final Model

$$LnY_t = u_t - \underset{(.235)}{.471}P_{114\text{-}116} - \underset{(.129)}{.214}I_{117}$$

Adjusted $R^2 = .01$

Effect of Nebraska seat belt law, September 1985-June 1986: -19.3%*

* p<0.10, one-tailed test.

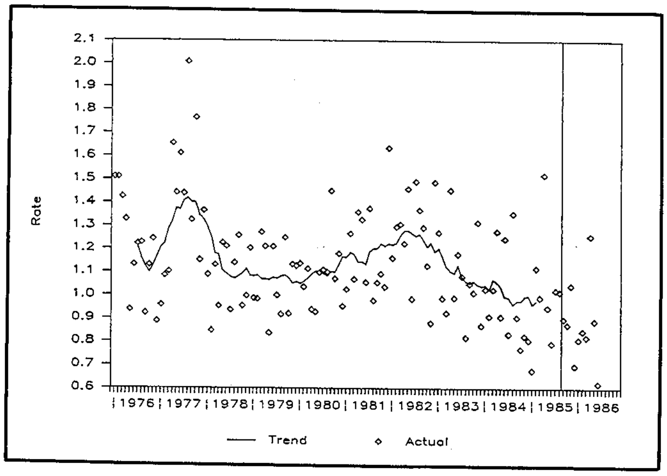

Figure A.13. Relative rate of front-seat fatalities age 10 and over per VMT: Texas versus Georgia

Final Model

$$(1\text{-}B)\,LnY_t = (1 - \underset{(.045)}{.860}B)u_t - (1\text{-}B)\,\underset{(.099)}{.123}I_{117}$$

Adjusted $R^2 = .18$

Effect of Texas seat belt law, September 1985-June 1986: -11.6%

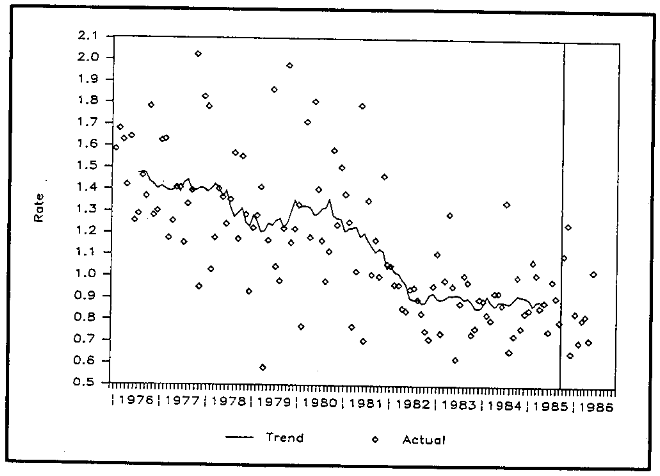

Figure A.15. Relative rate of front-seat fatalities age 10 and over per VMT: Missouri versus Tennessee

Final Model

$$(1\text{-}B^{12})\,LnY_t = (1 - \underset{(.031)}{.882}B^{12})(1 + \underset{(.093)}{.176}B^{7})u_t - (1\text{-}B^{12})\,\underset{(.046)}{.356}X_{73} - (1\text{-}B^{12})\,\underset{(.077)}{.045}I_{118}$$

Adjusted $R^2 = .43$

Effect of Missouri seat belt law, October 1985-June 1986: -4.4%

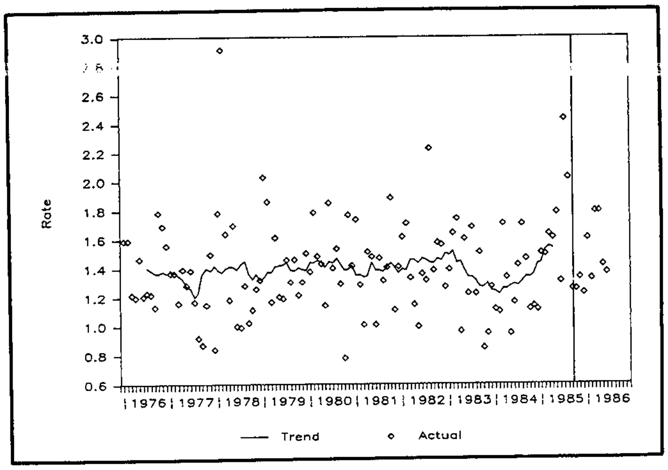

Figure A.16. Relative rate of front-seat fatalities age 10 and over per VMT: North Carolina versus Virginia

Final Model

$$(1\text{-}B)\,LnY_t = (1 - .992B)u_t + (1\text{-}B)\,.046I_{118}$$
$$\quad(.004)\qquad\qquad(.063)$$

Adjusted $R^2 = .02$

Effect of North Carolina seat belt law, October 1985-June 1986: +4.7%

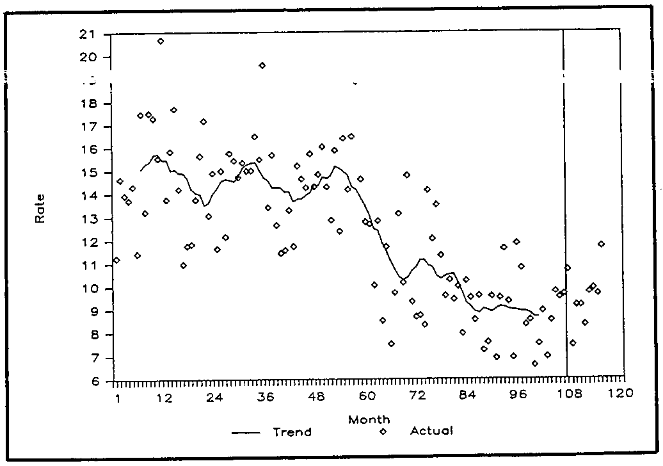

Figure A.18. Aggregate rate of rear-seat fatalities age 10 and over per VMT for eight belt- law states

Final Model

$$(1\text{-}B)\,LnY_t = (1+.259B^{12})\,(1 - .665B - .307B^4)\,u_t - (1\text{-}B)\,.366X_{60} - (1\text{-}B)\,.009I_{108}$$
$$(.091)\qquad(.062)\quad(.072)\qquad\qquad(.069)\qquad\qquad(.093)$$

Adjusted $R^2 = .67$

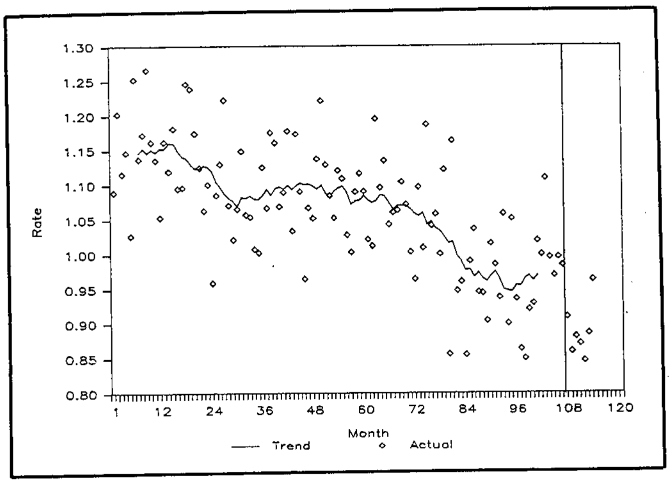

Figure A.17. Aggregate relative rate of front-seat fatalities age 10 and over per VMT for eight belt-law versus eight comparison states

Final Model

$$(1\text{-}B)\,LnY_t = (1 - .857B - .089^3)u_t - (1\text{-}B)\,.098X_{79} - (1\text{-}B)\,.091I_{108}$$
$$(.075)\quad(.079)\qquad\qquad(.028)\qquad\qquad(.034)$$

Adjusted $R^2 = .51$

Aggregate effect of seat belt laws, eight post-law months: -8.7%*

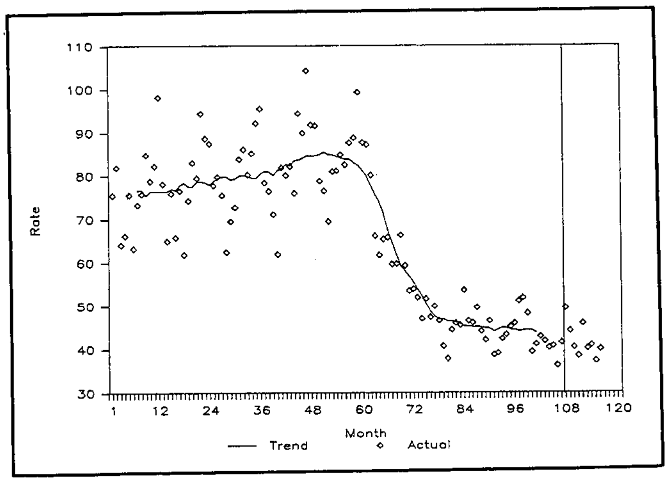

Figure A.19. Aggregate rate of nonoccupant fatalities age 10 and over per VMT for eight belt-law states

Final Model

$$(1\text{-}B)(1\text{-}B^{12})(1+.551B^{12})\,(1+.347B)\,LnY_t = (1+.551B^{12})\,(1+.347B)u_t$$
$$(.081)\qquad(.093)\qquad\qquad(.081)\qquad(.093)$$

$$-\,(1\text{-}B)(1\text{-}B^{12})\,.062X_{62} + (1\text{-}B)(1\text{-}B^{12})\,.090I_{108}$$
$$(.085)\qquad\qquad(.103)$$

Adjusted $R^2 = .87$

* $p<0.05$, one-tailed test.

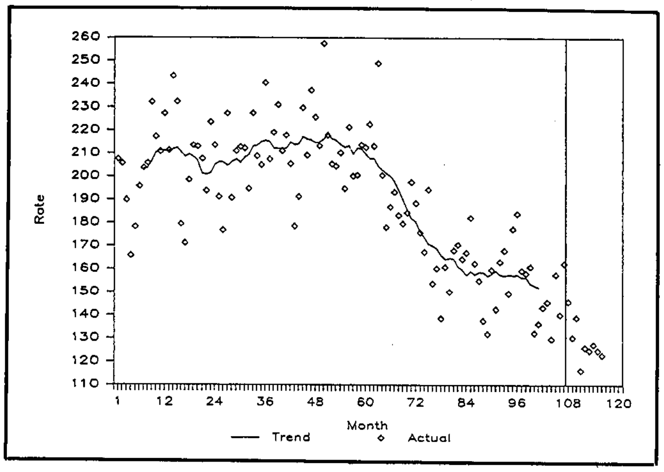

Figure A.20. Aggregate rate of front-seat fatalities age 10 and over per VMT for two primary enforcement provision states

Final Model

$$(1\text{-}B)(1\text{-}B^{12})\,LnY_t = \underset{(.033)}{(1\text{-}.873B^{12})}\,\underset{(.068)}{(1\text{-}.713B)}u_t - (1\text{-}B)(1\text{-}B^{12})\,\underset{(.052)}{.104}I_{108}$$

Adjusted $R^2 = .81$

Effect of primary states' seat belt laws, eight post-law months: -9.9%*

* p<0.05, one-tailed test.

† New York and Texas. North Carolina is not included as law not enforced until 1-1-87.

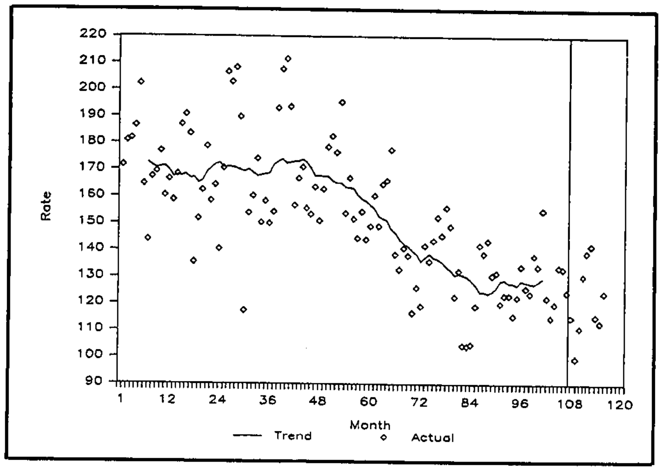

Figure A.21. Aggregate rate of front-seat fatalities age 10 and over per VMT for four secondary enforcement provision states

Final Model

$$(1\text{-}B^{12})\,LnY_t = \underset{(.097)}{(1+.228B)}u_t - (1\text{-}B^{12})\,\underset{(.034)}{.144}X_{66} - (1\text{-}B^{12})\,\underset{(.039)}{.070}I_{108}$$

Adjusted $R^2 = .69$

Effect of secondary states' seat belt laws, eight post-law months: -6.8%*

† New Jersey, Michigan, Illinois and Nebraska. Missouri is not included as law not enforced until 7-1-87.

Injuries to Restrained Car Occupants; What are the Outstanding Problems?

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Abstract

This Paper reports on the results of a comprehensive crash-injury investigation currently underway in England and has principally addressed the remaining injury problems associated with restrained occupants.

As a result of legislation introduced in 1983, seat belt wearing rates are of the order of 95 per cent for front seat occupants. The fitment and therefore the wearing rate for rear seat occupants is low but it is hoped that compulsory fitting of rear belts in new cars sold after April 1987 will lead to a corresponding reduction in rear seat occupant casualties.

The head and chest are seen as vulnerable areas requiring added protection, principally due to steering wheel contact (drivers) in frontal impacts and intruding objects with side impacts. The majority of struck-side occupant injuries for both chest and abdomen are due to door contact usually supported by an external object. Footwell intrusion is seen as a major source of lower limb injuries.

Impact zones and collision speeds have shown the contact areas to be considered for type approval testing. The majority of accident-involved cars are impacted by other vehicles which confirms that the vehicle structure and any modification to it still has a major part to play in occupant protection. A feature of the study is the estimation of velocity change in most of the impacts, the resulting indications being valuable for the selection of test conditions for regulatory tests.

The correct use of padded structures, particularly to the steering wheel, together with seat belt pretensioners could further assist in the reduction of casualties amongst the restrained car occupant population.

Introduction

After considerable deliberations by experts followed by much debate in Parliament, the seat belt law came into force in the United Kingdom on 31 January 1983. It is the personal responsibility of all *front* seat occupants over 14 years of age (with very few exceptions) to wear an approved restraint whilst travelling in cars and light vans. It is also the legal responsibility of the driver to ensure that children under 14 years old wear an approved front seat child restraint or an adult belt when travelling in the front passenger seat. Children under one year old must be in an approved child restraint designed for the child's age and weight. This law resulted in an initial saving of more than 200 deaths and 7,000 serious injuries per year compared to the pre-legislation period (1). Since the introduction of the law, compliance has remained at about 95 per cent. All cars sold from 1 April 1987 must have adult belts fitted in the rear but there is no legislation at the moment to cover their compulsory wearing. Approved child restraints are available for use in the front or rear and their fitting and use is encouraged in various ways.

A detailed clinical study was carried out by Rutherford et al (2) to assess the effectiveness of the seat belt legislation in injury terms. Casualty information was obtained from 14 participating hospitals one year before and one year after the legislation date of 31 January 1983. The study was organized as a series of hypotheses predicting changes in injury patterns and other key factors. The most important of these are presented in Table 1 and show that the hypotheses were confirmed in most cases.

Skull and facial fractures did increase slightly for drivers and the present study explores this further.

National Data

The police are usually called to the scene of a road traffic accident where there is injury, allegations of traffic offences and/or in cases of a potential traffic hazard. Frequently all three factors are present. These are not hard and fast criteria and sometimes depend upon police resources available at the time and other factors. When a police accident report is completed it is used for (a), their own purposes and (b), the compilation of national accident statistics (3). Only injury accidents are used in the latter and information on 'damage only' impacts is not available nationally. However information on such accidents reported to the police is often available at local level.

The injury severity of any casualty and therefore the accident severity is assessed by the police officer based on medical evidence available to him/her at the time. The categories of 'Fatal', 'Serious' and 'Slight' injury are used, working to agreed criteria. These assessments tend to vary slightly between officers and the descriptors cannot always relate to subsequent clinical findings (with the possible exception of fatalities). However, they are accepted as being adequate for national accident data which forms a large databank whose output is particularly useful for monitoring accident trends. Due to the relatively large number of cars in use, it is not surprising that these are involved in more road traffic accidents than any other class of vehicle. This results in a correspondingly higher proportion of car occupant casualties than any other road user group (Table 2).

National figures also show that the majority of pedestrian and motorcyclist accidents involve a car. Potential engineering solutions aimed at reducing casualties amongst these two road user groups will be presented by TRRL at this Conference.

It is interesting to note that, despite an increase in traffic over recent years, there has been no corresponding increase in casualty rates when related to cars and drivers (Figure 1).

However in spite of the above trends and the success of front seat restraint legislation, there is still cause for concern that over 2,000 car occupants are killed and over 27,000 are seriously injured each year. This Paper considers the detailed injury mechanisms and related factors from a sample of car accidents investigated as part of a comprehensive crash-injury study.

Study Background

As an essential part of an initiative on vehicle safety, the Transport and Road Research Laboratory

Table 1. Medical effects of seat belt legislation (Rutherford et al).

Factor	Prediction (↘ Decrease ↗ Increase)	Outcome		
		Drivers (%)	F.S.P. (%)	Confirmed
Out-Patients	↘	−10	−22	Yes
In-Patients	↘	−23	−43	Yes
Bed Days	↘	−27	−35	Yes
Severe Injuries	↘	−20	−24	Yes
Lung Injuries	↘	−33	−58	Yes
Sprained Necks	↗	+22	+ 8	Yes
# Sternum	↗	+24	+14	Yes
Brain Injuries	↘	−30	−57	Yes
Skull Fractures	↘	+ 8	−71	No
Minor Facial	↘	−44	−63	Yes
Eye Injuries	↘	−38	−40	Yes
Facial #	↘	+10	−46	Not Sig. Overall

Table 2. Casualties in Great Britain 1985.

Severity \ Road User Group	Car Occupants N (%)	Pedestrians N (%)	TWMV* N (%)	All Others N (%)	All Casualties N (%)
Killed	2061 (40)	1789 (35)	796 (15)	519 (10)	5165 (100)
All Severities	149452 (47)	61390 (19)	56591 (18)	50091 (16)	317524 (100)

*Two Wheel Motor Vehicles (Riders and Passengers)

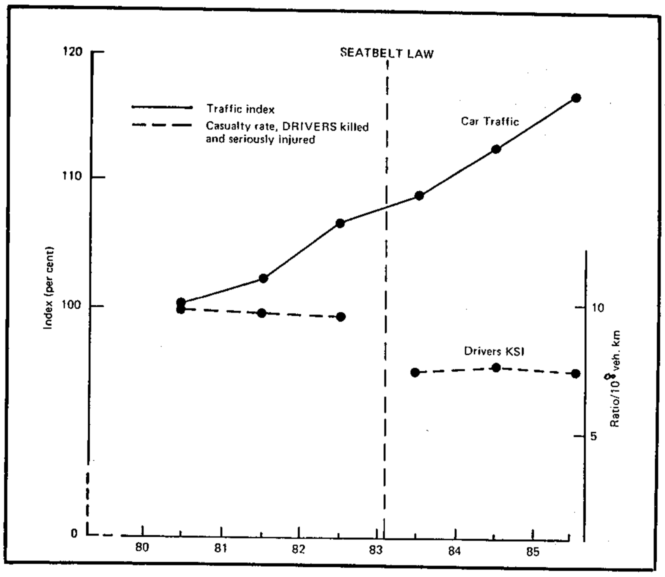

Figure 1. Traffic index and car driver casualty rate. Rel. to 1980 (Great Britain)

(TRRL), acting on behalf of the Department of Transport, are principal sponsors of a car occupant crash-injury research programme. The work is actively supported by the Rover Group, Jaguar Cars and also the Ford Motor Company acting as co-sponsors. The majority of investigations are carried out under contract by the Accident Research Units (ARU) of the University of Birmingham and the Institute for Consumer Ergonomics at Loughborough, with both Units analysing the data. In addition, the Department of Transport's own Vehicle Examiners play an important role in data collection and assessment. Altogether, information on approximately 650 accidents/year is sampled from various locations throughout England. Scientific staff at TRRL are involved in project management, data verification and analyses. All investigators work to standard data forms and collection procedures which have been fully described elsewhere (4,5). All data elements are computer codeable and stored electronically at the ARU's and TRRL. Data consistency checks are carried out by specialist TRRL staff in order to enhance data quality and check for possible errors, the latter being corrected in cooperation with ARU staff. After removing identifying features, all principal users of the data receive complete copies of the appropriate data sheets together with a set of photographic slides. Part analyses from earlier periods of this study have already been reported (6,7). However, the results presented in this Paper will be the first time that a more comprehensive database has been analysed. The study is on-going and subsequent papers and presentations will be able to call on a growing and enhanced database.

The Basic Data

Detailed information on 1,618 vehicles and 2,720 occupants collected between January 1984 and June 1986 is currently available for computerized analysis. To be considered for investigation, a 'case' vehicle must be less than 6 years old at the time of the accident and sufficiently disabled to be towed from the scene. The accident would also have been reported to the local police. Resources do not permit the investigation of every car accident within a given catchment area so the accidents are further stratified according to the police assessment of severity previously described. Using these categories, investigating teams endeavour to cover all fatalities, a high proportion of 'police serious' and lesser proportions of 'police slight' and 'police non-injury'. By their nature the latter category are bound to be underrepresented when compared to the possible numbers involved—particularly as only a proportion are reported to the police. For this reason damage-only accidents have not been used in this particular paper. The non-fatal accidents are chosen at random from police information. Weighting factors are computed to relate numbers of accidents investigated in a given geographical area to those reported to the police. Data 'weighting' is an accepted statistical technique for estimating the total population who may be involved in any particular study. A major advantage is that weighted numbers enable a true balance to be made between frequencies of accidents of different severities. A minor disadvantage of weighting is the possibility of spurious affects when applied to small numbers and consequently firm conclusions cannot be drawn when this occurs. Weighted and unweighted data are also compared for similarity of trends. Rounding off numbers and percentages will lead to minor differences in totals in several of the analyses. It is not possible to compare all parameters in the weighted sample with those in national accident statistics due to restrictions on the latter. However comparisons have been made of such factors as accident severity, casualty severity and seating position (Tables 3, 4 and 5).

The ratio between national data and the weighted sample shows close agreement and similar trends. Any minor differences could be due to the fact that a 'case' vehicle is less than 6 years old and has been

Table 3. Accident sample distribution (cars less than 6 years old).

Maximum Injury Severity/Vehicle	Unweighted Sample N	(%)	Weighted Sample N	(%)	National Data N	(%)
Fatal	97	(10)	125	(2)	1048	(2)
Serious	508	(50)	1392	(27)	11771	(25)
Slight	401	(40)	3607	(71)	33777	(72)
All Severities	1006	(100)	5124	(100)	46596	(100)

Table 4. Percentage age/seating position distribution—casualties in cars (Known values—all severities).

Position \ Age	0-4	5-9	10-16	17-24	25-34	35-44	45-54	55-59	60-64	65+	N.K.
Drivers											
National	-	-	<1	33	24	18	11	4	3	6	1
Weighted Sample	-	-	<1	23	24	18	10	6	5	4	9
Front Pass.											
National	<1	<1	6	35	17	11	9	4	4	9	4
Weighted Sample	<1	-	5	23	16	14	10	4	7	7	14
Rear Pass.											
National	10	12	17	31	9	5	5	2	2	6	1
Weighted Sample	10	12	10	24	9	4	3	3	3	4	18

towed away whereas national figures include vehicles of all ages, not necessarily towed away. There may be other minor differences, missing, incorrect data but overall it is considered that the weighted sample is fairly representative of national data. Thus the weighted number of cars is 5,131 and this is used for analysis.

Impact Configurations

The majority of the vehicles in the sample were saloon or hatchback cars reflecting the popularity of this body style on British roads. The majority were struck by other vehicles—principally cars. Roadside furniture of various types was also struck (Table 6).

In many of these impacts, the car structures together with in-built safety features have a major part to play in occupant protection and may possibly be further improved.

Principal direction of force (PDOF) was recorded using the Collision Deformation Classification system (CDC) (8) and shows that the majority (67 per cent) of impacts were to the front of the vehicle and 23 per cent to the side—more to the driver's off-side (Table 7). These proportions agree with national data. A more detailed analysis related to impact direction and location will be presented in later Sections.

Table 5. Percentage comparisons—casualties in car accidents.

Severity/Position	National Data (%)	Weighted Sample (%)
Slight Injury*		
Drivers	57	54
Front Passenger	25	25
Rear Passenger	18	21
Serious Injury*		
Drivers	58	55
Front Passenger	24	22
Rear Passenger	18	19
Fatal Injury*		
Drivers	62	64
Front Passenger	22	22
Rear Passenger	16	14
All Severities*		
Drivers	57	56
Front Passenger	25	24
Rear Passenger	18	20

*Using Police Severity Criteria - see text

Table 6. Principal object hit relative to bodystyle (weighted data).

Object hit \ Body style	Saloon	Hatch-back	Estate	Convert-ible, top on	Light Goods Multi-purpose	3 Wheeler	Total
Car	1145	1720	192	16	20	0	3093
Light Goods Vehicle	65	121	2	0	0	10	198
Heavy Goods Vehicle	202	217	40	0	1	0	460
Public Service Vehicle	16	70	0	3	0	0	89
Other Vehicle	7	55	15	0	0	0	77
Pedal - Motorcycle	6	22	0	0	0	0	28
Sign, Post or Pole	123	171	27	5	17	0	343
Crash Barrier	3	50	0	0	0	0	52
Wall	41	108	5	0	0	0	154
Road Furniture	72	32	17	0	0	0	122
Tree	56	52	3	1	0	0	112
Ditch/Natural Object	18	58	5	0	8	0	89
Roll over - no impact	35	75	3	0	2	0	146
Roll over + Sig. impact	54	52	5	0	0	0	115
Other	7	37	0	0	0	0	44
Not known	12	6	0	0	0	0	18
Total	1863	2860	314	36	49	10	5131

Where possible, change of speed on impact (ΔV) estimates were computed using the CRASH 3 routine (9). Values were available for 49 per cent of the weighted sample (Figure 2), the majority being in the 11-40 km/h band with a peak between 20 and 30 km/h. It is presumed that low speed impacts would either not have been reported to the police or not sampled.

Restraint Use Related to Seating Position and Age

The majority of vehicles were fitted with three point inertia reel belts in the front seating positions. Less than one per cent of adult restraints were fitted in the rear. The distribution of restraint use related to seating position is give in Table 8. The 'Used' category indicates that there are witness marks on the belt or webbing and/or appropriate injuries to the wearer. 'Used-Unproven' is where an occupant states that he or she wore the belt at the time of the accident (in response to a questionnaire) but there is no visual or medical evidence to support this. The latter two

Table 7. Principal direction of force (PDOF) related to struck side (vehicles—weighted data).

PDOF \ Struck side	Front	Right Side	Left Side	Rear	Top	Under	Unclass.	Total
Rollover	7	23	22	15	123	4	65	259
1 o'clock	615	71	0	0	0	0	0	686
2 o'clock	115	112	0	0	0	5	0	233
3 o'clock	12	315	0	0	0	0	0	327
4 o'clock	0	66	0	5	0	0	0	71
5 o'clock	0	75	0	71	0	0	0	147
6 o'clock	0	2	0	225	0	0	0	226
7 o'clock	0	0	21	5	0	0	0	26
8 o'clock	0	0	42	0	0	0	0	42
9 o'clock	0	0	182	0	0	0	0	182
10 o'clock	107	0	157	0	0	0	0	264
11 o'clock	460	0	39	0	3	0	0	502
12 o'clock	2104	52	7	0	3	0	0	2168
Total	3420	716	470	320	128	9	65	5131

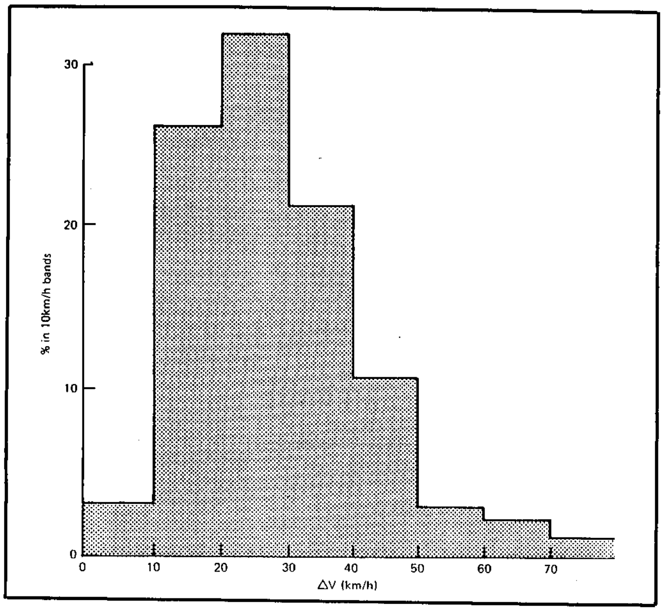

Figure 2. Distribution of velocity change on impact (ΔV). Cars in injury accidents (weighted data)

categories will be combined for some later analyses. Some of the vehicles involved were stationary with the drivers seating position unoccupied. This would account for a slight difference between the number of drivers and number of vehicles, the difference being magnified when weighted.

In addition to the percentage distribution given in Table 8, the data also shows an estimated 93 per cent of drivers and 91 per cent of front seat passengers wore restraints, ignoring 'Not Known' values. These figures indicate a slightly lower use rate than the national average of 95 per cent. The data also shows that only 4 per cent of the rear seat occupants wore restraints. Age related to restraint use shows that the young are at risk (Table 9).

Analysis of restraint use by age shows that most of the unrestrained were in the rear seats although a significant proportion were front seat occupants, particularly those in the middle years age group (Table 10).

The data shows the increase in casualty involvement in the 17-24 year age group. This particular age group

Table 8. Restraint use and seating position in injury accidents (weighted data).

Belt Use \ Position	Driver N	Driver (%)	Front Pass N	Front Pass (%)	Rear Pass N	Rear Pass (%)	N.K. N	Total N	Total (%)
Used	3412)	(82)	1277)	(80)	55)	(4)	2	4746)	(66)
Used-Unproven	750)		434)		17)		0	1202)	
Not used	306	(6)	164	(8)	1692)	(94)	10	2172	(24)
Not known	620	(12)	258	(12)	36	(2)	4	918	(10)
Total	5089	(100)	2133	(100)	1801	(100)	16	9038	(100)

Table 9. Age related to restraint use—all seating positions (weighted data).

Belt Use \ Age	0–4 %	5–9 %	10–16 %	17–24 %	25–34 %	35–44 %	45–54 %	55–59 %	60–64 %	65+ %	Age N.K. %	Total [N]
Used	11	<1	28	56	58	60	57	54	53	45	51	4746
Used-Unproven	4	–	4	8	13	23	21	12	23	11	9	1202
Not used	85	96	67	26	15	8	12	24	20	28	28	2172
Not known	–	3	2	10	14	9	10	10	4	16	11	918
Total [N]	186	208	291	2105	1721	1352	782	440	436	484	1033	9038

Table 10. Unrestrained car occupants in injury accidents percentage per age group (weighted data).

Position \ Age	0–4 %	5–9 %	10–16 %	17–24 %	25–34 %	35–44 %	45–54 %	55–50 %	60–64 %	65+ %	N.K. %	Total N
Driver	–	–	1	13	35	39	26	43	5	12	3	306
Front Pass	–	–	2	7	1	5	25	2	38	26	8	164
Rear Pass	100	100	97	79	64	57	49	56	58	63	87	1692
Other /NK	–	–	–	<1	–	–	–	–	–	–	2	10
Total [N]	158	200	195	545	258	102	95	105	88	137	287	2172

ranks high on the national road traffic casualty scale and almost 80 per cent of their accidental deaths are due to road traffic accidents.

Frontal Impacts

Table 7 showed that the majority of impacts were to the front. In order to eliminate end swipe accidents for this part of the analysis, only impacts of between 11 and 01 clock direction will be considered. This represents 62 per cent of the weighted sample.

Table 11 shows the CDC vehicle front body region impacted in relation to object hit. Collisions with other vehicles, particularly cars, still predominate. Two-thirds of the frontal impacts involved the car centre and the offside front was struck more frequently than the nearside. Table 8 showed that only 4 per cent of rear seat occupants were restrained in the accidents. Therefore, in order to give more precise definitions of injuries and mechanisms, only restrained front seat occupants will be considered in this part of the analysis. Two accidents involving total fire

Table 11. Principal object hit relative to vehicle body zone cars in injury accidents—frontal impacts (weighted data).

Object hit \ Part Impacted	Left	Centre	Right	Left + Centre	Right + Centre	Full Width	Total
Car	288	14	346	234	545	585	2013
Light Goods Vehicle	22	0	33	13	18	36	121
Heavy Goods Vehicle	16	9	76	34	74	71	280
Public Service Vehicle	10	0	5	2	4	17	38
Other Vehicle	48	0	1	1	0	20	70
Sign, Post or Pole	59	55	59	34	24	0	232
Crash Barrier	6	0	6	0	0	22	34
Wall	27	5	30	8	14	44	128
Road Furniture	6	19	53	1	0	3	82
Tree	17	12	13	2	14	10	67
Ditch/Natural Object	8	0	8	2	0	47	83
Other	2	2	5	6	12	16	42
Total	507	116	636	338	704	870	3171

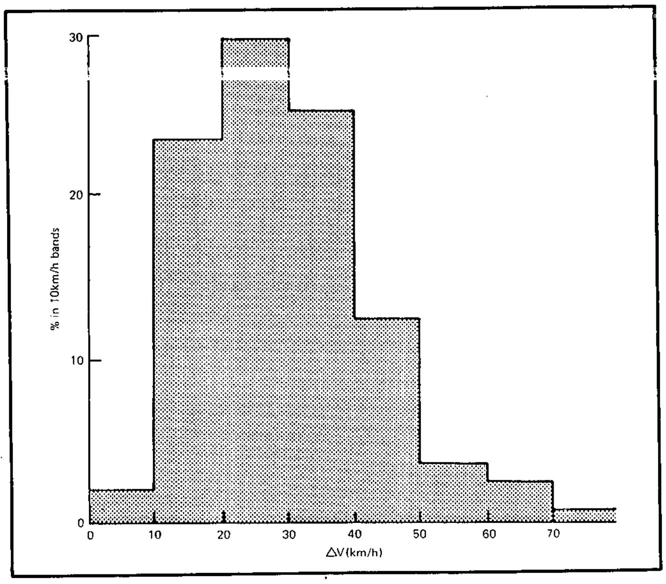

Figure 3. Distribution of velocity change on impact (ΔV). Cars in injury accidents—frontal impacts (weighted data)

loss with corresponding fatal injuries, 5 light goods and 3 fibre glass body vehicles have also been eliminated. With this slightly reduced sample, ΔV values are shown in Figure 3 and again indicate predominant values in the 11-40 km/h bands.

Injury Severity

One-third of the front seat occupants were uninjured and approximately half only had injuries of MAIS = 1 which strongly suggests that seatbelts play a major role in injury reduction (Table 12).

The work by Hobbs and Mills (10) related probability of injury to changes in ΔV and also indicated the injury shift between the restrained and unrestrained front seat occupant. Relating injury severity to impact severity for restrained Drivers and Front Seat Passengers in frontal collisions shows that MAIS 3 injuries occur from approximately 45 km/h and also shows little difference exists between these two groups of front seat occupants until higher values of ΔV are reached (Figure 4).

The importance of injuries to each body region can be assessed by several methods. The proportion of occupants with an injury to a particular region is a

Table 12. Overall injury severity distribution (%)—restrained front occupants in frontal collisions (weighted data—injury accidents).

Position \ MAIS	Uninjured	1	2	3	4	5	6	Total N (%)
Driver	31	45	16	4	<1	<1	<1	2602(100)
Front Pass	35	51	11	2	<1	<1	-	1092(100)
Total N (%)	1188(32)	1731(47)	545(15)	133(4)	30(<1)	23(<1)	9(<1)	3694(100)

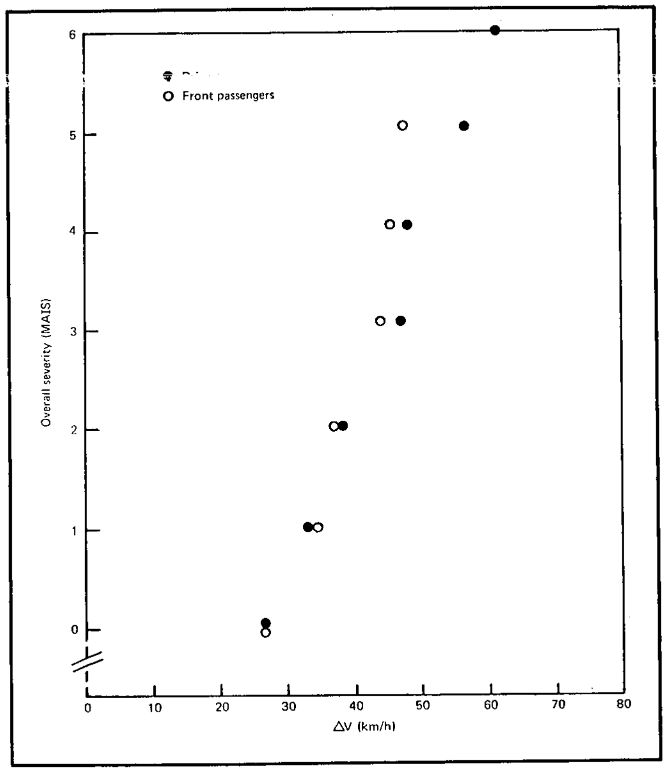

Figure 4. Max. injury severity vs. mean impact severity. Restrained front occs. of cars in frontal impacts

measure of injury incidence. Using this technique the lower limbs are the most frequently injured (81 per cent) followed by the head (52 per cent), as shown in Table 13.

A further measure is to only consider severe injuries ie, AIS 3+. (195 front occupants). A third method suggested by Malliaris (11) is to further weight the AIS values by a 'harm' factor. This is based on injury cost which would vary between countries for a variety of economic reasons. However, proportions are likely to be similar and will serve in this instance to illustrate the point. Using the HARM scale an AIS 6 injury has a weight 378 times greater than an AIS 1 injury (Table 14).

Table 13. Maximum AIS for body regions of restrained front occupants frontal impacts (weighted data).

Body region \ Max AIS	1	2	3	4	5	6	All injuries N	(%)*
Head/face	990	252	24	23	11	4	1304	(52)
Neck	156	11	16	-	3	3	189	(8)
Upper limbs	801	138	47	-	-	-	986	(39)
Chest-Upper back	978	180	43	8	6	4	121	(49)
Abdomen-Lower back	245	4	17	12	10	-	288	(11)
Pelvis and Hips	410	22	13	-	-	-	445	(18)
Lower limbs	1345	178	52	-	-	-	2020	(81)

* Expressed as a percentage of 2506 injured, front occupants (weighted)
NB: Values are not mutually exclusive.

Table 14. Harm weighting factors based on injury costs (Malliaris).

AIS value	1	2	3	4	5	6
HARM weighting factor	0.7	3.0	9.2	56.7	232.5	264.9

Table 15 summarises the ranking of body areas using the three methods. It is considered that the HARM system is the most useful rank order as it is the only scale accommodating both frequency and severity of injury.

Table 15. Rank order of body area importance by three methods restrained front occupants in frontal impacts (weighted data).

Rank Order	All Injuries Body Region	(%)	Injuries AIS 3+ Body Region	(%)	Total harm Body Region	(%)
1	Lower limbs	(81)	Head/Face	(32)	Head/Face	(33)
2	Head/Face	(52)	Chest-Upper back	(31)	Chest-Upper back	(23)
3	Chest-Upper back	(49)	Lower limbs	(27)	Abdomen-Lower back	(17)
4	Upper limbs	(39)	Upper limbs	(24)	Lower limbs	(10)
5	Pelvis and Hips	(18)	Abdomen-Lower back	(20)	Neck	(9)
6	Abdomen-Lower back	(11)	Neck	(11)	Upper limbs	(7)
7	Neck	(8)	Pelvis and Hips	(7)	Pelvis and Hips	(2)

Based upon weighted data both the serious injury and the HARM scale suggest that the priorities are to reduce head and chest injuries. It should be noted that percentage totals exceed 100 as injuries are not mutually exclusive.

Injury Mechanisms

Contact locations giving rise to AIS 2+ injuries to a vulnerable body regions show that the steering wheel has a major involvement (Table 16).

Table 16. Restrained front occupants in frontal collisions contact locations for vulnerable body regions (AIS≥2) (weighted data).

Contact locations \ Body Region / AIS	Head/Face 2-3	Head/Face 4-6	Neck 2-3	Neck 4-6	Chest-Upper back 2-3	Chest-Upper back 4-6	Abdomen-Lower back 2-3	Abdomen-Lower back 4-6	Total N	Total (%)
Steering wheel	144	15	–	3	22	8	4	6	202	(34)
Belt Webbing	–	–	–	–	178	2	5	4	189	(32)
Other vehicle	14	9	–	1	2	2	–	2	30	(5)
'A' Pillar	19	1	1	–	4	3	–	–	28	(5)
Windscreen, surround	32	–	–	–	–	–	–	–	32	(5)
Fascia	13	–	–	–	7	–	–	2	22	(4)
Other Occupant	6	1	1	–	1	–	6	1	16	(3)
Bonnet	18	2	–	–	–	–	–	–	20	(3)
Roof	6	3	4	–	–	–	–	–	13	(2)
Glass	10	–	–	–	–	–	–	–	10	(2)
Front Header Zone	10	–	1	–	–	–	–	–	11	(2)
Own Seat	1	–	–	–	6	–	–	–	7	(1)
Other	6	–	–	–	3	–	1	4	14	(2)
								Total	594	(100)

(NB. Values are not mutually exclusive)

The above values are not mutually exclusive in that it is possible to obtain a chest *and* head injury from say, a steering wheel contact in higher energy impacts. The other vehicle intruding into the occupant space as well as the 'A' pillar and windscreen/surround were also responsible for a significant number of head injuries.

Seat belt webbing was responsible for a high number of AIS 2-3 chest injuries. These were frequently minor rib/sternum fractures and anticipated (Table 1). They could almost be considered a 'trade-off' in relation to serious and fatal injuries occurring to the unrestrained in similar accidents. There is a possibility that seat belt pre-tensioners could reduce a number of the head/chest steering wheel related and also seat belt induced injuries.

Referring to non-vulnerable body regions, foot-well intrusion was responsible for the majority of injuries to the lower limbs (Table 17). Intrusion related injuries will be dealt with in a later Section. The fascia area also featured in lower limb injury causation.

A further area of concern is neck sprains. These are not life-threatening and often present sometime after the accident. They sometimes cause severe discomfort and have been known to persist for considerable periods of time. They have been coded as 'Other' injuries in this particular study and an estimated 653 such lesions (11 per cent) occurred to restrained front seat occupants in frontal impacts. At the moment there is insufficient data to assess the effectiveness of head restraints in reducing this type of injury. However, it is hoped to carry out such an analysis at a later date.

Intrusion Related Injuries

The study notes whether passenger compartment intrusion was a factor causing injury. This is a difficult area as it could be argued that a particular body region may have struck the vehicle interior anyway without intrusion taking place. A further factor is that the body region concerned did not have to travel so far before impact which may have lessened the injury severity in some cases. However previous information (Table 17) indicated that foot-well certainly caused a high percentage of lower limb

Table 17. Restrained front occupants in frontal collisions contact locations for non-vulnerable body regions (AIS 2-3) (weighted data).

Vehicle Location \ Body Region	Upper limbs	Lower limbs	Hips/ Pelvis	Total N	Total (%)
Pedals/Intrusion	–	206	3	209	(44)
Fascia	18	61	1	80	(17)
Belt Webbing	41	–	–	41	(9)
'A' Pillar	18	8	2	28	(6)
Glass	18	–	–	18	(4)
Steering wheel	16	–	–	16	(3)
Column area	–	14	1	15	(3)
Door	11	–	2	13	(3)
Front firewall	–	6	3	9	(2)
Loose object	8	–	–	8	(2)
Car centre	–	5	3	8	(2)
Bracing	3	–	3	6	(1)
Other	16	6	1	23	(5)
			Total	474	(100)

(NB. Values are not mutually exclusive)

injuries. Another side of the argument is that any decrease in occupant space will tend to negate any safety features which may have been designed to operate with an optimum size occupant compartment and the structures/components associated with it. Also the effects of body shell distortion arising out of intrusion cannot be ignored.

Although some form of energy absorption exists on the steering columns, the intruding steering wheel played a major part in injury causation, particularly to the head/face and chest (Table 18). Impact distortion of the vehicle front and firewall area may have contributed to steering assembly movement.

The intrusion of the other vehicle involved in the accident was responsible for the next highest group of injuries, principally to the head area and usually via the glazed sections of the vehicle.

Footwell intrusion and pedal involvement featured highly in relation to the lower limb injuries (Table 19).

Intrusion in this area is likely to be an indirect result of engine compartment distortion, sometimes involving the front road wheel assembly.

General Discussion—Restrained Front Occupants In Frontal Impacts

The current study shows that approximately 67 per cent of car accidents are frontal, the striking object principally being another vehicle and to a less extent roadside furniture. The impact zone involves either the centre or the full width of the car front. The majority of the impacts in this study occur in the 11-40 km/h$\triangle$V range, peaking in between 20-30 km/h.

Using either 'harm' or severe injury ranking order, the head and chest are body regions where further protection is needed. The steering wheel ranks highly as causing injury to these two body regions, either as a direct contact with the trunk and head going forward following impact or as the result of steering mechanism intruding into the passenger area. Forward body movement is also likely to be a related feature and could possibly be countered by seat belt pretensioners. Seat belt induced injuries occur to the chest, principally bruising and simple fractures. Although these are still undesirable injuries, they could be considered as a 'trade-off' relative to the more serious or fatal injuries that could occur to unrestrained occupants in similar collisions.

The area of concern for the non-vulnerable body regions is the high proportion of lower limb injuries resulting from footwell intrusion. However the introduction of anti-intrusion measures into car occupant compartments must also take account of high peak decelerations that may occur to the occupants unless energy absorbing countermeasures are introduced at the same time. This point has been well recognized in the work by Hobbs et al and reported at this Conference (12). Sufficient energy absorption must be provided ahead of the passenger compartment. However, with a high percentage of smaller cars, the depth available for such absorption is limited by physical constraints within the engine compartment and surrounding areas. In this case the dynamic stiffness of the absorbing structure must be high leading to an even stiffer passenger compartment to ensure that the latter does not collapse. In an experimental design, an Austin Metro has been modified to take account of these guidelines and will be fully reported at this Conference (12).

There is concern that only an estimated 4 per cent of the rear seat occupants were restrained in the accident sample. This situation should improve with the compulsory fitting of adult rear belts in cars. However it is too early to judge the likely extent of their wearing rate and child restraint use must be seen as a feature of this. It has been suggested that bench-type rear seats may present problems in relation to belt mounting points and the lie of the webbing across the body. Collapsible seat backs, either full width or split, as in the hatchback/estate type of car,

Table 18. Restrained front occupants in frontal collisions intrusion contacts for vulnerable body regions (AIS $\geq$2) (weighted data).

Location \ Body Region	Head/face		Neck		Chest-Upper back		Abdomen-Lower back		Total	
AIS	2-3	4-6	2-3	4-6	2-3	4-6	2-3	4-6	N	(%)
Steering wheel	82	10	–	2	17	7	4	6	128	(51)
Other vehicle	14	9	–	1	2	2	–	2	30	(12)
'A' Pillar	17	1	–	–	3	3	–	–	24	(10)
Windscreen, surround	19	–	–	–	–	–	–	–	19	(8)
Roof	6	3	4	–	–	–	–	–	13	(5)
Bonnet	10	2	–	–	–	–	–	–	12	(5)
Fascia	6	–	–	–	6	–	–	–	12	(5)
Front header area	4	–	1	–	–	–	–	–	5	(2)
Door	–	–	–	–	2	–	–	2	4	(2)
Other	1	–	–	–	2	–	–	–	3	(1)
								Total	250	(100)

(NB. Values are not mutually exclusive)

Table 19. Restrained front occupants in frontal collisions intrusion contacts for non-vulnerable body regions (AIS 2-3) (weighted data).

Location \ Body Region	Upper Limbs	Pelvis/Hips	Lower Limbs	Total N	(%)
Footwell intrusion	–	3	128	131	(51)
Pedals	–	–	27	27	(11)
Fascia/Parcel shelf	12	–	14	26	(10)
'A' Pillar	6	2	8	16	(6)
Steering column area	–	1	13	14	(5)
Door	9	2	–	11	(4)
Steering wheel	9	–	–	9	(3)
Front Firewall	–	3	3	6	(2)
Other Vehicles	3	1	–	4	(1)
Other	6	–	5	11	(4)
			Total	255	(100)

(NB. Values are not mutually exclusive)

are additional difficulties. These points will be monitored as rear belt usage increases.

Side Impact Characteristics

Improvements in side impact protection have been suggested by Hobbs et al and reported at this Conference (13). A standard hatchback car has been modified and is currently on display.

Vehicles that sustained their most severe impact on the right side with a direction force of 2, 3 or 4 o'clock or to the left side with a direction force of 8, 9 or 10 o'clock were defined as side impacts. Within the population of 51 31 vehicles there were 831 (17%) that sustained a side collision as the impact that dissipated the most energy within the vehicle structure. These numbers were based on an original unweighted sample of 193 vehicles. Of these 831 cars, 29 (4%) were involved in fatal accidents, 277 (33%) were in serious injury accidents and 525 (63%) in slight injury accidents. These severities were determined using the UK Police classification system.

Directions of impact

The most common directions of force were the perpendicular ie, 3 o'clock and 9 o'clock, 482 (58%) of the 831 vehicles receiving a side-impact. There were 501 vehicles (60%) struck on the right (driver's) side at ± one clock point either side of perpendicular.

The direction of force of side impacts is described for each injury severity in Table 20 but expressed in terms of the angle from straight ahead. Amongst those vehicles in which the maximum injury severity was AIS 1 or AIS 2/3 the most common impact direction was perpendicular to the vehicle in 368 (60%) and 105 (53%) cases respectively. To simulate accidents of these severities an impact would correspond well to reality if the direction of force were perpendicular. If an impact were to attempt to reproduce the most severe type of side impact in which AIS 4+ injuries are found only 10 (33%) would be simulated by a perpendicular impact. A larger proportion (66%) of fatal impacts would be simulated in an impact with a direction of force of ± 60° from straight ahead.

Table 20. Direction of force for each injury severity (°from straight ahead): all side impact vehicles.

Directions from Straight Ahead(°)	Maximum AIS in Vehicle		
	0/1	2/3	4-6
± 60°	172 (28%)	74 (37%)	20 (67%)
± 90°	368 (60%)	105 (53%)	10 (33%)
± 120°	77 (12%)	19 (10%)	0
Total	617(100%)	198(100%)	30(100%)

Table 21. Location of damage along car side, all vehicles with restrained occupant.

Position	AIS of most severe injuries			
	1	2-3	4-6	All known severities
Front alone	143 (34%)	62 (26%)	0	205 (30%)
Front and passenger compartment	110 (26%)	54 (23%)	15 (45%)	179 (26%)
Passenger compartment alone	91 (22%)	74 (31%)	15 (45%)	180 (26%)
Passenger compartment and rear	48 (11%)	44 (19%)	1 (3%)	92 (13%)
Rear alone	19 (4%)	0	0	19 (3%)
Whole side	12 (3%)	2 (1%)	2 (6%)	16 (2%)
Total population estimate	423(100%)	236(100%)	33(100%)	692(100%)

Damage location

The collision deformation classification (CDC) was used to describe the nature of the direct contact on the vehicle. The location of the damage along the side of the car is shown in Table 2 for all impacts, and for each severity, classified using the fourth character of CDC.

Amongst all severities of accident there were 451 (65%) cars with the direct crush overlapping the passenger compartment. There were therefore 241 (35%) cars in side impacts where there was no overlap of the passenger compartment by the striking object. However, these vehicles tended to be involved in the less severe accidents. Amongst those 423 vehicles in which only AIS 1 injuries occurred, 162 (38%) did not have direct contact over the passenger compartment while of the 236 involving AIS 2 or 3 injuries, 62 (26%) did not. Of the 33 cars where the maximum severity was AIS 4+ all had direct contact over the passenger compartment. When the passenger compart-

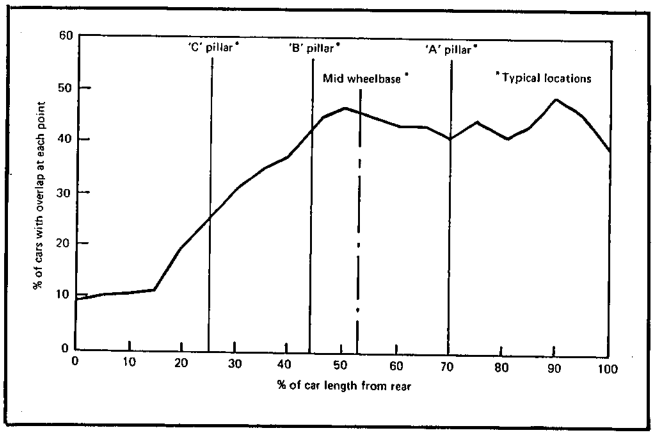

Figure 5. Probability of direct contact, all side impacts (all injury severities—weighted data)

ment is not directly struck the resulting injuries only rarely become serious or fatal. It can also be concluded that measures that strengthen the passenger compartment to reduce the effects of intrusion are unlikely to benefit the occupants of 26% of the cars involving AIS 2-3 injuries and the 38% involving AIS 1 injuries. Such measures might, however, be expected to benefit the remainder of the occupants of cars in each category.

The location of the direct contact along the side of all cars is shown in Figure 5. It is rare that cars in which an occupant is injured is struck towards the rear. Less than 10% of these cars have direct contact within the rear 15% of the car side. The front 60% of the car side is most likely to be involved with typically 40% of cars being struck in this area. The positions of the A and B pillars, the mid point of the wheelbase and the rearmost part of the passenger compartment were measured in a small group of cars—large and small hatchbacks and saloons. The location of these points was found to vary only a little and their average positions have therefore been superimposed upon the graphs as a guide.

The probability of direct contact along the car side for fatal side impacts is shown in Figure 6 with typical positions of car structures indicated. As shown, the zone most commonly involved in direct contact is narrowly concentrated between the 'B' pillar and slightly forward of the wheelbase centre. This corresponds to 20 per cent of the car length. The part most frequently directly contacted is the area between the 'B' pillar and wheelbase centre where 70 per cent of side impacted vehicles in which someone is killed is struck.

Proposals have been made concerning a legislative side impact performance test, these frequently involve a mobile barrier. The proposed width of the EEVC barrier is 150cm, and the NHTSA barrier is 168cm. These correspond to 41% and 36% of a typical car length. Both barriers fit the direct contact distributions of side impacts of all severities fairly closely. However, both barriers appear to be double the length necessary for a close fit to the fatal side impact damage distribution.

Impact Severity

The CRASH 3 computer program was used to estimate the speed change within the impact. The values for 383 were calculated, the remainder violated at least one of the assumptions of CRASH 3. The known delta-v of the cars sustaining a side impact is shown in Figure 7. The mean value of delta-v for the whole population is 25 km/h, for AIS 4+ impacts it is 50 km/h, 31 km/h for AIS 2-3 and 22 km/h for AIS 1 impacts.

If a test impact were to attempt to reproduce the most common side impacts, it is recommended that the delta-v for the side impact vehicle should be 25 km/h. If the test impact were intended to reduce the

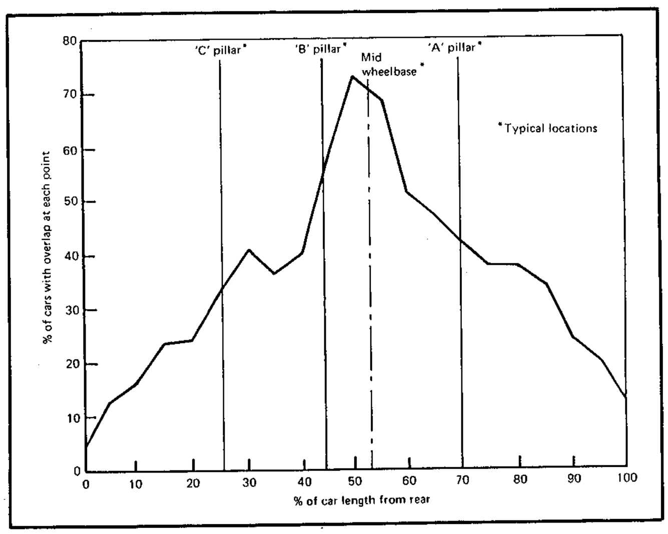

Figure 6. Probability of direct contact, all side impacts (fatalities—weighted data)

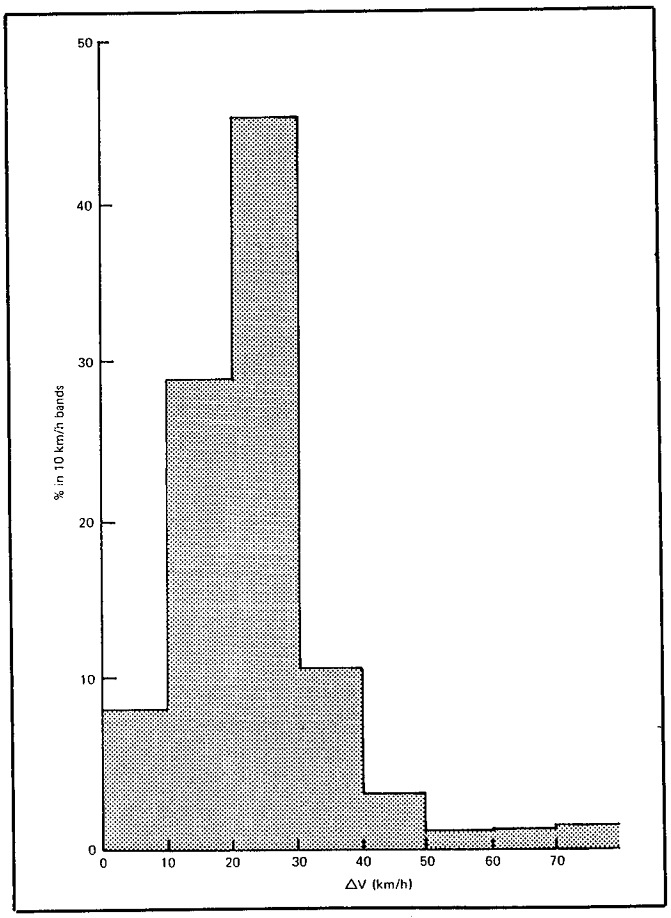

Figure 7. Distribution of velocity change on impact (ΔV). Cars in injury accidents—side impacts (weighted data)

Table 22. Seating positions of restrained occupants maximum AIS = 1+.

	Struck Side	Non-struck Side
Front side	304 (74%)	176 (61%)
Front left	104 (25%)	114 (39%)
Rear right	2 (0.5%)	0
Rear left	0	1 (0.5%)
Total	410 (100%)	292 (100%)

incidence of the most severe injuries it should be performed at a delta-v of at least 50 km/h although some reduction in fatalities might be expected with a test involving a lower delta-v.

Occupant Characteristics

There were 701 injured occupants in the 692 cars involved in a side impact who were also restrained. Their seating positions are shown in Table 22.

Since there is no legal requirement in the UK that rear seat occupants be restrained, 99% of those examined in this analysis are front seat occupants. The severity of the injuries all 701 sustained is shown in Table 23 for the Police severity and Table 24 for the maximum AIS for struck side and non-struck side occupants separately.

Table 23 shows that 42% of struck side occupants were in the serious or fatal categories compared with 25% of the non-struck side occupants. This pattern is repeated in Table 24, 15% of struck side occupants sustained injuries of AIS 3 or above compared with 7% of non-struck side occupants. These differences cannot automatically be attributed to any additional protection non-struck side occupants may receive as these occupants happened to be passengers in vehicles with lower impact severities than struck side occupants as Table 25 shows.

There were 171 struck side occupants with a known delta-v and 167 non-struck side occupants. Above 30 km/h the two distributions are closely similar but significant differences lie in the lower bands. 85 (50%) of the struck side occupants were in vehicles with a delta-v of 20-29 km/h compared with only 63 (38%) of the non-struck side occupants. Also 66 (39%) of non-struck side occupants had a vehicle delta-v of 10-19 km/h compared with 42 (24%) of struck side occupants.

Table 23. Police severity of restrained occupants with maximum AIS = 1+.

Severity	Struck side	Non-struck side
Uninjured	11 (3%)	7 (3%)
Slight	229 (56%)	214 (73%)
Serious	158 (39%)	66 (23%)
Fatal	12 (3%)	4 (2%)
Total	410(100%)	292(100%)

Table 24. Maximum AIS of restrained occupants with injuries AIS = 1+.

Maximum AIS	Struck side	Non-struck side
1	215 (53%)	218 (75%)
2	132 (32%)	50 (17%)
3	42 (10%)	10 (4%)
4	7 (2%)	4 (1%)
5	8 (2%)	8 (3%)
6	6 (2%)	1(0.4%)
Total	410(100%)	292(100%)

Struck side occupant injuries

Table 26 shows the distribution of the severities of injury for each body region of the 294 struck-side occupants. Also shown in Table 26 is the contribution towards the total HARM for each body region which was calculated using the weightings derived by Malliaris shown in Table 14.

The importance of injuries to each body area can be assessed using several methods. The proportion of occupants with an injury in a particular body region is a measure of incidence of injury. Using this technique the head is the most frequently injured, 283 (69%) of the 410 of struck-side occupants sustained a head injury.

When the number of occupants with an injury of AIS 3 or above is expressed as a proportion of all injuries, this produced a measure of the severity of those injuries that do occur. Under this measure the abdomen is most frequently the body area sustaining severe injuries, 52% of the injuries found in the

Table 25. Delta-v of struck side and non-struck side occupants vehicles where known (km/h).

Delta-v	Struck side	Non-struck side
1 - 9	8 (5%)	2 (1%)
10 - 19	42 (24%)	66 (39%)
20 - 29	85 (50%)	63 (38%)
30 - 39	21 (12%)	24 (14%)
40 - 49	9 (5%)	6 (4%)
50 - 59	1(0.6%)	2 (1%)
60 - 69	3 (2%)	1(0.8%)
70+	3 (2%)	4 (2%)
Total	171(100%)	167(100%)

Table 26. Distribution of maximum AIS within each body area related to harm for struck side occupants. Numbers of injuries and proportions of known severities.

Body Area		Struck Side of Occupants: Maximum AIS 0	1	2	3	4	5	6	Amount of Harm: No's	%
Head/face	No.	127	143	116	12	6	3	3	2391	33
	%	31	35	28	3	1	0.8	0.8		
Neck	No.	332	76	0	1	0	0	0	50	0.7
	%	81	19		0.3					
Chest	No.	245	123	12	19	1	5	5	2840	39
	%	60	30	3	5	0.2	1	1		
Abdomen	No.	364	22	0	14	5	5	NA	1590	22
	%	89	5		4	1	1			
Arms	No.	255	102	51	1	NA	NA	NA	231	3
	%	62	25	12	0.3					
Legs	No.	276	115	9	10	0	NA	NA	200	3
	%	67	28	2	3					
Total Harm									7302	100%

Table 28. Distribution of maximum AIS within each body area related to harm for non struck side occupants. Numbers of injuries and proportions of known severities.

Body Area		Non-Struck Side of Occupants: Maximum AIS 0	1	2	3	4	5	6	Amount of Harm: No's	%
Head/face	No.	210	52	14	6	1	8	1	2315	76
	%	72	18	5	2	0.3	3	0.4		
Neck	No.	275	14	0	3	0	0	0	37	1
	%	94	5		1					
Chest	No.	185	96	10	0	0	0	0	97	3
	%	63	33	4						
Abdomen	No.	271	9	6	3	4	0	NA	279	9
	%	93	3	2	0.9	1				
Arms	No.	188	77	26	1	NA	NA	NA	141	5
	%	65	26	9	0.3					
Legs	No.	77	211	1	2	0	NA	NA	169	6
	%	27	72	0.4	0.7					
Total Harm									3038	100%

abdomen were AIS 3 or above, a substantially higher proportion than any other body region.

A third measure is to calculate the contribution to the total value of 'harm' found within the side impact population of injuries to each body area. Using these factors chest injuries are the most expensive and contribute 39% of the total harm.

Table 27 summarises the rankings of body areas using each of these three methods. It is considered that the most useful rank order is produced using 'harm' as it is the only scale that reflects both the frequency and severity of injury.

Table 27 shows that chest injuries are frequent, being sustained by 40% of struck side occupants, they are also severe with 18% of injuries being of AIS 3+. Additionally Table 26 shows that 10 out of the 30 AIS 3+ chest injuries have a very heavy weighting of harm. These 3 factors combine to make the chest the most significant in terms of harm.

Table 27 also shows that head/face injuries are very common but seldom severe whilst abdominal injuries are rare but frequently severe. However, it is considered likely that protection against chest injuries is likely to reduce abdominal injuries due to the coupled mechanisms of injury. It should be noted that for the purposes of this analysis the abdomen includes the pelvis and hip joints.

The maximum AIS possible for arm and leg injuries is 4 although such scores are rare. The contribution towards the total value of harm of each body area is therefore low at 3% each.

Finally Table 27 shows that neck injuries are the least commonly injured in struck side impacts and are rarely severe. Neck injuries only contribute 0.7% to the total harm. The harm scale suggests that the reduction of the chest injuries of struck side occupants should be the principle objective of side impact protection. Further objectives should be the reduction of head/face and abdominal injuries in that order.

Non-struck Side Occupant Injuries

The pattern of injuries of the 292 non-struck side occupants in the population is shown in Table 28. It is rare for an injury above AIS 2 to occur in any body region. The rank order of the body areas was calculated using the methods previously applied, and the results are shown in Table 29. This table shows that the greatest contribution to the total harm by far is from head/face injuries. The contribution is 76% and is a result of a 27% incidence rate within the

Table 27. Rank order of body area importance using frequency of injury, frequency of AIS 3+ injury and harm—struck side occupants.

Rank	Body Region and % Injured		Body Region and % of Injuries AIS 3+		Body Region and % of Total Harm	
1	Head/face	69%	Abdomen	52%	Chest	39%
2	Chest	40%	Chest	18%	Head/face	33%
3	Arms	38%	Head/face	8%	Abdomen	22%
4	Legs	33%	Legs	7%	Legs	3%
5	Neck	19%	Neck	1%	Arms	3%
6	Abdomen	11%	Arms	0.6%	Neck	0.7%

Table 29. Rank order of body area importance using frequency of injury, frequency of AIS 3+ injury and harm: non-struck side.

Rank	Body Region and % Injured		Body Region and % of Injuries AIS 3+		Body Region and % of Total Harm	
1	Legs	73%	Abdomen	33%	Head/face	76%
2	Chest	37%	Head/face	20%	Abdomen	9%
3	Arms	35%	Neck	6%	Legs	6%
4	Head/face	28%	Arms	1%	Arms	5%
5	Abdomen	7%	Legs	1%	Chest	3%
6	Neck	6%	Chest	0%	Neck	1%

population and a high proportion of AIS 3+ injuries. Below head/face the abdomen accounted for 9% of the total harm, these injuries being extremely severe amongst non-struck side occupants but only rarely occurring.

The remaining 4 body areas each contributed only a little to the total harm. AIS 3+ injuries were rarely found to occur to the legs, chest or arms. They occurred in 6% of neck injuries but only 6% of non-struck side occupants sustained a neck injury. These 4 body regions accounted for only 15% of the total harm.

Struck and Non-struck Side Occupants Compared

Tables 26 and 28 showed that the total harm to struck and non-struck side occupants was 7302 and 3038 'harm' units respectively. When all body areas of *all* occupants are considered together, the greatest contribution to the total value of 10340 harm units was 2840 harm units from struck side occupants chest injuries. The head/face injuries of struck side occupants followed, this accounting for 2391 harm units. The third highest contribution however was 2315 from the head injuries of non-struck side occupants. Inspection of Tables 26 and 28 shows that 9 non-struck side occupants sustained AIS 5 or 6 head injuries compared with only 6 struck side occupants. The weighting factors of harm give this difference of 3 injuries a value of 633 harm units.

Contact Analysis

As part of the procedure of vehicle examination in the study, the interior was inspected for evidence of occupant contact. This information was later combined with a detailed description of the injuries sustained, coded using AIS. Modern cars tend to be built using materials that do not always show contact evidence particularly if the contact is light. It is therefore not always possible to determine the part of the car that caused the injury and a "not known" code is used in these circumstances.

The coding procedure used allows up to 2 contacts to be associated with each body injury. Also the gross body areas used for this analysis may be constructed of several smaller units. For example, when injuries to the body area "chest" are initially coded they are categorized as being sustained either to the upper back and spine or to the remaining part of the chest. There are therefore potentially 4 contact codes that may be associated with injuries to the body area "chest". For these reasons the numbers of contacts in the tables presented are likely to be different from the numbers of injuries in each body area.

Injuries are only seldom associated with 2 contacts but when this occurs there is no discrimination as to which is the more severe. For example, a common contact in a side impact is with an intruding door supported by a striking vehicle. Both door and vehicle would be given a contact code.

Similarly an occupant's head may break the side window and strike a tree, both window and tree would be coded and would feature in the tables of contacts presented in this analysis. It would not be true to assume that both contacts played an equal role in the causation of the injury.

When the contacts that cause injuries of any severity are examined, this effect is not relevant as every contact caused an injury of some severity. However, amongst the contacts that caused the more severe injuries, the presence of less severe contracts may cloud any conclusions drawn. For example, Table 30 shows both side glass and intruding external objects to be associated with AIS 3+ head injuries. These areas of possible misinterpretation are highlighted in the text.

For each body area the tables presented show the most common contacts that together account for at least 67% of all identified contacts.

Head and Face Injuries

Of the 410 struck side occupants, 283 sustained a classifiable injury to the head or face and when measured using harm this body area ranked second. Of these occupants, the vast majority, 259 (92%), received minor to moderate (AIS 1 and AIS 2) injuries, whilst the remaining 24 (8%) sustained more severe AIS 3+ injuries (see Table 26). Table 30 shows the frequency of contacts causing injuries of any severity and AIS 3+ injuries respectively.

When considering the contacts causing injuries of any severity, the most commonly occurring were with glazing materials; there were 146 (32%) contacts from this source. The steering wheel was contacted 47 times. All of these occupants involved except one were involved in multiple impacts, one of which was a

Table 30. Contacts causing the head and face injuries of restrained struck side occupants.

Struck Side Occupants					
All Head/Face Injuries			AIS 3+ Head/Face Injuries		
Contact Source	No. of Contacts	Relative Frequency %	Contact Source	No. of Contacts	Relative Frequency %
Glazing materials	146	32	Intruding external object	28	52
Steering wheel	47	15	Side window glass	17	31
Intruding external object	40	12	Other known contacts	19	17
Side header rail	32	10			
Other known contacts	57	31			
Total	322	100%	Total	54	100%

frontal. The next most frequent was from intruding external objects accounting for 40 (12%) of the contacts. (This includes such objects as other vehicles, roadside furniture etc, without the occupant being ejected. There were 32 (10%) contacts with the side header rail. These contact sources accounted for 79% of the known injury causes.

If the contacts causing the AIS 1 and AIS 2 injuries within this group are excluded then there are 54 remaining contacts causing AIS 3+ injuries; the majority, 28 (52%) were from intruding external objects and 17 (31%) were from side window glass. These 2 contacts accounted for a total of 73% of the known injury causes. Because more than one object can be contacted by the same body area, the raw data was examined revealing that those occupants sustaining AIS 3+ injuries from side window glass also made contact with an intruding external object. Therefore, the most significant cause of serious to fatal head injuries for struck side occupants is from intruding external objects.

Table 28 showed that only 82 non-struck side occupants sustained head or face injuries and of these 66 (80%) had AIS 1 or AIS 2 injuries whilst 16 (20%) sustained AIS 3+ injuries. The analysis also shows that there were slightly less than one third the number of non-struck side occupants receiving a head or face injury when compared with those on the struck side. However, in terms of harm, they were the most important body area to be injured by non-struck side occupants.

The contacts causing these injuries at any severity level is shown in Table 31 where it can be seen that, as for the struck side occupants, glazing materials are the most common source of injury (27%). However the second most common cause is quite different, coming from contact with another occupant (20%) rather than from an intruding external object. There were 10 (16%) contacts with the facia and 7 (11%) cases of injury without contact. These contacts account for 66% of all known sources.

Table 31 also shows that AIS 3+ injuries occur most commonly from contact with a door (24%), other occupants (20%) and then equally frequent from A-pillar (17%) and rear windows. It should be noted however, that the total number of non-struck side occupants sustaining AIS 3+ injuries was small, there were only 13 contacts.

Table 31. Contacts causing the head and face injuries of restrained non-struck side occupants.

Non-Struck Side Occupants					
All Head/Face Injuries			AIS 3+ Head/Face Injuries		
Contact Source	No. of Contacts	Relative Frequency %	Contact Source	No. of Contacts	Relative Frequency %
Glazing materials	17	27	Door	3	24
Other occupant	12	20	Other occupant	3	20
Facia	10	16	A-pillar	2	17
Injury without contact	7	11	Rear window	2	17
Other known contacts	15	26	Other known contacts	3	20
Total	61	100%	Total	13	100%

Neck Injuries

Table 26 showed that only 77 (19%) of the 410 struck side occupants received a neck injury. Of these, 76 (97%) received AIS 1 injuries and only 1 sustained an AIS 3 injury. The low numbers of neck injuries is also reflected in Table 27 where they are shown as the least important in terms of harm, accounting for only 0.7 of the total harm distribution among all body areas.

Table 32 shows that by far the most common cause was non-contact injuries (ie deceleration without contact); a total of 55 contacts 36 (65%), were of this type. The single AIS 3 injury was caused by an unidentified contact.

Table 28 showed that a lesser proportion of non-struck side occupants sustained a neck injury. There were 17 such casualties representing only 6% of non-struck side occupants. Neck injuries were also the least important in terms of harm. Of the 17 neck injuries, 14 (82%) were AIS 1 and 3 were AIS 3. The contacts causing these injuries are shown in Table 33 and the findings are similar to those for struck side occupants with the vast majority sustaining their injuries as a result of deceleration without contact. The contacts causing the AIS 3 neck injuries were from an unknown source.

Chest Injuries

It can be seen from Table 27 that chest injuries for struck side occupants are the most important in terms of harm, accounting for 39% of the total harm distributed amongst all the body areas. 165 struck side occupants received classifiable injuries and of those

Table 32. Contacts causing the neck injuries of restrained struck side occupants.

Struck Side Occupants					
All Neck Injuries			AIS 3+ Neck Injuries		
Contact Source	No. of Contacts	Relative Frequency %	Contact Source	No. of Contacts	Relative Frequency %
Non-contact injury	36	65	Non-identified	0	
Other occupant	8	15			
Other known contacts	11	20			
Total	55	100%	Total	0	

Table 33. Contacts causing the neck injuries of restrained non-struck side occupants.

Non-Struck Side Occupants					
All Neck Injuries			AIS 3+ Neck Injuries		
Contact Source	No. of Contacts	Relative Frequency %	Contact Source	No. of Contacts	Relative Frequency %
Non contact injury	5	83	None identified		
Other known contacts	1	17			
Total	6	100%	Total	0	

Table 35. Contacts causing chest injuries of restrained non-struck side occupants.

Non-Struck Side Occupants					
All Chest Injuries			AIS 3+ Chest Injuries		
Contact Source	No. of Contacts	Relative Frequency %	Contact Source	No. of Contacts	Relative Frequency %
Seat belt webbing	71	78	No injuries	0	0
Other known contacts	19	22			
Total	90	100%	Total	0	

125 (76%) were AIS 1 or AIS 2. The remaining 40 (14%) were AIS 3+ injuries (see Table 26).

Table 34 shows the frequency of contacts occurring for any severity of chest injury to struck side occupants. From a total of 172 contacts, 84 (48%) were from seat belt webbing closely followed by the door 75 (43%) as the next most common. These 2 contact sources accounted for 91 of all the known injuries. If all the AIS 1 and AIS 2 contacts are excluded then the most common frequent contact causing an AIS 3+ injury is the door (66%).

A total of 107 non-struck side occupants received chest injuries and all were either AIS 1 or AIS 2. Harm ranks these injuries fifth, only 2% lower than arm injuries but significantly lower than the head injuries for non-struck side occupants.

Table 35 shows that the most common cause of chest injuries for non-struck side occupants is seat belt webbing. Of the 90 contacts the vast majority 71 (78%) were from this source.

Abdomen Injuries

Table 26 showed that 46 struck side occupants received injuries to the abdomen representing 11% of the total population of struck side occupants. These injuries ranked third measured using harm (22%). Twenty-two (48%) received AIS 1 or AIS 2 injuries and the remaining 24 (52%) received more serious AIS 3+ injuries. The contacts causing these injuries are shown in Table 36.

When considering the causes of abdomen injuries for any severity the door and door furniture are by far the most common (70%). Of these 101 contacts 82 (59%) were caused by the door, 15 (11%) by the door handle and 4 (3%) by the door bin. Amongst those with AIS 3+ injuries there were 33 (57%) door contacts and 17 (30%) contacts with intruding external objects.

Although the door is shown to be the most important cause of both slight and severe injuries to the abdomen, it is quite likely that any one occupant could have received the same injury by contacting both the door and the intruding external object, if that object was supporting the door at the time of contact. Further investigation of the raw data revealed that this was the case for the 17 AIS 3+ contacts, thus; intruding external objects such as other vehicles and roadside furniture, were by far the most injurious abdomen contact for struck side occupants.

Table 28 showed that 22 (7%) of the non-struck side occupants received an injury to the abdomen. However, the majority 15 (68%) were slight to moderate AIS 1 and AIS 2 injuries, with 7 (32%) AIS 3+ injury. Measured using harm these injuries ranked second (9%), only 3 more than leg injuries, but they were substantially less important than head injuries for non-struck side occupants. The analysis shows, therefore, that the population by far the most likely to receive a serious abdominal injury are struck side occupants.

Table 34. Contacts causing the chest injuries of restrained struck side occupants.

Struck Side Occupants					
All Chest Injuries			AIS 3+ Chest Injuries		
Contact Source	No. of Contacts	Relative Frequency %	Contact Source	No. of Contacts	Relative Frequency %
Seat belt webbing	84	48	Door	25	66
Door	75	43	Other known contacts	11	34
Other known contacts	13	9			
Total	172	100%	Total	36	100%

Table 36. Contacts causing the abdomen injuries of restrained struck side occupants.

Struck Side Occupants					
All Abdomen Injuries			AIS 3+ Abdomen Injuries		
Contact Source	No. of Contacts	Relative Frequency %	Contact Source	No. of Contacts	Relative Frequency %
Door and door furniture	101	73	Door and door furniture	33	57
Other known contacts	36	27	Intruding external object	17	30
			Other known contacts	7	13
Total	137	100%	Total	57	100%

Table 37. Contact causing the abdomen injuries of restrained non-struck side occupants.

Non-Struck Side Occupants					
All Abdomen Injuries			AIS 3+ Abdomen Injuries		
Contact Source	No. of Contacts	Relative Frequency %	Contact Source	No. of Contacts	Relative Frequency %
Seat belt webbing	43	70	Seat belt webbing	4	42
Other known contacts	18	30	Other occupant	3	34
			Other known contacts	2	24
Total	61	100%	Total	9	100%

The contacts causing the abdomen injuries to non-struck side occupants are shown in Table 37. Only one contact type, seat belt webbing accounted for 70% of all the non-struck side occupants' abdomen injuries. The belt (42%) and other occupant contacts (34%) were the most frequent source of AIS 3+ injuries.

Arm Injuries

Table 26 showed that 155 of the 410 struck side occupants had an injury to the arm with a classifiable AIS. Arm injuries, however, ranked fifth when measured using harm. The contacts by the arms of these 155 occupants are shown in Table 38. The most common contact was with the door, occurring 71 (49%) times.

Table 26 showed only one occupant sustaining AIS 3 arm injuries so for this region the contacts causing AIS 2 and 3 injuries are shown separately also in Table 38. This table shows that of the more serious injuries to arms 48 (71%) also arise from door contact.

Table 39 shows the sources of arm injuries amongst non-struck side occupants. There were 104 such occupants with an arm injury but only 50 contacts were identified. The most common contact to cause an injury of any severity was with another occupant accounting for 22 (43%) of all contacts. Other relatively frequent contacts were flying glass and clothing. The low total number of contacts means it is difficult to place a reliable rank order on these contact regions. There were 18 identified arm contacts causing an AIS 2+ injury, all were with another occupant.

Table 39. Contacts causing the arm injuries of restrained non-struck occupants.

Non-Struck Side Occupants					
All Arm Injuries			AIS 2+ Arm Injuries		
Contact Source	No. of Contacts	Relative Frequency %	Contact Source	No. of Contacts	Relative Frequency %
Other occupant materials	22	43	Other occupant	18	100
Flying glass	7	13			
Clothing	6	12			
Other known contacts	15	33			
Total	50	100%	Total	18	100%

Leg Injuries

Leg injuries ranked fourth in Table 27 when harm was considered although they are relatively frequent, 33% of struck side occupants sustaining these injuries. Table 40 shows the contacts causing the leg injuries of struck side occupants for injuries of all severities. The contacts for AIS 2+ leg injuries are also shown (AIS 3 leg injuries being rare).

The contacts causing leg injuries were the same for both groups of severities. The most common contact was the door accounting for 155 (72%) of injuries of any severity and 20 (49%) of AIS 2+ injuries.

Amongst AIS 2+ injuries the second most common was the intruding footwell and pedals, accounting for 15 (36%) of contacts.

Table 28 showed that leg injuries were sustained by 215 (73%) of all non-struck side occupants. Of these 211 (98%) were AIS 1 but leg injuries, while contributing only 6% of the total harm, ranked second. Table 41 shows the contacts that were associated with these injuries. There were 227 identified contacts associated with injuries of any severity. The most common two were with the centre console and the lower facia. These two contacts occurred 68 (30%)

Table 38. Contacts causing the arm injuries of restrained struck side occupants.

Struck Side Occupants					
All Arm Injuries			AIS 3+ Arm Injuries		
Contact Source	No. of Contacts	Relative Frequency %	Contact Source	No. of Contacts	Relative Frequency %
Door	71	49	Door	48	71
Glazing materials	38	26	Other known contacts	20	29
Other known contacts	36	25			
Total	145	100%	Total	68	100%

Table 40. Contacts causing the leg injuries of restrained struck side occupants.

Struck Side Occupants					
All Leg Injuries			AIS 2+ Leg Injuries		
Contact Source	No. of Contacts	Relative Frequency %	Contact Source	No. of Contacts	Relative Frequency %
Door	155	72	Door	20	49
Other known contacts	60	28	Intruding footwell	15	36
			Other known contacts	6	15
Total	215	100%	Total	41	100%

Table 41. Contacts causing the leg injuries of restrained non-struck side occupants.

Non-Struck Side Occupants					
All Leg Injuries			AIS 2+ Leg Injuries		
Contact Source	No. of Contacts	Relative Frequency %	Contact Source	No. of Contacts	Relative Frequency %
Centre console	68	30	Gear lever	1	50
Lower facia	64	28	Self	1	50
Pedals and bracketry	23	10			
Other known contacts	72	32			
Total	227	100%	Total	2	100%

and 64 (28%) times respectively. There were also 23 (10%) contacts with the pedals and bracketry.

There were only 2 identified contacts associated with the more severe injuries, these were with the gear lever and another part of the occupant.

Other Impact Configurations

Table 5 showed that 6 per cent of the impacts were to the rear and 4 per cent involved a rollover. Tables 42 and 43 show the injury severity distribution for restrained front seat occupants where the vehicles have been involved in these two impact configurations.

This shows the outcome in rear impacts is less severe than for frontal and side accidents ie, two-thirds were uninjured and a third only sustained an injury of MAIS 1. The estimated mean $\triangle V$ was 26 km/h.

Not surprisingly, only a third of the occupants were uninjured in rollovers and half had a maximum severity level of MAIS 1. Multiple contact must also be a feature of this type of impact as the restraint may not be performing in its normal mode of operation and webbing could be slack. It is often difficult to ascribe contact points for the injuries sustained to occupants in this type of accident. It is conceivable that belt pre-tensioners may assist in reducing injuries provided they can be initiated in this type of impact. Selective padding will also assist when further information is available on the structure contacted.

Table 42. Maximum injury severity distribution—restrained front occupants in rear impacts (weighted data)..

Position \ Severity	Uninjured (%)	MAIS 1 (%)	MAIS 2-3 (%)	MAIS 4-6 (%)	Total N (%)
Driver	(64)	(33)	(2)	<1	481 (100)
Front Passenger	(64)	(31)	(5)	-	104 (100)
Total N (%)	378 (65)	190 (32)	15 (3)	2 (<1)	585 (100)

Table 43. Maximum injury severity distribution.

Position \ Severity	Uninjured (%)	MAIS 1 (%)	MAIS 2-3 (%)	MAIS 4-6 (%)	Total N (%)
Driver	(29)	(52)	(16)	<1	256 (100)
Front Passenger	(32)	(52)	(13)	4	104 (100)
Total N (%)	108 (30)	188 (52)	54 (15)	6 (2)	359 (100)

Conclusions

General

This Paper has reported on the results of a comprehensive crash-injury study currently underway in England. Weighted data has been used throughout and compared with national accident figures where appropriate. The following is concluded:

The study sample is reasonably representative of national accident data.

Although compulsory seat belt legislation has brought about a wearing rate of approximately 95 per cent amongst front seat car/van occupants, there is still cause for concern that over 2,000 occupants are still being killed and 27,000 seriously injured per year. However the driver casualty rate is not increasing at the same rate as traffic volume. An estimated 94 per cent of the rear seat occupants in the sample were unrestrained. It is hoped that the introduction of compulsory fitting of rear belts in new cars from April 1987 will encourage their use and reduce mortality and injury amongst rear seat occupants. Problems that may arise in respect of belt mounting and body fit will have to be monitored as usage increases.

An analysis of the objects impacted shows that the majority are struck by other cars. This fact, and also the other structures impacted indicates that the car structure and any improvements to it, still have a major part to play in occupant protection.

The majority (67 per cent) of impacts were found to be to the front of the vehicle with a significant number (23 per cent) to the side—with rather more to the driver's side than to the nearside.

Young children featured highly amongst the unrestrained occupants.

Frontal Impacts

Two thirds of the impacts in this direction involved the front centre with a high proportion extending across the whole front. Corner impacts also occurred as an important sub-group.

Changes of speed on impact ($\triangle V$) indicates the majority were in the 11-40 km/h band.

Injury severities of MAIS 3+ to restrained occupants are likely to result from impacts in excess of 45 km/h. (△V).

Assessment of injury severity shows the head and chest to be the most vulnerable areas with contact to the steering wheel featuring highly as an injury mechanism - either as direct contact or when the wheel itself has intruded into the passenger compartment. Contact with the 'A' pillar and by the other vehicle in the accident also produced a number of injuries. It is reasoned that seat belt pre-tensioners could greatly assist in preventing contact or intrusion-related injuries as described.

Footwell intrusion was responsible for 51 per cent of lower limb injuries. Although not life threatening, this type of injury could lead to prolonged recovery times and permanent disability in later life.

The effect of head restraints in reducing neck sprains amongst restrained occupants in frontal collisions requires monitoring as the introduction of such restraints increases.

Side Impacts

To reproduce the characteristics of side impacts of all severities of injury, a test would be perpendicular to the side of the car involving A and B pillars at a △V of 21-28 km/h.

To reproduce fatal side impacts a test could be angled 30° forward of perpendicular, with an overlap of 20 per cent of the car length between B pillar and wheelbase center at an average △V of 51 km/h. This represents a balance between fatal impacts of much lower speeds and very severe impacts in which the car was completely destroyed. It is not feasible to build in protection for the latter but might be realistic to attempt to prevent fatalities that occur at the lower edge of the speed range.

Within the population studied, restrained non-struck side occupants were less severely injured than restrained struck side occupants. They did, however, tend to be involved in impacts with a lower △V.

The use of 'harm' measures suggests that priority should be given to the reduction of struck side occupants chest and head injuries, non-struck side occupants head injuries and struck side occupants abdomen injuries, in that order.

The most common contacts associated with struck side occupants AIS 3+ chest injuries are: side door (66 per cent), often supported by an intruding external object.

The most common contact associated with struck side occupants AIS 3+ head injuries is an intruding external object (51 per cent). The most common contacts associated with non-struck side occupants AIS 3+ head injuries are: door (24 per cent), other occupant (20 per cent).

The most common sources of struck side occupants AIS 3+ abdominal injuries are the side door (40 per cent), including those supported by an intruding external object (28 per cent).

Concluding remarks

Further analyses of the growing database derived from real-world accidents will enable detailed information to be presented to design engineers. This will take many forms and greatly assist legislators and others responsible for proposed type-approval test procedures.

This Paper represents a significant analysis of a comprehensive crash injury database which in turn represents the efforts of many people. Other analyses will doubtless be refined in the future with many detailed studies of particular aspects of the total problem.

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Experience from the Analysis of Accidents with a High Belt Usage Rate and Aspects of Continued Increase in Passenger Safety

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Abstract

This paper discusses the experience obtained from accident research which resulted from the increase in the belt-wearing rate, now over 90% in Germany. The changes in the accident characteristics resulting from this high belt-wearing rate, for example the relative importance of head-on and side-collisions are examined. The new priorities in the case of "injuries in spite of fastened safety belts" are described.

The risk to the rear-seat passenger and the problem of interaction between belted front and unbelted rear-seat passengers are discussed, and possible improvements derived from real-live accidents are presented.

On the basis of the "injury characteristics in spite of fastened seat belts" now available proposals are made for futher increasing vehicle safety, especially with regard to optimizing the steering column, to the use of a belt pretensioner, and the airbag.

The History of Wearing Safety Belts in Germany and the Effects of a High Belt-Wearing Rate

In the last 30 years the number of road traffic accidents in Germany has roughly increased threefold (in 1985: 1,840,295), the number of fatalities rising from about 13,000 in 1957 to more than 19,000 in 1970 and then declining continuously to 8,400 in 1985 [1].

The number of car occupants killed also showed a similar trend to the total number of fatalities; thus in 1970 about 9,000 people died in cars (the peak was reached in 1972 with just under 9,500), and in 1985 it had come down to only 4,182. This drastic drop in the number of car occupants killed is closely connected with the development of the use of a safety belt in Germany. On 1st July, 1986, what was for the time being the last chapter in the history of safety belt use was concluded when a fine was introduced for failure to use a seat belt on the rear-seat as well.

With regard to the use of the safety belt in Germany the following regulations were of decisive importance [2]:

- As from 1.1.1974 it was compulsory for the front-seats of all new cars to be equipped with belts,
- as from 1.1.1976 it was compulsory for the driver and the front-seat passenger to wear belts; it was made compulsory for all cars registered after 1.4.1970 to be subsequently fitted with seat belts,
- as from 1.5.1979 it was compulsory for all new cars to be fitted with seat belts for the rear-seats,
- as from 1.8.1984 a fine of 40.- DM was introduced for failure to wear belts on the front-seats; it was made compulsory for rear-seat passengers to wear belts,
- as from 1.7.1986 a fine of 40.- DM was introduced for failure to wear safety belts on the rear-seats.

It is especially the introduction of a fine for failure to wear safety belts on the front-seats as from August 1984 which has resulted in a drastic reduction in the number of car occupants killed and injured, as the belt-wearing rate on the front-seats jumped from about 60% to over 90% (Figure 1). In the period

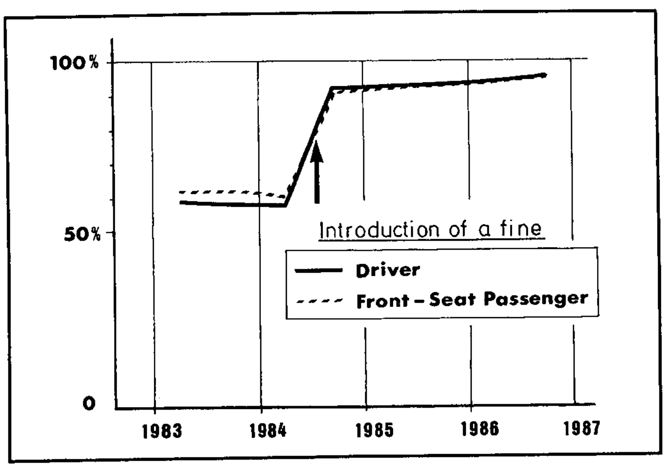

Figure 1. Average rate of belt-wearing for drivers and front-seat passengers in Germany

Car occupant casualties	August - July 1983/84	August - July 1984/85	change
Fatalities	5,744	4,283	−1,461 −25.4%
Seriously injured	67,131	52,040	−15,091 −22.5%

Figure 2. Drop in figures of injured car occupants in period from August 1984 to July 1985 compared with previous year

from August 1984 to July 1985 the number of car occupants killed dropped by 1,461 (Figure 2) and the number of seriously injured car occupants by 15,091 in comparison with the previous year [3].

A reduction in the injury figures on this scale had already been predicted at the beginning of the 80s in the light of a HUK-prognosis model. The principle of the prognosis model, which is described in more detail in [4], is based on the fact that risk rates for belted and unbelted front-seat car occupants can be derived from real-life accidents (Figure 3) and they also make it possible to estimate the influence of a change in the belt-wearing rate on the number of car accident casualties. The correctness of the risk rates ascertained for drivers and front-seat passengers can be tested by comparing the estimated numbers of injured persons in the case of a known, actual belt-wearing rate with the number of persons injured as given by the official statistics. It may be assumed that the risk rates are correct if there is a high measure of agreement in this comparison.

The sudden increase in the belt-wearing rate on the front-seats from 60% to over 90% has now provided the possibility of testing the HUK-prognosis model in a "large-scale test", and it has been shown that there

Injury severity	Driver: Relative Risk with belt : without belt	Driver: Risk Distribution with belt	Driver: Risk Distribution without belt	Front-Seat Passenger: Relative Risk with belt : without belt	Front-Seat Passenger: Risk Distribution with belt	Front-Seat Passenger: Risk Distribution without belt
Fatal	1 : 1.8	1.0%	1.8%	1 : 1.8	1.1%	2.0%
Serious	1 : 1.6	11.0%	18.0%	1 : 1.8	14.0%	25.0%
Minor	1 : 1.6	31.0%	51.0%	1 : 1.2	49.0%	59.0%
Uninjured	—	57.0%	29.2%	—	35.9%	14.0%
Total	—	100.0%	100.0%	—	100.0%	100.0%

Figure 3. Relative risk and risk distribution of driver and front-seat passenger, with/without belt

	Reduction in injured car occupants	
	Actual reduction with 90-95% belt-wearing rate	HUK prognosis with 100% belt-wearing rate
Fatalities	– 1,461	– 1,487
Seriously injured	–15,091	– 15,037

Figure 4. Comparison of HUK-prognosis with actual reduction in car occupants injured between August 1984 and July 1985 compared with previous year

is a high measure of agreement between the prognosis model and the accident scene in real life (Figure 4).

Because of the very high belt-wearing rate and the reduction of people killed and injured resulting from this it can be assumed that the injury consequences in the car have changed since August 1984 and, moreover the priorities in general have shifted somewhat in the car accident scene: the high belt-wearing rate has increased the significance of side-collisions, since the benefit of the belt in side-collisions is less than in head-on collisions, although the dominant position of head-on collisions still remained unchanged.

The Significance of Different Types of Collision

Frequency of collision types in car-to-car accidents

The car-to-car accidents can be broken down into a large number of different collision types [5], but in order to avoid splitting them up too much it is necessary to limit the types to a reasonable number. For the purposes of HUK accident research the 12 main collision types presented in Figure 5 are chiefly used for classifying car-to-car accidents.

The frequency distribution of the individual collision types, based on some 9,000 car-to-car accidents from the HUK accident material, shows a high proportion of the types referred to as "rear/front" and "side/front", and the types termed "front/front" come only in third position; considering the severity of collisions, however, the head-on collisions, as shown below, come first.

Frequency of collision types in single-car accidents

For the analysis and systematic presentation of single-car accidents collision types were also defined, a similar procedure to that adopted for car-to-car accidents. The frequency distribution for about 3,000 single-car accidents is given in Figure 6. Here the head-on collisions dominate, followed by the side-

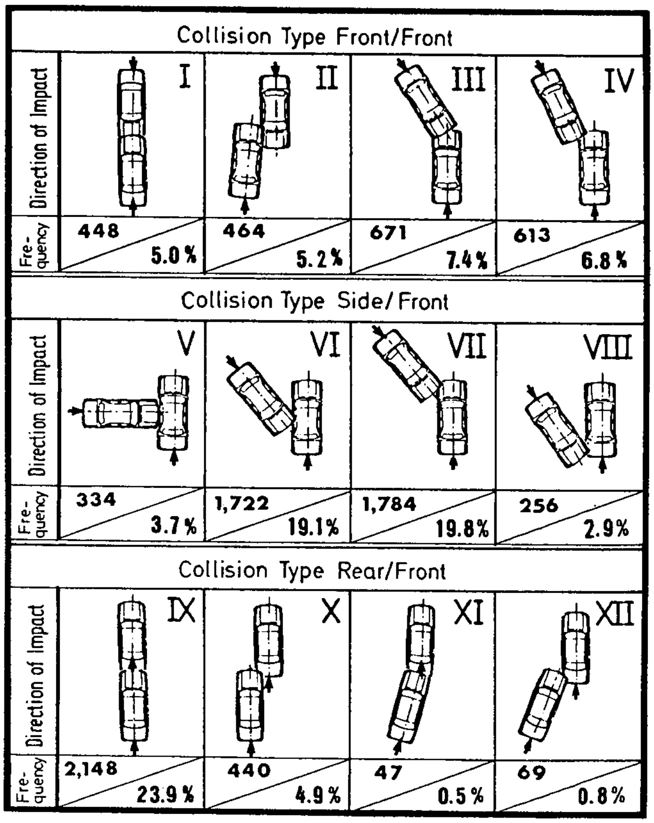

Figure 5. Frequency of collision types in car-to-car accidents

collisions. More detailed explanations can be found in [6].

The relevance of collision types in car-to-car and single-car accidents

In answering the question as to which accident constellation is of outstanding significance for the safety of the car occupants it must, however, not be assumed that individual collision types occur with relative frequency, but what is decisive is the frequency and severity of the injuries arising in the case of the individual collision types. In [6] it was stated that—assuming a belt-wearing rate of 100—about 50% of the car occupants fatally injured in Germany each year in the car-to-car collision types I/III and V/VI and in the single-car accident collision types 02 and 08 would be killed (Figure 7).

Figure 7 shows that in the case of a belt-wearing rate of 100%, the side-collisions (collision type V/VI and 08) are of very great significance but that nevertheless most of the car occupants killed lost their lives in head-on collisions (collision types I/III and 02). For this reason priority must still be given to optimizing the vehicle's *front* in our efforts to achieve greater passive safety. This is all the more important in view of the fact that in [7] it was stated that a considerable part of occupant protection in side-

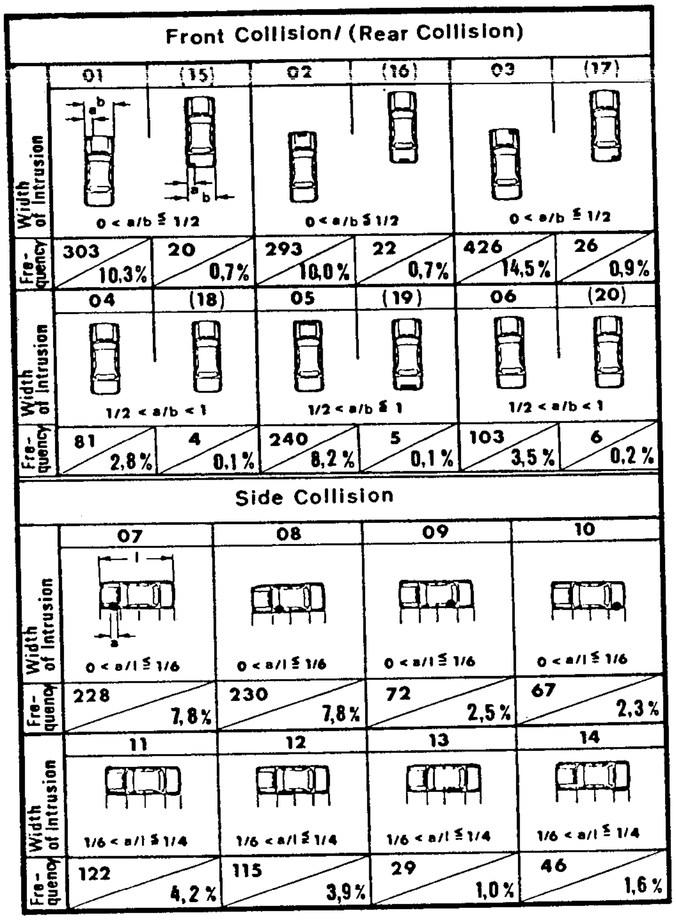

Figure 6. Frequency of collision types in single-car accidents

collisions can be attained by the appropriate design of the front structures of impacting vehicles.

Injury Patterns of Belted Front-Seat Car Occupants

The HUK-Verband's current car accident material now comprises about 1,200 detailed evaluations of accidents with belted occupants and with vehicles which are no older than 5 years. For the following

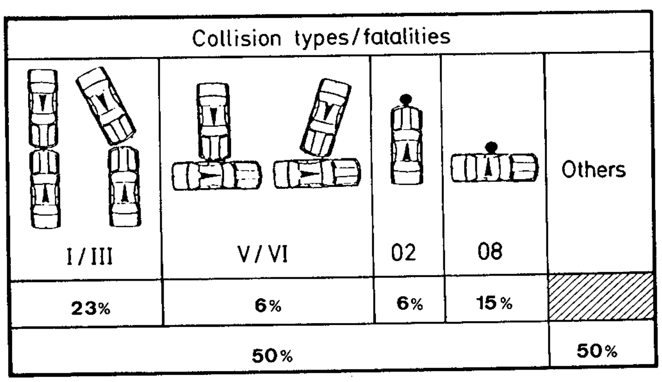

Figure 7. Distribution of car occupants killed to the relevant collision types

	MAIS Driver with belt						Total
	0	1	2	3	4/5	6	
Number	76	290	78	28	15	12	499
%	15.2	58.2	15.6	5.6	3.0	2.4	100.0

Figure 8. MAIS distribution of belted car drivers in head-on collisions

considerations only those cars were selected which were involved in a head-on collision with another car or a truck; the speed change of the cars damaged at the front was 15 km/h and above.

Drivers with belt (number: 499)

The frequency distribution of the maximum injury severity (MAIS) [8] for the belted driver in head-on car collisions is given in Figure 8. Out of the total of 499 belted drivers 76 (15.2%) remained uninjured. The proportion of slightly injured occupants (MAIS 1) was just under 60%, and the proportion of fatalities was 2.4%.

Figure 9 shows the distribution of severity of the individual injuries for three parts of the body: head, chest and abdomen, i.e. for those parts of the body which play an important part in a car crash test. Of the three parts of the body considered the head is altogether injured most frequently. Of the 423 injured belted drivers, 213 (50.4%) sustained an injury to the head, 146 (34.5%) a chest injury and 35 (8.3%) an abdominal injury.

This results in the following injury priorities for the belted driver:

- Head injuries are still the dominant injuries and naturally also account for the largest proportion of serious injuries. The decisive question is to what accident intensity critical head injuries can be delayed by wearing a belt, which causes then are present when the serious head injury occurs in spite of a belt and which further counter-measures can be imagined.
- Chest injuries come second as far as the frequency is concerned; serious internal injuries in the region of the chest are, however,

Injury severity / Location of injury	AIS Driver with belt: 1		2		3		4/5		6		Total		Relative proportion of injuries Basis: 423 injured ≙ 100.0%
	Number	%	Number	%	Number	%	Number	%	Number	%	Number	%	
Head	161	75.6	32	15.0	5	2.3	8	3.8	7	3.3	213	100.0	50.4%
Chest	123	84.2	16	11.0	5	3.4	1	0.7	1	0.7	146	100.0	34.5%
Abdomen	22	62.9	4	11.4	5	14.3	1	2.9	3	8.5	35	100.0	8.3%

Figure 9. AIS distribution to head, chest and abdomen for belted drivers in head-on collisions

very rare and injuries to the internal organs seem to remain within limits. On the basis of experience gathered from the HUK accident material so far it can be stated that the current biomechanical loading when wearing a seat belt is in keeping with human tolerance levels and does not result in overloading.

- Although abdominal injuries are very rare, when they do arise they relatively frequently result in a high level of injury severity. In this connection rate, but nevertheless repeatedly occurring, hidden injuries (ruptures of the liver or spleen) are typical; but the considerably improved design of the lap belt in more modern vehicles (the lock of the belt is fastened to the seat) has further reduced the frequency of these injuries. An increase in the number of injuries caused by submarining has not been observed in Germany.

Front-seat passengers with belt (number: 207)

The frequency distribution of the maximum injury severity (MAIS) for the belted front-seat passenger is shown in Figure 10. Although only 5.8% of the belted front-seat passengers remained uninjured (drivers: 15.2%), the proportion of fatalities—compared with the driver—is lower, namely 1.0% (drivers: 2.4%). In Figure 11 the severity of injury to the three important parts of the body head, chest and abdomen is again indicated. A completely different distribution of injuries emerges compared with the belted driver: here it is chest injuries which are most frequent (44.1%), followed by head (39.0%) and abdominal injuries (9.2%).

The injury priorities for the belted front-seat passenger can be summarised as follows:

- Injuries to the chest occur most frequently; but they are usually of a slight nature (AIS 1). Critical/dangerous chest injuries are even more rare than with the driver. Although the front-seat passenger is restrained by the belt alone without any other particular support in an accident, his biomechanical tolerance levels are also maintained.

	MAIS Front-seat passenger with belt 0	1	2	3	4/5	6	Total
Number	12	133	42	15	3	2	207
%	5.8	64.3	20.3	7.2	1.4	1.0	100.0

Figure 10. MAIS distribution of belted front-seat passengers in head-on collisions

Injury severity / Location of injury	AIS Front-seat passenger with belt 1 Number	1 %	2 Number	2 %	3 Number	3 %	4/5 Number	4/5 %	6 Number	6 %	Total Number	Total %	Relative proportion of injuries Basis: 195 injured ≙100.0%
Head	57	75.0	17	22.4	1	1.3	1	1.3	–	–	76	100.0	39.0%
Chest	72	83.7	10	11.6	4	4.7	–	–	–	–	86	100.0	44.1%
Abdomen	16	88.8	1	5.6	–	–	1	5.6	–	–	18	100.0	9.2%

Figure 11. AIS distribution to head, chest and abdomen for belted front-seat passengers in head-on collisions

- Head injuries come second with regard to frequency; the absence of the "steering wheel" as an injury risk and the greater possible space for the body to move forward are reflected in the low frequency of head injuries compared with the driver. Critical/dangerous head injuries are also more rare than in the case of the driver. Critical head injuries to the front-seat passenger can be observed expecially if the space available to move is considerably restricted by massive intrusion, i.e. by the instrument panel being pushed backwards or by penetration by external objects. Of course, here the possible protective effect of the belt is exhausted and it can thus be seen that an adequate assessment of the injury can only be made if at the same time the circumstances of the accident are known. In the case of no intrusion taking place, however, the front-seat passenger always has a lower head injury risk than the driver, as direct comparisons of head-on collisions show.
- Abdominal injuries occur relatively rarely; if they do occur, however, they account for a not inconsiderable proportion of the critical/dangerous injuries. Here the same applies as for the driver.

It can be observed for both the driver and the front-seat passenger that foot and/or leg injuries have declined considerably; this applies, however, only if no intrusion of the foot or leg space takes place. Especially in head-on collisions with the high offset, however, relatively frequently extreme local intrusion takes place (Figure 12), which then often results in very serious thigh injuries—an accident sequence in which the protective function of the belt ought better to be complemented by additional structural measures to the vehicles.

Spinal injuries to drivers and front-seat passengers were not conspicuous in the material available to date. Although the restraining effect of the belt has led to a certain increase in the slight or, at worst, moderate whiplash injuries, severe spinal injuries to belted occupants hardly ever occur.

Figure 12. Typical car damage with high offset and extensive intrusion on the left in the area of the driver

Figure 13. Head-on collision of moderate severity with unbelted rear-seat occupant and the consequence of very severe eye injuries caused by impact with the windscreen

In assessing the new characteristics of the injuries in spite of the use of a safety belt, one must, however, always be aware of the fact that the injuries to the individual parts of the body have been drastically reduced compared to those sustained by unbelted occupants; thus, for example, the decline in head injuries to the driver and front-seat passenger amounts to around 75%.

It is also known from cooperation with eye hospitals in Germany that the extremely severe eye injuries of earlier years have been reduced by the use of the belt by 46% to 94%. Each year in Germany the use of belts saves at least 400 people from serious eye injuries, often resulting in blindness.

Injury Risk of Rear-Seat Occupants

Up to now, safety questions concerning the rear-seat passengers have only been dealt with in a few works in comparison with studies on the front-seat occupants. Generally it was assumed that the rear-seat passengers ran relatively little risk simply because of their seating position. The idea that the rear-seat provides special protection, however, is not in keeping with the facts.

Our studies [9] have shown that rear-seat passengers who are not wearing a safety belt reveal roughly the same risk as the unbelted driver with regard to the injury severity and the injuries to the different parts of the body.

But wearing belts on the back seats is not only necessary for the protection of the rear-seat passengers, but failure to wear a seat belt in the rear-seat also endangers the other passengers in at least 13% of the car accidents. Rear-seat occupants who have not fastened their safety belts can cause considerable, additional loading of the belted front-seat occupants, although the belt in the case of the driver and the front-seat passenger still affords some benefit but cannot develop its optimum effect.

In head-on collisions the unbelted rear-seat passenger can even be thrown forward as far as the windscreen; cases have even come to light in which these accidents have caused serious eye injuries (Figure 13).

Even in the case of head-on collisions with a speed change of about 40 km/h, the front-seat can often no longer provide any protection, so that the rear-seat passenger then strikes the roof of the car with particular frequency (Figure 14) and often sustains serious/critical spinal injuries.

Protective measures by interior design for the rear-seat passengers themselves and for avoiding mutual injury risks of the occupants sitting in front and in the rear are possible and necessary; but on the rear-seat, too, any decisive improvement in safety can only be achieved by wearing the safety belt.

Although since August 1984 in Germany it has been compulsory for rear-seat passengers to wear a safety belt, even with a fine being imposed since the 1st of July 1986, nowadays the belt-wearing rate of rear-seat occupants is on average only 41% in relation to all cars and 52% in relation to the cars with rear-seat belts, assuming that 80% of the cars are fitted with belts [2].

Figure 14. Impact of an unbelted rear-seat passenger against the car's roof

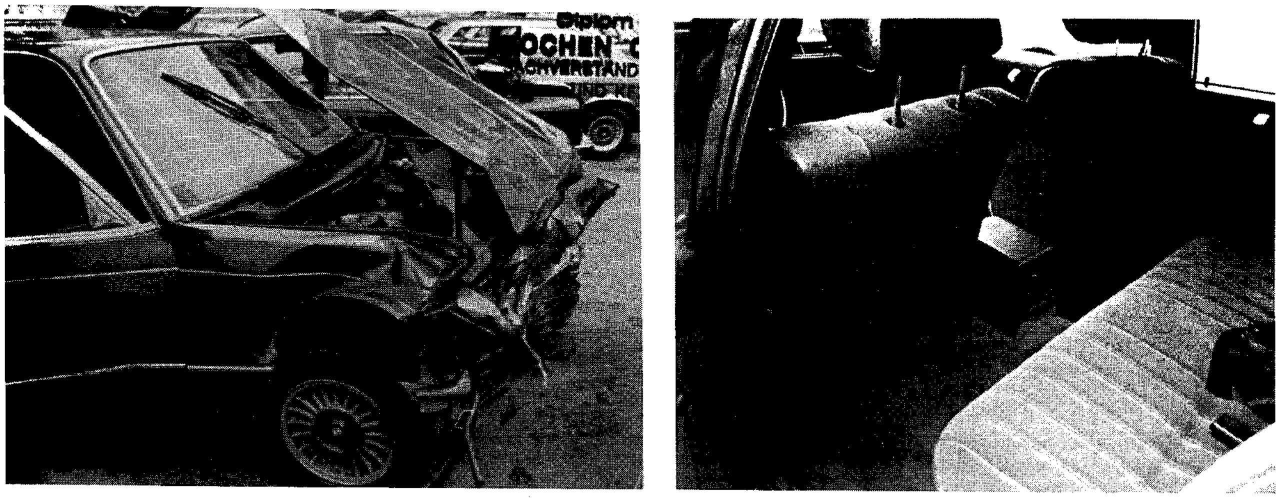

Figure 15. Serious head-on collision with relatively slight injuries to the belted passenger on the left-hand rear-seat

Up to now, the HUK-Verband has not carried out a comprehensive, large-scale analysis of accidents with *belted* rear-seat occupants, but it can be assumed from the finding we have made that the injury-reducing effect of a rear-seat belt amounts to at least 50%, probably even 70%.

The serious head-on collision shown in Figure 15 is to demonstrate the considerable protective effect of the safety belt on the rear-seat: the belted rear-seat passenger on the left only sustained a collar bone fracture and bruises. What is also of fundamental importance in this accident is the fact that the belted driver was exposed to no additional loading by the rear-seat passenger.

In the field of the passive safety of cars—with regard to both the front and the rear-seat occupants—a very high standard has now been achieved but further improvements are still possible.

Conclusions for Possible Safety Measures

Improvements in passive safety on the rear-seat

In the future, car manufacturers should devote the same attention to the requirements of an optimum belt and belt positioning on the rear-seat as on the front-seats. Although present-day rear-seat belts basically operate satisfactorily, the following improvements are still necessary and possible:

- As a matter of principle only three-point inertia reel belts should be used for the outside seats in the rear.
- The upper anchorage point of the rear-seat belt must be optimized. According to studies made by the Technischer Überwachungs-

50 km/h crash test

Source: Audi AG

Injuries to the belted driver:

- Commotio cerebri
- Bruises to chest
- Grazed right arm

Figure 16. Crash test and real-life accident of the same accident severity with only slight injuries to the belted driver

verein [10] the anchorage points at the back are in some cases not ideally positioned for reasons of strength; their height should also be adjustable.

- The lock of the belt should be attached to belts/belt mountings which are as short as possible, in order to avoid the belt lock lying across the abdominal region; in addition, the depth of the springs in the rear-seat should be reduced towards the front edge to avoid submarining.
- To improve the positioning of the rear-seat passengers the seating area should be contoured.
- The resistance of the backs of the front-seat to penetration must be increased, especially in the case of vehicles with fairly small interior dimensions.
- Generally, all areas for a possible head impact should be padded with an energy-absorbing material.

These improvements, of course, cannot be carried out at short notice, since some of them are connected with a structural optimization of the car, but because of the injury risks of the rear-seat passenger and in particular of the interaction between the occupants further protective measures are necessary.

Improvements in passive safety on the front-seats

Although the restraint systems on the front-seats are better than on the rear-seats, here, too, further improvements are possible. In spite of the considerable reduction in head injuries they continue to dominate in serious and fatal injuries—only, however, in accidents of very great accident intensity. Nevertheless, it still seems possible to considerably extend the present-day protection area by technical measures.

But to do this a knowledge of the injury mechanisms in real-life accidents is necessary. Because of the high belt-wearing rate in Germany today, we can now acquire this knowledge far more quickly than hitherto. Today it is an established fact that a 50 km/h wall impact (Figure 16) can most probably be withstood without serious injuries if a restraint system is used. But also in the case of considerably more serious accidents, corresponding, for example, to a wall impact speed of 60 to 70 km/h, it can be noticed that frequently no severe head injury occurs (Figure 17) and that this confirms earlier observations which can be described by the slogan: "No serious head impact—no serious head injury". That means that normally no severe head injuries can be observed as long as a hard impact on the steering wheel or other parts of the interior is avoided by the restraining effect of the belt. In the HUK-Verband's studies it has so far not been possible to find any references to "rotation acceleration without direct impact" as a cause of injury. On the contrary, the serious head or brain injuries were practically always linked with flesh injuries or also skull fractures in the region of the face—i.e. with clear indications of a direct head impact. If our view is confirmed, it will provide us with the opportunity to take technical action, namely, to prevent a head impact in the interior under all circumstances or to reduce the intensity of the impact.

In the past the greatest potential for improving passive safety in the car existed above all and most clearly in the greatest possible increase in the belt-wearing rate. Today, now that on average 95% of all car occupants on the front-seats in Germany wear seat belts (situation in September 1986), a search must be made for other possibilities, which, however, are to be mainly found in the technical sector. Development should concentrate on the following points:

- The design of the steering column and steering wheel is still in need of improvement in many vehicles. In accordance with regulation ECE-R 12, the upper end of the steering column must not penetrate more than 127 mm into the interior in a 48 km/h wall impact—but this applies only in a horizontal direction and parallel to the longitudinal axis of the vehicle.
- In serious head-on collisions, however, we repeatedly noticed that the driving column is pushed up inside the car, and thus the steering wheel or its rim penetrates directly

Injuries to the belted driver:

- Skull bruises
- Chest crushed with pneumothorax, right
- Multiple fracture of shinbone, right
- Fracture of ankle joint, left
- Inside ankle fracture, left
- Fracture of second metacarpal, right
- Multiple grazing and bruising

Figure 17. Extremely serious head-on collision with only slight head injury to the belted driver

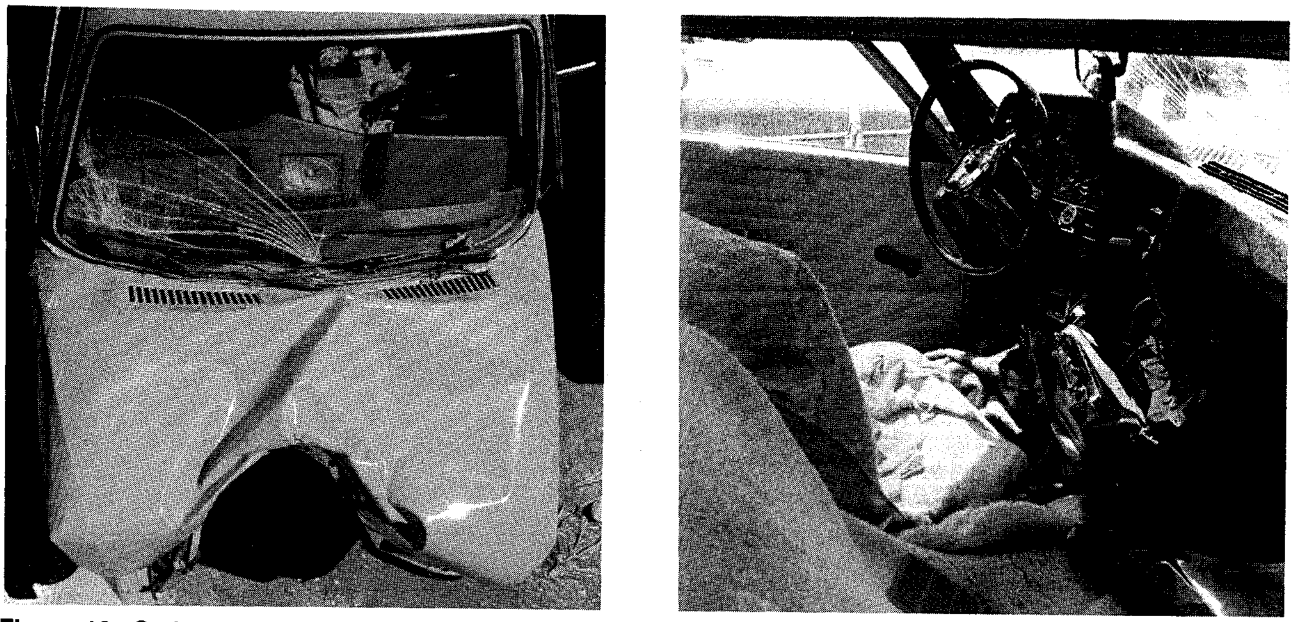

Figure 18. Serious head-on collision with steering column thrust up and into the car

into a belted driver's area of movement (Figure 18). This results in an unnecessarily high head injury risk for the driver. An urgent demand for a change in the safety regulations should be made. In many vehicles the steering wheel hub is still quite inadequately padded and is not designed to form an impact area.

- In spite of the good protective effect of the present-day restraint systems on the front-seats, a further improvement in three-point belts is possible: their user-friendliness must be further improved by making it possible to adjust the upper belt anchorage point on the B-column and by attaching the belt lock to the seat in *all* vehicles.
- That the belt reduces the consequences of injuries has been confirmed beyond doubt. A further improvement in the protection from injuries is possible by means of the belt pretensioner. On the one hand, it lowers the loading on the occupant and, on the other, reduces his forward movement by about 15%, which decreases the risk of an impact against the steering wheel. The problem of electronic systems—at least for the time being—lies in the costs of the components necessary to activate them; this problem must and can be overcome by new systems, as the development of a mechanical system by a German car manufacturer has shown.
- Although head injuries have been reduced by about 75% by wearing seat belts, the head impact of the driver against the steering wheel still constitutes a dominant cause of injury when injuries occur in spite of using the belt. Measures to the steering wheel by design optimization, padding etc. are still of greatest importance; one of the most crucial means of providing safety is the steering wheel airbag, although other systems—for example, the above-mentioned mechanical system of a German car manufacturer—can be imagined which result in a considerable reduction in the head injury risk.
- There are still problems in head-on collisions with high offset (see Figure 12), in which the deformation is concentrated in the outer area outer area of the passenger cell and in which there are considerable risks for the belted front-seat occupants as well, even in collisions of only moderate intensity. Because of the high level of intrusion, injury risks occur especially in the leg area [11]. Head-on impacts with trees/posts also often cause deep intrusion resulting in relatively high injury risks.
- In the automobile industry's crash tests, even more attention than hitherto should be paid to the fact that there must be a balance between self protection and partner protection. It would be desirable if compatibility criteria could be tested in a supplementary test (see also [12]).

Summary

Although there has been a great increase in the number of road traffic accidents altogether in Germany in the last few decades, the number of fatalities has clearly declined since 1970; in 1985 8,400 people were killed in traffic accidents in Germany.

Parallel to the total number of fatalities, the number of car occupants killed has also continuously decreased. This clear decline in the number of car occupants killed—and seriously injured, too—is mainly due to the sudden rise in the belt-wearing rate on the front-seats of cars in August 1984 from about 60% to over 90% (August 1984 to July 1985 compared with the previous year: −1,461 fatalities and −15,091 seriously injured passengers); the high belt-wearing rates for drivers and front-seat passengers even rose slightly up to September 1986 to as much as 95%. Because of these facts it has been possible to assume that a change has taken place in the injury consequences and the priorities in the car accident scene since August 1984. Although the high belt-wearing rate has resulted in raising the significance of side-collisions, head-on collisions still come first as far as the number of car occupants killed is concerned.

Although using the belt has reduced head injuries to the belted driver by about 75%, the head is the part of the body which is injured most frequently, followed by injuries to the chest and abdomen.

In the case of a belted front-seat passenger the chest is injured most frequently, followed in second place by injuries to the head and in the third place abdominal injuries.

In the past the significance of safety questions relating to rear-seat passengers has usually been underestimated; but in the HUK accident studies it was possible to obeserve a not inconsiderable injury risk for unbelted rear-seat occupants. Moreover, unbelted rear-seat occupants represent an increased risk for the other occupants in at least 13% of all car accidents.

Raising the belt-wearing rate even further can be regarded as a most important measure for increasing the safety of rear-seat occupants; in September 1986 this rate amounted to only 50% in Germany. Apart from this, however, technical measures are also necessary and possible, as they can also help to raise the safety of a rear-seat occupant. Some of these are: Generally fitting cars with three-point inertia reel belts, shorter belt locks (integrated into the rear-seat bench if possible), shaping the seating area and optimizing the rigidity of the seats to avoid submarining, optimizing the front-seats to avoid unnecessary injury risks, and providing more padding for possible head impact areas.

In order to further increase the safety on the front-seats of cars the following measures in particular should be considered: improving the design of the steering column and wheel, avoiding excessive intrusion in the case of accidents with high offset or collisions with a tree, making the upper belt anchorage point adjustable and attaching the belt lock to the seats in *all* vehicles, equipping cars with belt pretensioners, and the airbag or systems which are equally effective in reducing a possible head impact in very severe accidents.

Although introducing these technical measures will no longer result in such a great reduction of the numbers of injured people in cars as occurred in Germany as a result of the sudden increase in the belt-wearing rate, considerable successes will also be possible in the future, but in smaller steps; successes which will express themselves in a further decline in the numbers of car occupants killed and injured.

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Residual Injuries to Restrained Car Occupants in Front- and Rear-Seat Positions

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Abstract

The highly protective quality of the safety belt remains undisputed. Nevertheless, it is evident, that almost the same injuries may occur whether or not a safety belt is used. Cognitions can be derived from this study in which accident and collision situations injuries are inflicted to belt-protected drivers, co-drivers and rear-seat passengers. On the basis of documented traffic accidents, investigated by a scientific research team, a total of 1,865 single injuries incurred by 605 belt-protected front-seat and 35 rear-seat car passengers were analyzed. It became evident that in frontal collisions and protection by safety belt, a high injury risk still remained for car passengers, for instance for the knee region by the dashboard, for thorax and head region by the steering wheel. In lateral collisions, injuries to the whole body were caused by the side structures of the doors. This situation must be regarded as a challenge for modification by car constructors.

Introduction

Ever since it was introduced, there have been those in favour of the safety belt, pointing out the high protective quality for injury avoidance and mitigation. But there were also those who opposed belt usage, accentuating the negative effects of the belt or belt strap. Through the arguments many car occupants were in doubt if they would not be safer without belt in accidents like overturning or car fires. Especially in view of the fact that serious abdominal injuries could be attributed to the belt. Consequently, there was quite a difference of opinion about the acceptance of this system. In 1983, the rate of belt-wearing in Germany amounted to just 45 percent for drivers in town districts—on motorways, however, to 81 percent. This rate increased to over 90 percent for both road systems, only after legislation of a fine. It was presumed that rear-seat passengers faced relatively little risk to get injured, due to their seating position. Only as late as 1979, safety belts were installed in rear seats in Germany. Since 1986, non-observance of belt-usage laws is fined in the German Federal Republic. The rate of belt usage for rear-seat passengers amounts to approximately 39 percent inside towns, and to approximately 66 percent on federal motorways.

This development was due to a multitude of effectivity analyses. Especially in frontal collisions of a vehicle, the highly protective effect of the safety belt was clearly demonstrated. The passengers were restrained by the elastic material, and an impact with the interior structures was prevented by the belt. In lateral collisions, however, the protective effect is limited. The passenger on the impact side is mainly injured, due to intrusion of the compartment, the belt, however, reduces the risk of being flung out. For passengers sitting in the opposite direction, an increased protective effect is apparent.

From all accident analyses it was evident that basically injuries can not be completely eliminated by the belt, and injuries can still occur despite belt usage. Publications (2,3) often deal with the problem of intra-abdominal injuries caused by the belt. The downward-directed movement of the passenger led to the conception of the so-called 'submarining movement', a motion causing intra-abdominal injuries, attributed to the belt strap. However, detailed accident analyses proved this to be a wrong conclusion, as basically these injuries can be caused in accident situations with intense intrusion of the passenger compartment(4).

On the one hand, the great protective effect of the safety belt is recognized; on the other hand, a multitude of injuries may still occur with belt usage. It is the objective of this study to demonstrate and analyse the injuries which still occur, despite belt usage, in certain accident situations.

In order to indicate injuries and accident mechanisms, a detailed accident analysis is required, which can not be obtained from police reports or isolated medical reports. The basis for such special detailed accident analyses is provided by special documentations of the 'investigations at the scene of accident'.

Description of the Investigation Material

In the greater vicinity of Hannover (town and country district of Hannover), an interdisciplinary research team of doctors and engineers drive to the place of accident immediately after the accident event. The team is informed of the accident by the Rescue Headquarters in Hannover. It drives to the place of the accident in vehicles equipped with blue lights and sirens. Accident traces like vehicle deformations, impact points on people are investigated and recorded immediately. Photogrammetric true-to-scale drawings are produced by a stereo-camera. These drawings enable a reconstruction and analysis of the accident as well as the collision phases. The injuries are documented divided into types, localisation and severity degree (AIS), and the injury mechanisms assigned within the framework of a technical/medical accident reconstruction.

608 front car passengers with automatic belts, in 445 cars built in 1976 and later, were analyzed, after collisions with other cars and objects like trees, walls, leading planks etc. Table 1 illustrates the distribution of cars, according to collision type, year of construction, weight category and impact deceleration, established by reconstruction and indicated by △V. Frontal collisions of the accident vehicles predominate in the accident scene with 51.4 percent. Multiple collisions, i.e. when a vehicle collides in the accident incident with several sides, are very frequent with 21.6 percent.

Injury Situation of Front-seat Passengers

Injury severity. The highest injury severity degrees MAIS were established for frontal impacts with impact direction, almost in axial length of the vehicle ($\pm/-30°$) and for the rectangular impact ($\pm/-30°$) with impact in the compartment region, and also for the multiple colliding car occupants as shown in table 1: More than 50 percent of the occupants remained still unhurt in impacts outside the compartment region. There were no injured occupants with severity degrees 5 and 6 in these collision constellations.

Impact decelerations to passengers occurred with △ values up to 30 km/h. In this group, 65.6 percent of the persons remained uninjured at this speed level, while a distinct increase in the maximum injury severity degree MAIS can be observed with a higher △V. In vehicles of heavier weight categories, more passengers with lesser injury severity degrees were found than in vehicles of lighter weight.

Injury causes. In frontal collisions the moment of inertia causes the car occupants to continue in their original direction of movement, and their exposed body regions impact the vehicle parts in their way, after deceleration of the vehicle. The belt reduces a luxation of the body trunk. The relatively free moving body parts like the head, arms and legs, however, are

Table 1. Frequencies of impact directions in traffic accidents in relation to injury severity grades: Vehicle registration year, vehicle weight and △V.

	cars n	cars %	occupants n	occupants %	Maximum Injury Severity-grade: uninjured	MAIS 1/2	MAIS 3/4	MAIS 5/6
					<---- 100 %			---->
total	445	100.0	608	100.0	44.9	43.2	8.6	4.1
impact direction								
frontal: frontal + 30	130	29.2	174	28.6	39.1	43.7	12.6	4.6
frontal: oblique/ rectangular	97	21.8	130	21.4	64.6	32.3	2.3	0 8
lateral: inside compartment								
rectangular + 30	25	5.6	37	6.1	18.9	56.8	10.8	13.5
oblique	31	7.0	38	6.3	44.7	39.5	7.9	7.9
lateral: outside compartment								
rectangular + 30	9	2.0	13	2.1	76.9	23.1	-	-
oblique	12	2.7	17	2.8	58.8	35.3	5.9	-
rear collision	23	5.2	34	5.6	79.1	20.9	-	-
multiple collision	118	26.5	165	27.1	27.3	56.4	11.5	4.8
vehicle-registration-year								
1976 - 1979	245	55.1	334	54.9	46.4	41.6	9.6	2.4
1980 - 1983	139	31.2	190	31.3	45.2	40.0	7.4	7.4
1984 -	22	4.9	30	4.9	50.0	50.0	-	-
unknown	39	8.8	54	8.9	22.2	61.1	11.1	5.6
vehicle weight								
- 900 kp	73	16.4	84	13.8	36.9	44.1	11.9	7.1
900 kp - 1200 kp	218	49.0	299	49.2	41.1	46.2	8.0	4.7
1200 kp -	139	31.2	205	33.7	52.7	36.1	8.8	2.4
unknown	15	3.4	20	3.3	30.0	70.0	-	-
delta-v								
- 30 km/h	210	47.2	288	47.4	65.6	32.3	1.8	0.3
31 km/h - 50 km/h	112	25.2	154	25.3	29.9	54.6	12.3	3.2
51 km/h - 70 km/h	25	5.6	33	5.4	9.1	39.4	27.3	24.2
71 km/h - 90 km/h	3	0.7	3	0.5	-	-	-	100.0
91 km/h -	4	0.9	4	0.7	-	-	50.0	50.0
unknown	91	20.4	126	20.7	23.8	57.9	13.5	4.8

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still mobile. A possible luxation of the body trunk is influenced by the elastic quality of the belt, the response of the belt strap and its slackness in at-ease position, and further by the thickness of the clothes and the body build, i.e. whether not-so-slim or obese. Depending on the seating position and body size, an impact of the extended body parts is likely.

It was established that, even with automatic safety belts, 30.4 percent of the drivers suffered a head impact, as shown in table 2. This figure was distinctly lower for co-drivers with 18.4 percent. Abdominal injuries occurred with 8.3 percent almost twice as often for drivers as for co-drivers with 4.6 percent.

Table 2. Kinds of impact by driver and co-driver in relation to injured body regions (100% = all persons each kind of impact separately for drivers and co-drivers).

injured body region driver/ co-driver	kind of impact: frontal %	lateral, occupant on impact side %	lateral, occupant on opponent side %	multiple collision %
driver				
total (n)	217	38	39	109
head	30.4	47.4	35.9	43.1
neck	8.8	21.1	2.6	12.8
thorax	28.6	31.6	25.6	39.4
arm	18.9	28.9	17.9	30.3
abdomen	8.3	15.8	7.7	14.7
pelvis	6.0	15.8	5.1	11.9
leg	28.1	42.1	23.1	27.5
co-driver				
total (n)	87	17	11	56
head	18.4	23.5	27.3	41.1
neck	9.2	-	-	16.1
thorax	33.3	11.8	27.3	46.4
arm	19.5	23.5	36.4	35.7
abdomen	4.6	5.9	-	12.5
pelvis	5.7	11.8	9.1	16.1
leg	24.1	29.4	9.1	44.6

6/87-2

There were hardly any differences between injuries for drivers and co-drivers to all other body parts. Beside the head which is for the driver the most frequently injured body part, the thorax (28.6% for drivers) and the lower extremities (28.1%) must be mentioned as especially endangered body parts, as can be seen in table 2. 23 percent of the drivers incurred soft-part injuries of the face, 10.6 percent suffered a skull-brain trauma. The proportion of facial fractures is with 4.6 percent for co-drivers relatively high, in view of the fact that on the whole they quite rarely suffer head impacts (see table 3). The head impacts here often, i.e. with 12.4%, the steering wheel, to 6% the upper A-posts and the windscreen region to 8.8%, as shown in table 4 for drivers.

Throat injuries especially occur after an impact with the front structures, within an indirect trauma. This happens to approximately 8.8% of the drivers and 9.2% of co-drivers. They are as a rule distortions, so-called whiplash injuries (4.6%). Fractures of the cervical vertebra were not recorded with belt-protected co-drivers, after frontal collisions.

The thorax is another body region exposed to great injury risk. Approximately one third of all drivers and co-drivers incur such injuries (table 2), especially soft-part lesions. These injuries can primarily be attributed to the belt, as 19.4% of the injured in frontal collisions incurred thorax injuries by the belt.

Table 3. Localization and kind of injuries for drivers and co-drivers in frontal collisions and for front seat occupants in lateral and multiple collisions (100% = all persons each column).

localisation and kind of injuries	kind of impact: driver frontal %	co-driver frontal %	lateral, occupant on impact side %	lateral, occupant on opposite side %	multiple collision %
total (n)	217	87	55	50	165
skull soft part	7.4	4.6	21.8	20.0	24.8
face soft part	23.0	10.3	23.6	22.0	23.0
skull fracture	4.6	1.1	3.6	4.0	3.0
facial fracture	5.1	4.6	1.8	-	3.6
cerebral injury	10.6	8.0	10.9	10.0	17.6
eyes	6.5	4.6	-	2.0	8.5
neck soft part	3.7	4.6	9.1	2.0	7.3
cervical spine					
-distorsion	4.6	4.6	5.5	-	4.2
-fracture	0.5	-	1.8	-	2.4
thorax soft part	25.3	27.6	16.4	22.0	33.3
rib fracture	6.0	3.4	10.9	4.0	9.7
sternum fracture	1.8	3.4	-	2.0	1.8
lung	2.8	-	3.6	2.0	4.2
diaphragm	0.9	1.1	-	2.0	0.6
heart, vessels	1.8	-	3.6	-	1.2
thoracic spine	0.5	-	-	-	-
clavicle fracture	1.8	1.1	-	2.0	3.6
scapula fracture	-	-	-	-	0.6
abdomen soft part	5.1	3.4	7.3	4.0	9.7
vessels	0.9	-	3.6	-	0.6
mesentery	0.9	-	-	-	1.2
liver, gall	0.9	-	-	2.0	1.8
spleen	-	-	5.5	-	2.4
kidney	0.5	-	-	2.0	1.2
stomach, intestine	0.5	-	-	-	1.2
lumbar spine	1.8	1.1	-	-	1.8
pelvis soft part	4.6	4.6	7.3	4.0	9.7
pelvis fracture	1.8	1.1	7.3	4.0	4.2
symphysis pubis	-	-	1.8	-	-
vessels	-	-	1.8	-	0.6
organs	0.5	-	1.8	-	31.5
legs soft part	26.3	21.8	36.4	20.0	31.5
thigh fracture	8.3	4.6	9.1	2.0	2.4
knee fracture/lux.	2.3	1.1	-	-	1.2
lower leg fracture	1.8	2.3	5.5	2.0	1.8
foot fracture/lux.	2.3	5.7	-	-	1.8
arms soft part	13.8	18.4	25.5	20.0	29.7
upper arm fracture	1.8	2.3	-	-	2.4
elbow fracture/lux.	0.5	-	-	-	-
forearm fracture	5.1	1.1	-	-	-
hand fracture/lux.	2.3	1.1	1.8	-	1.2

6/87-3

9.7% of the thorax injuries were caused by the steering wheel. With co-drivers, no injuries were found to the lung, the heart and the dorsal vertebra. They did, however, more frequently incur sternum fractures (3.4% of the co-drivers and 1.8% of drivers).

Injuries to the abdominal organs for drivers can with less than 1% be considered as quite rare. For co-drivers they were not registered at all in this collision situation. In frontal collisions, injuries to the spleen were not registered for either drivers or co-drivers. All injuries to the abdomen were exclusively soft-part injuries. They were found to 3.4% for co-drivers, and to 5.1% for drivers. For belt users in frontal collisions, bony pelvis injuries are almost exclusively indirect traumas which would lead to hip luxations and luxation fractures, due to a knee impact and power transversion, via the thigh and hip region. This applies to 1% of drivers and 1.1% of co-drivers. Thigh fractures were established for 8.3% and knee fractures or knee-joint luxations for 2.3% of the drivers; fractures of the ankle-joint and foot region

Table 4. Injured body regions and impacted vehicle parts of drivers by kinds of impact (100% = all persons each column).

injured body region and impacted vehicle parts	kind of impact: frontal %	lateral, occupant on impact side %	lateral, occupant on opposite side %	multiple collision %
total (n)	217	38	39	109
head				
windscreen region	8.8	5.3	10.3	13.8
dashboard	2.8	-	-	-
steering wheel	12.4	2.6	5.1	8.3
upper A-pillar	6.0	5.3	5.1	5.5
door	1.8	26.3	10.3	6.4
upper B-pillar	-	7.9	2.6	4.6
rear compartment	-	-	5.1	0.9
roof	-	-	2.6	3.7
seat belt	-	-	-	0.9
neck				
dashboard	0.9	-	-	-
steering wheel	-	-	-	0.9
upper A-pillar	0.5	-	-	0.9
door	-	5.3	-	0.9
B-pillar	-	-	-	0.9
seat belt	2.3	5.3	2.6	5.5
indirect	4.6	10.5	-	3.7
thorax				
windscreen region	0.9	-	-	0.9
dashboard	1.8	-	-	-
steering wheel	9.7	-	2.6	7.3
door	0.5	18.4	5.1	11.0
B-pillar	-	2.6	-	1.8
roof	-	-	-	0.9
seat belt	19.4	10.5	20.5	20.2
upper extremities				
windscreen region	5.1	2.6	5.1	7.3
dashboard	4.1	5.3	2.6	10.1
steering wheel	5.5	7.9	7.7	7.3
upper A-pillar	1.4	-	-	2.8
door	5.1	13.2	7.7	6.4
B-pillar	-	-	-	1.8
seat belt	0.5	-	-	-
abdomen				
steering wheel	3.2	-	2.6	4.6
door	-	10.5	2.6	0.9
seat belt	3.7	2.6	2.6	8.3
pelvis				
dashboard	-	-	-	1.8
door	-	10.5	2.6	4.6
B-pillar	-	-	2.6	-
seat belt	4.6	2.6	2.6	5.5
lower extremities				
dashboard	21.2	13.2	15.4	18.3
steering wheelent	0.9	-	-	4.6
front leg room	12.9	21.1	10.3	13.8
door	0.9	13.2	2.6	4.6
seat belt	1.4	2.6	-	0.9

6/87-4

for co-drivers are frequent with 5%. Approximately 25% of all front passengers incur soft-part injuries to the legs. The knee is especially involved, due to an impact to the dashboard, as is shown in table 4.

In lateral collisions, passengers experience an additional relative movement component in lateral direction, beside a frontal-directed deceleration. In this case, the passenger on the impact side experiences an extreme encumbrance by the deformed vehicle side structures. For the passenger sitting impact directed, almost all body parts can be considered as injury regions. For the passenger sitting impact averted, only injuries to the exposed body parts like head, thorax, lower and upper extremities are frequent, as can also be seen in table 2. Impact averted as well as impact directed sitting passengers incurred to 42% and 45.4% respectively soft-part injuries to the head (table 3). These included, up to approximately 20% injuries to the vault of the cranium. Passengers sitting impact directed very often receive soft-part injuries to the throat region (9.1%). Distortions and fractures of the cervical vertebra are established exclusively for this passenger group. Rib fractures are also extremely frequent (10%) for those sitting impact directed. They also incurred in 5.5% ruptures of the spleen. These ruptures were exclusively established in this group with belt usage. To 7.3% bony injuries in the pelvis region were also found in this group; this is considered as bursting of symphysis and ileosacral with these persons. Another frequent injury cause for impact directed sitting passengers are door structures (table 4). They were responsible to 26.3% for head injuries, to 18.4% for thorax injuries, to 10.5% for injuries to abdomen and pelvis, and to 13.2% for injuries to the lower extremities. The B post is also a frequent cause for injuries. It was responsible for head injuries to 7.9% of car occupants, who were sitting impact directed. However, not all injuries in lateral collisions are inflicted by parts of the compartment side. In oblique impacts and subsequent oblique relative movement of the occupant, injuries are also caused by front parts, for instance by the dashboard, especially to those sitting impact directed. In oblique impacts, a motion of the upper body trunk and the head region may occur, as far as the front interior parts like A posts, dashboard and steering wheel. The occupant sitting opposite directed, more often incurs minor injuries (90.6% severity degree AIS 1 and 2). A higher injury frequency by the belt can be registered for this person, due to a more frequent restraint of the body (20.5% impact averted, 10.5% impact directed).

Multiple collisions, in which the vehicle is submitted to several successive collisions during the whole accident phase, are of special significance, as far as the injury severity is concerned. It appears that in this case injuries may be caused by almost all parts of the

Table 5. Proportion of all injuries with AIS 1/2 for front-seat passengers in frontal and lateral collisions in relation to injured body regions and injury causing parts (100% = all injuries each causing parts and body region).

injured body region and injury causing parts	portion of all injuries with AIS 1/2			
	driver frontal %	co-driver frontal %	lateral, occupant on impact side %	lateral, occupant on opposite side %
head				
windscreen region	88.1	96.0	100.0	100.0
dashboard	100.0	87.5	-	-
steering wheel	98.0	-	100.0	100.0
upper A-pillar	83.9	-	85.7	100.0
door	100.0	-	92.0	90.0
upper B-pillar	-	-	42.9	100.0
rear compartment	-	-	-	66.7
roof	-	-	-	100.0
neck				
dashboard	100.0	-	-	-
upper A-pillar	100.0	-	-	-
door	-	-	50.0	-
seat belt	100.0	100.0	100.0	100.0
indirect	100.0	100.0	80.0	-
thorax				
windscreen region	100.0	-	-	-
dashboard	100.0	77.8	-	-
steering wheel	56.5	-	-	100.0
door	100.0	100.0	42.9	-
upper B-pillar	-	-	100.0	-
seat belt	100.0	85.7	100.0	92.3
upper extremities				
windscreen region	100.0	100.0	100.0	100.0
dashboard	100.0	93.3	100.0	100.0
steering wheel	68.2	-	100.0	100.0
upper A-pillar	57.1	-	-	-
door	88.9	100.0	100.0	100.0
seat belt	100.0	-	-	-
abdomen				
steering wheel	44.4	-	-	-
door	-	-	33.3	-
seat belt	100.0	100.0	100.0	100.0
pelvis				
front compartment	60.0	-	-	-
door	-	-	62.5	75.0
B-pillar	-	-	-	100.0
seat belt	100.0	100.0	100.0	100.0
lower extremities				
dashboard	94.2	83.8	85.7	92.3
front leg room	88.9	94.4	85.0	100.0
door	66.7	100.0	62.5	100.0
seat belt	100.0	-	100.0	-

6/87-5

interior, due to the actual impact situation and the consequent relative motion of the occupants.

For this collision type it also became evident that the steering wheel, the dashboard and the lateral door structures, inclusive A and B posts, are the most frequent cause for injuries. Here 47.8% of the occupants involved in multiple collisions incured soft-part injuries to the head (table 3), and 17.6% a skull-brain trauma. Eye injuries are also exceptionally frequent, with 8.5%. Fractures of the cervical vertebra result from a shoving-forward process (6). For this reason, such injuries occur more frequently (to 2.4%) in multiple collisions. In multiple collisions, all types of injuries to almost all body regions are exceptionally frequent.

Injury severity. As only 56.5% of thorax injuries caused by the steering wheel are of severity degree AIS 1 and 2, as shown in table 5, and 68.2% arm and 44.4% abdominal injuries, the steering wheel proved to be an especially injury-causing part for drivers in frontal collisions. Head injuries caused by the steering wheel are to 98% minor ones (AIS 1/2).

Serious injuries are quite rare for belt-wearing co-drivers in frontal collisions. The most frequent injury cause for them is the dashboard. They receive mostly injuries to the head (12.5% AIS >2) and the lower extremities (16.2% AIS >2). The high proportion of serious injuries inflicted by the dashboard (22.2% AIS > 2) must be attributed to an extensive intrusion of the passenger compartment.

The impact-directed sitting occupant often incurs serious injuries, especially to the head, in lateral impacts, for instance by the lateral posts (57.1%—AIS > 2, the A-post 14.3%—AIS > 2). The thorax, with 57.1%, abdomen with 66.7%, pelvis with 37.5% are seriously injured by the door structures.

Injury Situation of Rear-Seat Passengers

With a total of n=35 persons, only a small number of belt-protected rear-seat passengers were at our disposal for a detailed analysis. Due to this fact, a comprehensive illustration of the injury mechanisms was not possible. Instead, we prepared an abbreviated form about the injury situation of rear-seat passengers. 18 persons out of 35 incurred slighter injuries of severity degree 1 and 2, only one person incurred injuries of severity degree MAIS 4. The remaining 16 persons (45.5%) were uninjured. Approximately 50% of the rear-seat passengers were involved in frontal collisions, only three were seated in the middle and were protected by static belt. 20 persons were protected by automatic belt, three wore a static belt, and 12 used childrens' restraint systems. The primarily injured body regions were the head and lower and upper extremities. Exclusively soft-part injuries were established for thorax and pelvis which were attributed to the belt. It became evident that the relative movement of the belt-wearing rear-seat passenger procures an impact of the rather free moving extremities. In this process, the head impacts mainly head-rest and back-rest of the front-seat. The legs, and especially the tibia and knee hit the frame of the back-rest, the arms hit the lateral interior structures. As far as childrens' restraint systems are concerned, only soft-part injuries of the head and upper extremities were established for the 12 investigated cases.

Influence of Accident Severity

The definition of the vehicle deformation is an example for the accident severity. The stress caused to the occupants is mainly due to deceleration during the collision. This must be seen in connection with the inflicted injuries. The occurring accident stress is expressed by ΔV. For the above study, the injuries to passengers were analyzed, divided into body regions and injury cause, in dependance of ΔV values (figures 1 to 4). In frontal collisions, soft-part head injuries to belt-wearing front passengers occurred only above ΔV values of 20 km/h. With lower ΔV values, head injuries were only caused by the steering wheel. Especially in the driver's position, the distance to the steering wheel, which is often too short, could be the injury cause. 3.6 percent of involved drivers suffered soft-part head injuries by impacts to the windscreen region. Fractures were observed only with ΔV above 30 km/h. While head injuries by the dashboard only occurred with ΔV above 30 km/h, soft-part injuries caused by the upper A-post were already observed with ΔV above 10 km/h. It is evident that with a higher ΔV the injury frequency increases.

A soft-part injury to the thorax and pelvis region caused by the belt, the so-called belt mark, can be observed with ΔV values exceeding 10 km/h.

In conclusion, a significant risk for soft-part injuries with ΔV above 20 km/h is apparent, with the exception of injuries to the lower extremities and the belt-strap hematoma, which may already occur at 10 km/h. The injury risk for fractures is apparent above 30 km/h. The dashboard is to be regarded as especially dangerous, in view of the very great increase in frequency for leg injuries, with increasing ΔV.

In lateral collisions, soft-part injuries occur as a rule already with lesser ΔV values. Especially soft-part injuries to the relatively free moving head occur below 10 km/h, to the occupant sitting in impact direction. In contrast, soft-part injuries to the thorax by seat belt and side doors were observed above 10 km/h. The legs incurred injuries by the dashboard and the front foot room at speeds above 20 km/h. Door structures represent a great risk for fractures to the impact-directed occupant. Here 11.1 percent of the occupants in collisions of ΔV up to 10 km/h suffered pelvis fractures.

A low injury frequency for the whole ΔV region is evident for the impact-averted sitting passengers. For these persons fractures were observed only from ΔV of 40 km/h. This is due to the highly protective effect of the safety belt, which controls the relative movement of the impact-averted sitting occupant. This explains the fact that soft-part injuries by the belt may occur already with low ΔV values. Rib fractures may even occur in ΔV of 20 km/h.

Discussion

This study of real accidents proves that almost all injuries may occur despite belt usage, even if they are distinctly reduced in number by the highly protective function of the belt, and occur mainly in more serious accidents. Injuries which could be attributed to the belt are as a rule defined as hematoma and are caused by pressure of the belt strap to the soft parts. With belt usage, intra-abdominal injuries occur only due to incorrect belt fitting in the pelvis region, with soft and yielding seat bolsters and intense deformation of the department, superimposing the forward movement. A submarining movement causing intra-abdominal inju-

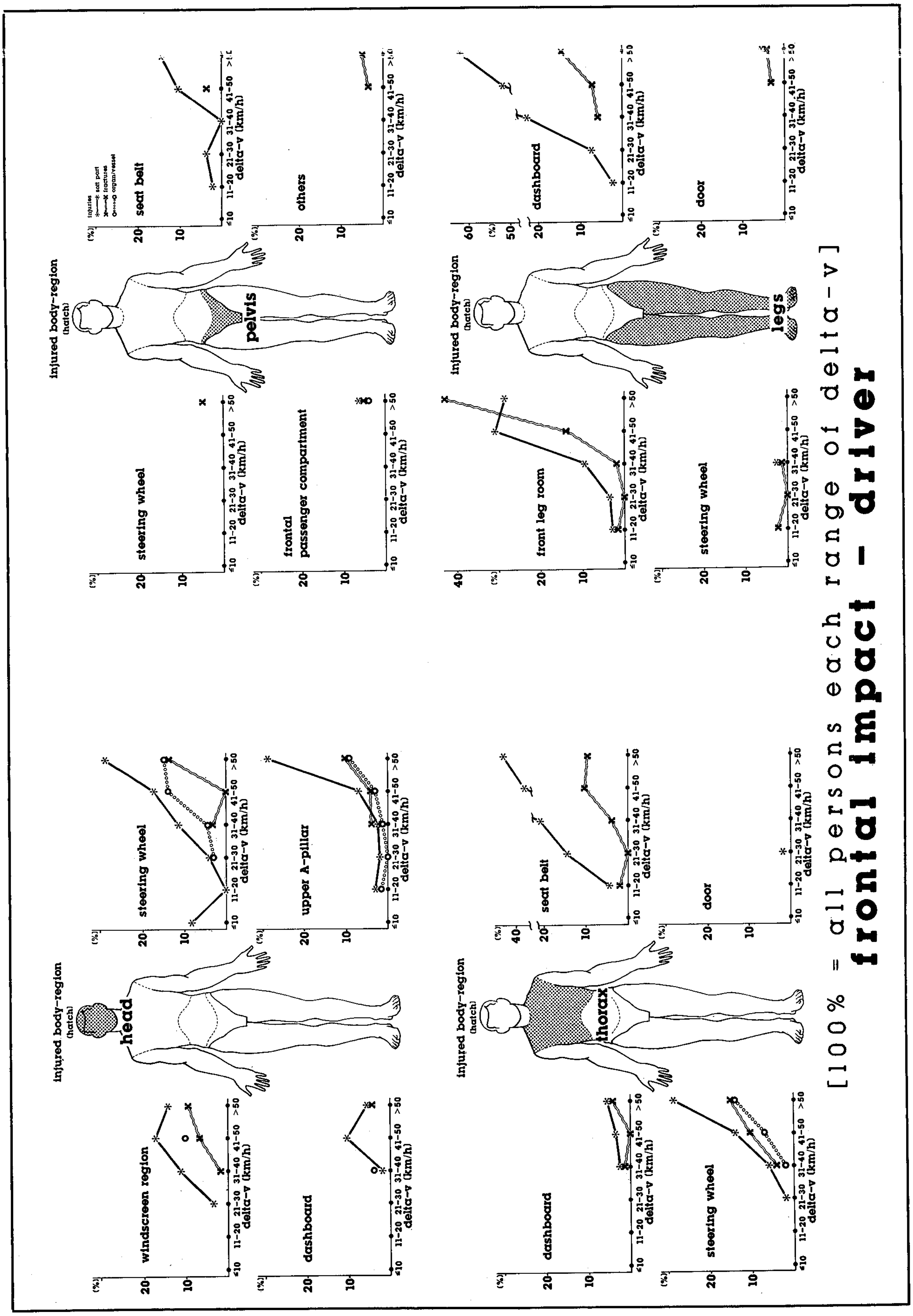

Figure 1. Relevant injury causing parts of selected body regions in relation to ΔV

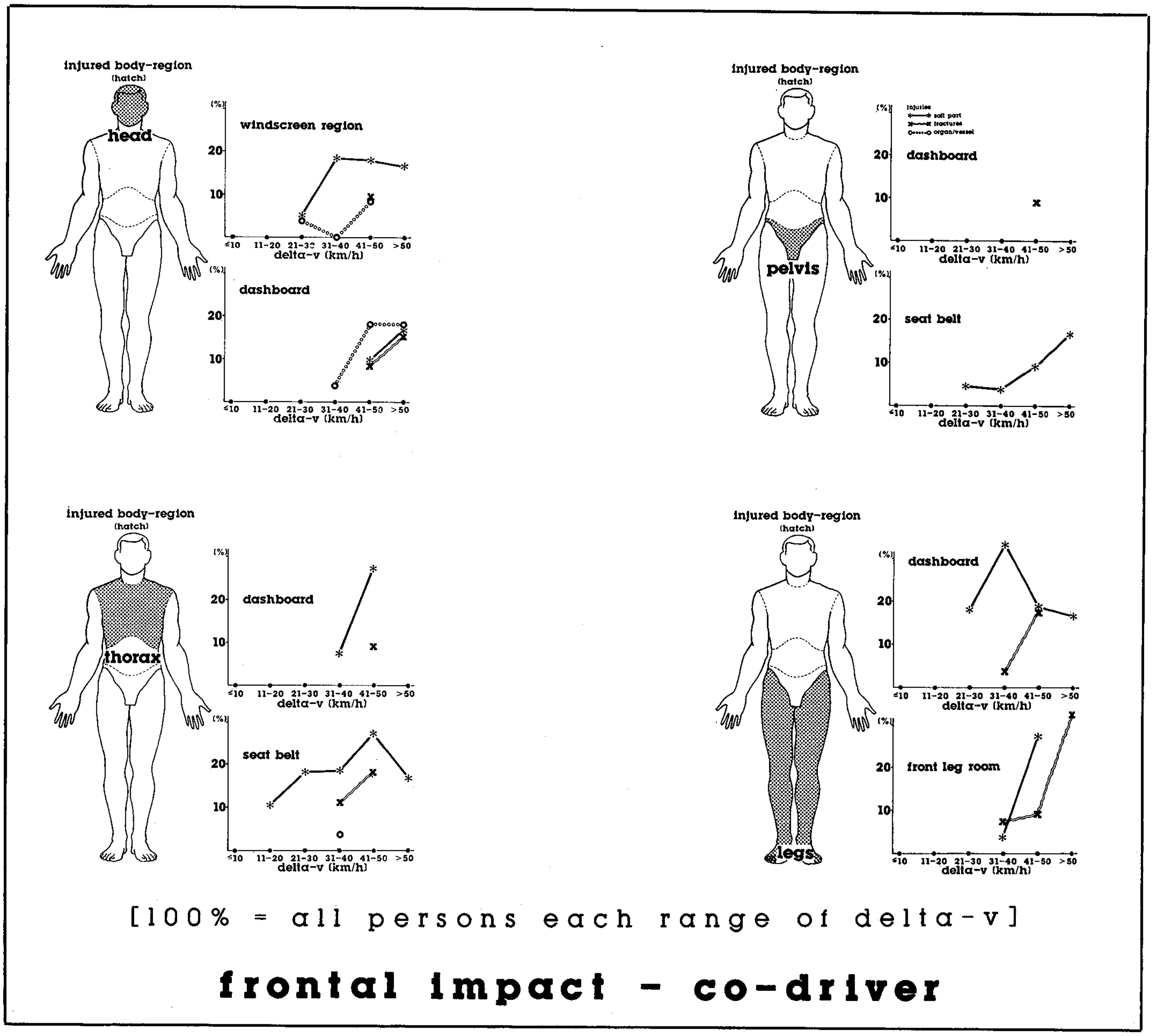

Figure 2. Relevant injury causing parts of selected body regions in relation to △V

ries could be established in a detailed analysis of 1.6 percent of all belt-using front passengers. It appears to be an essential cognition that intra-abdominal injuries must not necessarily be caused by a submarining movement, but that in fact other accident conditions like high intrusion or insufficient distance from the steering wheel could be responsible for the injury patterns. In cases of submarining, established beyond doubt, belt contusion marks were found in the abdominal and thorax region with characteristic patterns. These are

- marks of the belt strap on or just above both iliac crestlateral, while the belt impact mark in the front medial region of the stomach was clearly higher situated;
- mark on the thorax appears to have a distinct lower border, while the upper edge continuous blurred. This results in a broader belt mark than the original width of the belt.

Apart from these facts, the study demonstrates that, even with belt usage, a great injury risk still exists in frontal collisions, for the knee region by the dashboard, for the thorax and head region by the steering wheel, and in lateral impacts for the whole body by

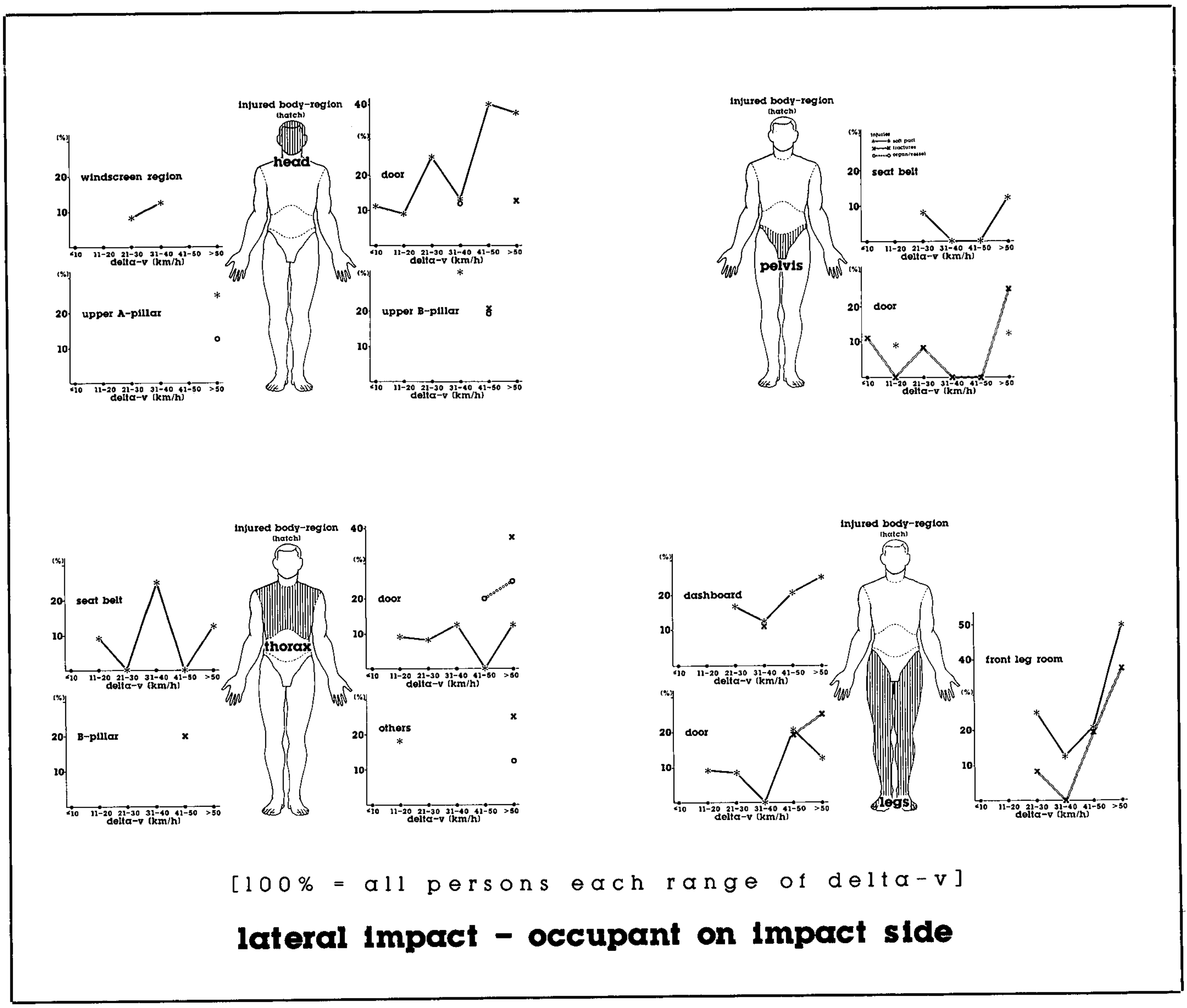

Figure 3. Relevant injury causing parts of selected body regions in relation to $\triangle$V

the side structures of the doors. Demands on the car constructors are, therefore,

- to extend the moving space for occupants,
- to bolster the front, but especially the lateral interior on impact exposed points.

In this connection, the efforts of some car manufacturers must be acknowledged, who are considering to reduce the injury risk by the steering wheel through modifications like airbag and Pro-Con Ten+. Also for the rear-seat region, the demand for the widest possible moving space and bolstering of the back rest and lateral vehicle structures has to remain.

Finally, the protective quality of the safety belt which was proved to be of benefit to front as well as rear-seat passengers, should be accentuated. This applies in lateral collisions especially for passengers sitting impact averted. In lateral and multiple collisions, it prevents the dangerous flinging out. It is a special attribution of the safety belt that the number of persons killed annually in road traffic, as well as the injury severity degree and also the total number of single injuries to an accident casualty has been distinctly reduced.

+Programmed contraction and tension by AUDI

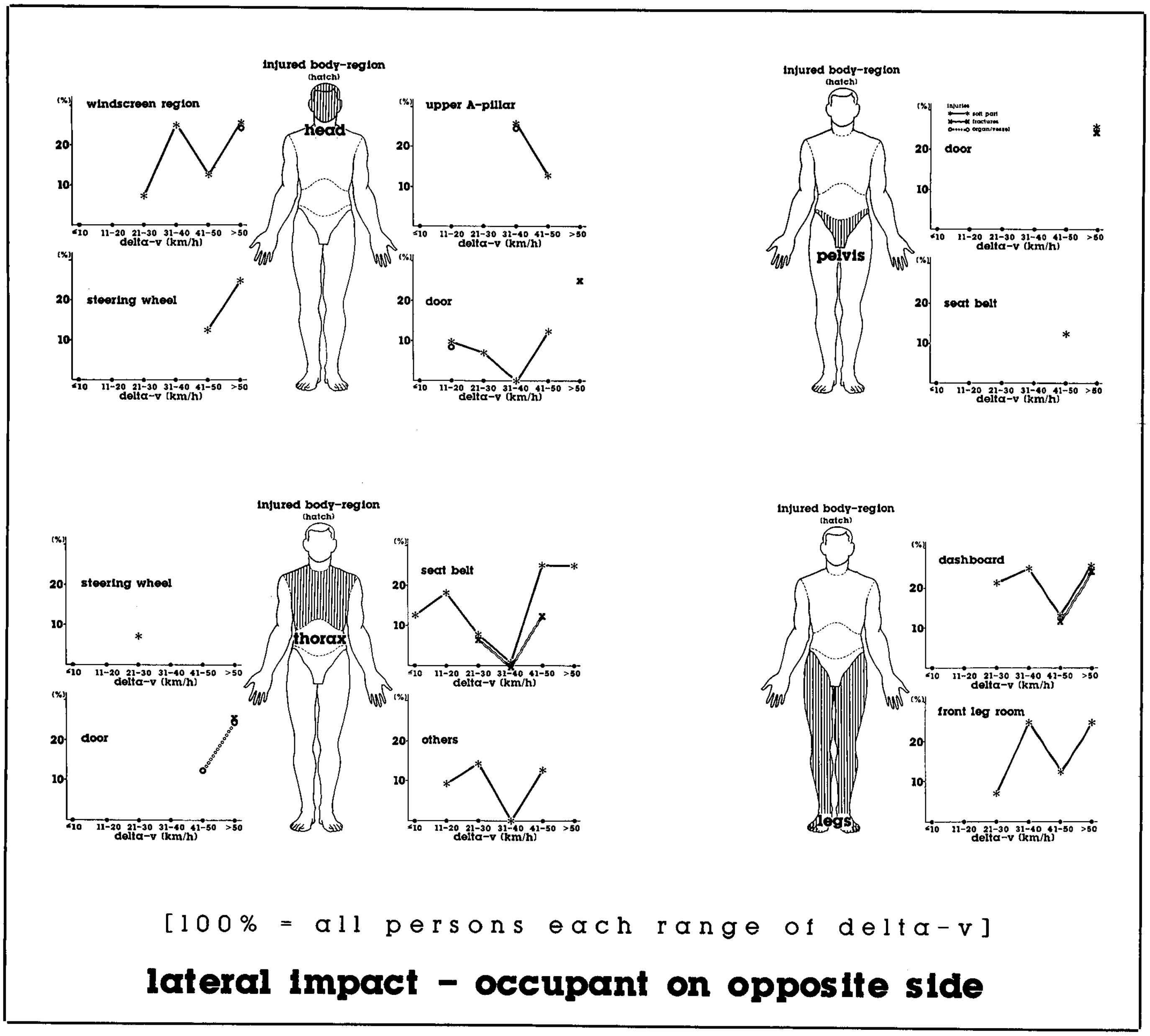

Figure 4. Relevant injury causing parts of selected body regions in relation to △V

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Occupant Protection Device Effectiveness in Preventing Fatalities

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Abstract

This paper summarizes the findings of a number of studies performed by General Motors Research Laboratories in which the double pair comparison method was used to determine the effectiveness of protection devices in preventing occupant fatalities. All the estimates use data from the Fatal Accident Reporting System, a data file maintained by the National Highway Traffic Safety Administration. This data file gives information on all fatal traffic crashes in the United States which occurred since January 1, 1975. It is found that lap/shoulder belts are $(43 \pm 3)\%$ effective in preventing fatalities to car front seat occupants, lap belts are $(18 \pm 9)\%$ effective in preventing fatalities to car rear seat occupants, and motorcycle helmets are $(27 \pm 9)\%$ effective in preventing rider fatalities. The errors are one standard error, and effectiveness means the fraction of a present population of fatally injured occupants not using the occupant protection device who would not have been killed had they been using the device, all other factors being equal.

Introduction

This paper presents an overview of recent estimates of effectiveness of three occupant protection devices in preventing fatalities in traffic crashes. The three occupant protection devices are lap/shoulder belts installed in the front outboard seats of passenger cars model year 1974 or later, lap belts in the rear outboard seats of passenger cars, and motorcycle helmets. By effectiveness we mean the reduction, expressed as a percent, in fatalities to a population not presently using the protection device which would result if all members were to become users, but not otherwise change their behavior.

All the estimates were obtained by applying the double pair comparison method [1] to data in the Fatal Accident Reporting System (FARS) [2]. Complete details, including all the raw data used in the determinations, are given in the original papers [3-5]. The present overview concentrates on the results.

Method

Data and the problem of exposure

Much information is available on essentially all fatal traffic crashes in the United States since 1975 in the Fatal Accident Reporting System (FARS) [2]. This computerized data file, which is maintained by the National Highway Traffic Safety Administration (NHTSA), which is part of the U.S. Department of Transportation, contains detailed records on over half a million people fatally injured in traffic crashes. From these data one can determine, for example, the number of belted and unbelted drivers who were killed in various types of crashes. Such information, however, provides no indication of safety belt effectiveness in the absence of some measure of the numbers of belted and unbelted drivers exposed to the risk of a fatality. Determining such exposure measures is one of the most difficult and pervasive problems in traffic safety research.

One possible approach is to obtain exposure data from independent sources, as is done in, for example [6,7]. However, difficulties usually arise because the exposure and accident data are generally collected for different purposes, and define variables in different ways (see, for example, [8]). The most fruitful approach to making inferences from FARS data is probably to develop methods that allow inferences using *only* information internal to the FARS files. One such approach which has been applied successfully [9,10] is the pedestrian exposure method described in [9]. In this, the number of pedestrians killed in crashes involving cars in some category (say, in the same mass range) is taken as a measure of the exposure of cars in that category to fatal crashes in general. This exposure method was used to estimate vehicle effects (for example, fatality risk in large compared to small cars); the double pair comparison method described below is used to estimate occupant effects (for example, fatality risk to belted compared to unbelted occupants).

The double pair comparison method

The method focuses on vehicles containing two occupants, referred to as a "subject" occupant and a "control" occupant, at least one of whom is killed. We here use the term "control" occupant, as used in [11], in preference to the term "other" occupant which I have used in earlier papers [1,3-5,12,13]. The method is used to determine how the likelihood that the subject occupant is killed depends on whether he has one of two characteristics, which, in this paper will always be whether he is using or not using some occupant protection device. The control occupant is used to estimate exposure. In the description below we take the driver as the subject occupant and the right front passenger (referred to as the passenger) as the control occupant. Two sets of crashes are considered.

The first set involves cars each containing a belted driver and an unbelted passenger, at least one of whom is killed. From the FARS data the numbers of belted drivers (=A, say) and unbelted passengers

(=B) killed in this set of crashes is determined. These data enable us to calculate a belted driver to unbelted passenger fatality ratio defined by

$$r_1 = \frac{A}{B} = \frac{\text{Number of belted drivers killed}}{\text{Number of unbelted passengers killed}} \quad (1)$$

If safety belts were 100% effective in preventing fatalities, then A (and r_1) would be zero. If they were near 100% effective, then r_1 would be small. If safety belts had no effect on fatality risk, then (assuming various other factors to be the same for driver and passenger) r_1 would reflect merely the relative safety of the driver and passenger seating positions.

The second set of crashes involves cars containing unbelted drivers and unbelted passengers, at least one of whom is killed. From these crashes we obtain the number of unbelted drivers killed (=C) and number of unbelted passengers killed (=D), which enables us to calculate an unbelted driver fatality to unbelted passenger fatality ratio defined by

$$r_2 = \frac{C}{D} = \frac{\text{Number of unbelted drivers killed}}{\text{Number of unbelted passengers killed}} \quad (2)$$

This ratio reflects only differences in the safety of the two seating positions (assuming various other factors to be the same for driver and passenger).

The ratio of Eqs 1 and 2, namely,

$$R = \frac{r_1}{r_2} \quad (3)$$

estimates the influence of the safety belt in preventing fatalities, normalized for any effects due to seating position. It is convenient to consider effectiveness, E, in reducing fatalities rather than the fatality ratio R, where E is defined by

$$\text{Safety belt effectiveness, } E(\%) = 100\ (1\text{-}R), \quad (4)$$

or, defined in terms of the original four fatality frequencies,

Safety belt effectiveness,

$$E(\%) = 100 \times \left(1 - \frac{A}{B} \times \frac{D}{C}\right) \quad (5)$$

The above discussion leading to Eq. 5 focussed on plausibility rather than rigor. In [1] it is shown with mathematical rigor that, subject to a number of assumptions which are discussed in detail, safety belt effectiveness is indeed determined by this simple expression involving four fatality counts readily extracted from FARS data. In the original papers [1,3-5] the quantities A, B, C and D are represented by d, e, m and n, respectively. The simpler terminology used here, and in [12], excludes some details in the original papers. Eq. 5 is the basis for all the estimates in this report.

The control occupant must be disaggregated by belt use because belt use by different occupants of the same vehicle is strongly correlated, so that an average passenger travelling with a belted driver is more likely than an average passenger travelling with an unbelted driver to survive the same crash because the one travelling with the belted driver is more likely to be belted. Such a systematic relationship between survivability of the control occupant and the factor being investigated (effectiveness of driver belt use) would violate one of the important assumptions on which the method [1] rests.

A strength of the method is that many estimates of device effectiveness can be obtained by choosing different control occupants, disaggregated not only by belt use (which is required) but, depending on the amount of available data, by other characteristics such as occupant age, or control occupant seating position. Below, safety belt effectiveness is estimated for drivers and right front passengers by obtaining a weighted combination of 46 individual estimates.

Data

All the estimates used Fatal Accident Reporting System (FARS) data [2]. As this data set contains information on fatal crashes only, the results apply exclusively to fatalities, and should not be generalized to other levels of injury. The data used to determine effectiveness of the three occupant protection devices are summarized in Table 1; all complete years of FARS data available when the original studies were performed were used. Data for which occupant protection device use was coded as unknown for either subject or control occupant were excluded. In all cases the analyses were confined to adult (16 years or older) occupants.

For the front seat study, only data for passenger cars model year 1974 or later were included in the analysis; 1974 was the first model year for which manufacturers were required to equip all cars with integrated three-point lap/shoulder belts for the driver

Table 1. Summary of data.

Occupant protection device	FARS years	Model years	Subject occupant	Fatalities	
				Device used	Not used
Lap/shoulder belt for car front seats	1975 thru 1983	≥1974	Right passenger	716	15 595
			Driver	711	14 738
Lap belt only for car rear seats	1975 thru 1985	All	Right passenger	322	12 211
			Left passenger	215	10 333
Helmet for motorcycle riders	1975 thru 1984	All	Driver	938	1 109
			Passenger	758	1 081

and right front passenger seating positions. All drivers or right front passengers in such cars coded in FARS as using any type of restraint are assumed to use this system. For the rear seat study, all model years were included, and any occupant coded as using any restraint was presumed to be using a lap belt. These assumptions regarding which restraint system was used were necessary because, due to coding problems in FARS, it is not possible to identify specifically which car restraint system was used. For the motorcycle study helmet use was specifically coded.

Note further that all the estimates assume that actual use, or non-use, of an occupant protection system is as coded in the FARS data. Any systematic errors in coding restraint use could, of course, generate systematic errors in effectiveness estimates. The influence of some possible coding errors on double pair comparison estimates is discussed in [13].

Effectiveness is estimated essentially from those cases in which there is a difference in protection device use between subject and control occupant. There is a strong tendency for use, or non-use, to be the same for different occupants of the same vehicle. Hence, sample sizes in the really important cells are even smaller than would arise if occupant protection device system use were distributed randomly among occupants to generate the totals in Table 1. The sample sizes, which were different for each subject and control occupant pair, are given in the original papers [3- 5].

Results

Lap/shoulder belt effectiveness for car front seat occupants

In order to obtain various essentially independent estimates of safety belt effectiveness, driver and passenger (in this section, right front passenger) were disaggregated into the following three age categories:

16-24 years

25-34 years

$\geq$35 years.

The calculation is illustrated by a specific example of "younger" drivers in the 16-24 years age category as the subject occupant and "older" passengers in the $\geq$35 years age category as the control occupant. From the FARS data for cars containing younger belted drivers and older unbelted passengers (at least one being killed) we find that 5 belted younger drivers were killed while travelling with older unbelted passengers, while 18 older unbelted passengers were killed travelling with younger belted drivers. For cars containing unbelted younger drivers and unbelted older passengers we find that 249 younger unbelted drivers were killed travelling with older unbelted passengers and 560 unbelted older passengers were killed travelling with younger unbelted drivers. Substituting these values into Eq. 5 gives, for this specific example,

$$E(\%) = 100 \times (1 - \frac{AD}{BC}) = 100 \times (1 - \frac{5}{18}\,\frac{\times 560}{249}) = 37.5\%. \quad (6)$$

As shown in [1], the standard error, $\triangle E$, in the estimate of E can be obtained as a function of the four fatality frequency counts, A, B, C and D; for the present example $\triangle E$ is calculated [3] as 32.5%. However, because the definition of E necessarily constrains it to values in the unsymmetrical interval from minus infinity to 100%, errors are not symmetric. It is therefore suggested in [4] that for large errors, it is more appropriate to display an error range. For the present example this range is from -6% to 63%, the large range reflecting the small amount of data in one of the cells (A = 5). The interpretation is that, based on the data for this specific case, there is a 68% chance that the true effectiveness is in the indicated range, a 16% chance it is higher than the upper error limit of 63% and a 16% chance it is lower than the lower error limit of -6%.

Applying the same calculation to the data for all combinations of age category and belt use (Table 2) of the control occupant generates 6 independent estimates of safety belt effectiveness for *drivers aged 16-24 years*. The weighted average, obtained as described in [1,3], for these six values is (55.7 ± 6.2)%.

Table 2. Calculation of the effectiveness of safety belts in preventing fatalities to drivers aged 16 to 24 years using as the control occupant passengers (in the right front seat) in three age categories.

Control occupant characteristics	A C	B D	r_1 r_2	R	E(%)	Error range (%)
Unbelted passenger aged 16-24 years	52 5150	108 5000	0.481 1.030	0.467	53.3	43 to 62
Unbelted passenger aged 25-34 years	9 673	23 703	0.391 0.957	0.409	59.1	38 to 73
Unbelted passenger aged ≥35 years	5 249	18 560	0.278 0.443	0.625	37.5	-6 to 63
Belted passenger aged 16-24 years	90 81	92 28	0.978 2.893	0.338	66.2	55 to 75
Belted passenger aged 25-34 years	19 7	15 5	1.267 1.400	0.905	9.5	-80 to 54
Belted passenger aged ≥35 years	8 3	23 3	0.348 1.000	0.348	65.2	13 to 86
Weighted average values				0.443	55.7	49 to 62

Average effectiveness of lap/shoulder belts in preventing fatalities to drivers 16 to 24 years = (55.7 ± 6.2)%

Applying the above procedure to data for drivers in the other age categories (raw data are given in [3]) generates the effectiveness estimates summarized in Table 3. By weighting the effectiveness for each age category by the number of fatalities in that age category one obtains an overall estimate for safety belt effectiveness in preventing fatalities to drivers based on the right front passenger as the control occupant of $(42.6 \pm 4.5)\%$.

All the above results are estimates for the effectiveness of safety belts in reducing driver fatalities, using the right front passenger as the control, or exposure estimating, occupant. The whole procedure is symmetric with respect to these two occupants, so that we can use the driver to estimate effectiveness for the right front passenger, leading to the results in Table 4. As before, by weighting by the number of fatalities in each age category, an estimate of effectiveness for right front passengers based on the driver as the control occupant is obtained as $(41.1 \pm 4.9)\%$.

The different effectiveness estimates for the different age groups in Tables 3 and 4 are probably reflecting, in part, that effectiveness is different in different types of crashes [14,15]. The crash types for which belts are most effective (frontal crashes, rollovers, single vehicle crashes) are associated more with younger drivers. Different fatality likelihood as a function of age [16] should not cause restraint effectiveness to depend on age in any obvious way, because the increased risk with increased age is present whether the occupant is belted or unbelted.

All the above is based on either the driver or the right front passenger as the control occupant. However, occupants in *any* seating position may serve as control occupants. Effectiveness estimates based on combining results for other occupants seated in the middle front or any of the three rear seating positions are shown in Table 5, together with the previously derived estimates.

Thus, by combining all the individual estimates (a total of 46 - see [3]) of effectiveness we estimate that lap/shoulder safety belt effectiveness in preventing fatalities is $(42 \pm 4)\%$ for drivers and $(39 \pm 4)\%$ for right front passengers.

In [3] these were combined by weighting by occupancy rates in driver and right front passenger seating positions, and further incorporating, with appropriate weighting, an effectiveness estimate based on summarizing a number of earlier field studies [17]. The overall conclusion was that if all members of a presently unbelted population of drivers and right front passengers were to use the provided three point lap shoulder belts, but not otherwise alter their

Table 3. Estimation of effectiveness of safety belt in preventing driver fatalities.

Driver age (years)	Safety belt effectiveness	
	E%	ΔE%
16-24	55.7	6.2
25-34	23.9	11.9
≥35	41.0	6.9
Weighted average	42.6	4.5

Table 4. Estimation of safety belt effectiveness in preventing right front passenger fatalities.

Passenger age (years)	Safety belt effectiveness	
	E(%)	ΔE(%)
16-24	53.4	6.8
25-34	29.3	12.1
≥35	31.5	8.4
Weighted average	41.1	4.9

Table 5. Summary of estimates of safety belt effectiveness using as the control occupant: 1) the driver or right front passenger disaggregated into three age categories; 2) passengers in the middle front seat or any of three rear seats.

Control occupant	Effectiveness estimates, %	
	Driver	Right front passenger
Right front passenger or driver disaggregated by age	42.6 ± 4.5	41.1 ± 4.9
Passenger in middle front or any rear seat	40.8 ± 7.0	34.2 ± 8.7
Weighted average	42.1 ± 3.8	39.2 ± 4.3

behavior, then traffic fatalities to this group would decline by $(43 \pm 3)\%$.

Lap belt only effectiveness for car rear seat passengers

Estimates of restraint system effectiveness for rear seats are more uncertain than the corresponding estimates for front seats because two effects combine to reduce greatly the number of data. First, occupancy rates are lower in rear than in front seats. Second, restraint system wearing rates are even lower in rear than in front seats.

Restraint system effectiveness is estimated for two of the three possible rear seating positions, namely right rear and left rear; there are insufficient data to estimate effectiveness for center rear seats. Drivers, right front passengers, and the non-subject outboard position rear passenger act as control occupants. There are too few data to disaggregate by occupant age, as was done for the front seat case. This is not considered an important disadvantage because for the front seat case the results obtained without age disaggregation were in close agreement with those obtained using age disaggregation (see p. 234 of [3]).

The data in Table 6, which are in the same form as those in Table 2, indicate that the effectiveness of lap belts in preventing fatalities to right rear passengers is $(17.3 \pm 8.7)\%$. The corresponding analysis for the left rear seat (see Table 2 of [4]) gives an effectiveness of $(19.4 \pm 10.0)\%$.

Helmet effectiveness for motorcycle drivers and passengers.

Helmet effectiveness is determined for motorcycle drivers and passengers by using each as the control

Table 6. Estimate of effectiveness of lap belts in preventing fatalities to right rear passenger.

Control occupant characteristics	A / C	B / D	r_1 / r_2	R	E(%)	Error range (%)
Unrestrained driver	31 / 4876	48 / 6264	0.646 / 0.778	0.830	17.0	-7 to 36
Unrestrained right front passenger	28 / 4390	46 / 5922	0.609 / 0.733	0.831	16.9	-8 to 36
Unrestrained left rear passenger	12 / 2536	23 / 2557	0.522 / 0.992	0.526	47.4	24 to 64
Restrained driver	110 / 228	93 / 160	1.183 / 1.425	0.830	17.0	-2 to 32
Restrained right front passenger	99 / 165	87 / 130	1.138 / 1.269	0.897	10.3	-11 to 28
Restrained left rear passenger	42 / 16	40 / 16	1.050 / 1.000	1.050	-5.0	-61 to 32
Weighted average values				0.827	17.3	8 to 26

Estimated average restraint system effectiveness in preventing right rear passenger fatalities = (17.3 ± 8.7)%

Table 7. Calculation of the effectiveness of helmets in preventing fatalities to motorcycle drivers.

Control occupant (passenger) characteristics	A / C	B / D	r_1 / r_2	R	E(%)	Error range (%)
Unhelmeted male	89 / 626	100 / 472	0.890 / 1.326	0.671	32.9	19 to 44
Unhelmeted female	28 / 410	34 / 475	0.824 / 0.863	0.954	4.6	-27 to 28
Helmeted male	453 / 34	353 / 14	1.283 / 2.429	0.528	47.2	26 to 62
Helmeted female	368 / 39	356 / 35	1.034 / 1.114	0.928	7.2	-21 to 29
Weighted average values				0.748	25.2	15 to 34

Average effectiveness in preventing motorcycle driver fatalities = (25.2 ± 9.4)%

for the other. The age distribution of fatally injured motorcycle riders (see Figure 1 of [5]) is considerably narrower than that for drivers in general [18] or for car drivers [19], or for all traffic crash victims [18]. This facilitates performing the analysis in such a way that subject and control occupant are similar in age. The data used in the analysis were confined to cases in which the driver age and passenger age were within three years of each other; this involved discarding about 40% of the data. However, by so doing we eliminate potentially confounding effects due to the dependence of survivability on age [16]. Potentially confounding effects due to the dependence of survivability on sex were removed by disaggregating the data by occupant sex. The data for female driver fatalities, being too few for analysis, were discarded.

Table 7 shows the analysis for drivers and Table 8 for passengers. These tables are from Tables 2 and 3 of [5].

Table 8. Calculation of the effectiveness of helmets in preventing fatalities to motorcycle passengers.

Occupant characteristics: SUBJECT=PASSNGR	Occupant characteristics: Control=driver	A / C	B / D	r_1 / r_2	R	E(%)	Error range (%)
MALE	Not helmeted	14 / 472	34 / 626	0.412 / 0.754	0.546	45.4	23 to 61
MALE	Helmeted	353 / 100	453 / 89	0.779 / 1.124	0.694	30.6	16 to 43
FEMALE	Not helmeted	35 / 475	39 / 410	0.897 / 1.159	0.775	22.5	-1 to 40
FEMALE	Helmeted	356 / 34	368 / 28	0.967 / 1.214	0.797	20.3	-6 to 40
Weighted average values					0.707	29.3	20 to 38

Average effectiveness in preventing passenger fatalities = (29.3 ± 8.9)%

It was shown in [5] that the effectiveness estimates were reasonably robust when the analysis was performed in a variety of ways. For example, including data for occupants of all ages, or keeping data only when subject and control occupant age agreed to within some amount, the amount having various values in addition to the three year criterion used in the main analysis.

A further result, not related to helmet effectiveness, was that for occupants of the same sex, same helmet use, and similar age, the fatality risk to the driver was $(31 \pm 2)\%$ greater than that to the passenger. The method used to investigate the relative safety of the two motorcycle seating positions is applicable to other vehicles, and is presently being used to investigate the relative safety of the various seating positions in passenger cars.

Summary and Conclusions

The previously discussed estimates are listed in Table 9. The error in the final composite estimate (which is without regard to the specific seating position) is in each case taken as the smaller of the two individual estimates; if, say, the estimates for driver and right front passenger were independent, then the composite error would be less than either individual error. However, as these two estimates are not independent, but are in fact based on a different organization of the same data, we make the conservative choice of not diminishing the smaller error further.

The overall effectiveness associated with each of the three devices is summarized in Table 10. The result for lap shoulder belts may be compared to the estimate of 40% to 50% based on summarizing the results of a number of prior studies [17]. The rear belt result may be compared to the estimate of 17% to 26% derived from analyzing FARS and State data [20]. The results may also be compared to estimates from a quite different type of study, one not dependent on coding of field accident data [21]. In [21], detailed crash case records for 706 occupants fatally injured in actual crashes were examined to determine, based on engineering judgment, the potential of various occupant protection devices to reduce fatalities. This led to the following effectiveness estimates; 31% for lap and shoulder belt; 17% for lap belt only; 18% for air cushion only; and 29% for air cushion and lap belt.

The effectiveness result for rear belts reflects only reductions in fatality risk to the wearers of the belts. However, recent research [22] indicates that the presence of an unbelted rear seat occupant increases the fatality risk to *front seat occupants* by $(4 \pm 2)\%$, presumably because of the increased loading force that the unbelted rear occupant imposes on the front occupant.

Table 9. Empirically determined effectiveness of three occupant protection devices. In all cases effectiveness means the reduction in fatalities which would occur if a population not using the protection device were to change to universal use. The uncertainty indicated is plus or minus one standard error.

Vehicle	Occupant	Protection device	Effectiveness in preventing fatalities	
Car	Driver	Lap/shoulder belt	(42 ± 4)%	(41 ± 4)%
	Right front passenger	Lap/shoulder belt	(39 ± 4)%	
Car	Left rear passenger	Lap belt	(19 ± 10)%	(18 ± 9)%
	Right rear passenger	Lap belt	(17 ± 9)%	
Motorcycle	Driver	Helmet	(25 ± 9)%	(27 ± 9)%
	Passenger	Helmet	(29 ± 9)%	

The results (Table 10) show that all three devices reduce fatalities. Fatality reductions of the magnitudes found for any of these devices would be viewed as major achievements if they were associated with a new medical procedure or drug. Some earlier discussions of protection devices have been characterized by unrealistically high expectations, stimulated in part by prior flawed analyses (see, for example, the discussion on p. 239 of [3]).

As discussed in formal mathematical terms in [23], overall effectiveness of any occupant protection device in preventing fatalities in actual use depends on two factors:

1. The specific dependence of effectiveness on severity in crashes, which flows from the engineering of the device and its relation to human biomechanics.
2. The actual distribution of crashes by severity that occurs in real traffic. This cannot be determined in the laboratory.

Table 10. Summary of the effectiveness of the three devices in preventing fatalities.

Occupant protection device	Effectiveness
Lap/shoulder belts in outboard front seats of cars	(41 ± 4)%
Lap only belts in outboard rear seats of cars	(18 ± 9)%
Motorcycle helmets	(27 ± 9)%

The second factor necessarily places an upper limit on the achievable effectiveness of any device. Basically, a sufficient number of crashes occur at such extreme levels of severity that there is little opportunity for mitigation of injuries. The great majority of crashes are at so low severity that even the unprotected occupant is not harmed, so that the device can generate no injury reducing benefit for these crashes. Laboratory testing naturally focusses, as it ought to, on the crash severity regime where the device is expected to provide benefits; a comprehensive examination of the relationship between laboratory tests and occupant protection can be found in [24]. An additional important consideration is that a surprisingly large number of fatal crashes are of a bizarre nature not readily encompassed in any laboratory testing program (for example, foreign objects entering the passenger compartment, cars being dragged for long distances along a railroad track, etc., etc.). Such events reduce overall field effectiveness, and will do so for occupant protection devices in general.

Thus the "window of opportunity" for occupant protection devices to generate benefits is sufficiently narrow that very high effectiveness is unlikely to be achieved by any practical means. Such an unavoidable conclusion underlines the paramount importance of avoiding crashes [25], which is 100% effective at preventing injury. Crash avoidance, improving occupant protection, and increased use of available occupant protection devices all have important roles to play in reducing occupant injury.

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Fatal Injuries to Restrained Children Aged 0-4 Years, in Great Britian 1972-86

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Abstract

Little information is available on the performance of child restraints in severe accidents; some has come from 'in-depth' studies of relatively few accidents and from less detailed investigations of a large number of accidents, many of them rather minor. For this reason, a study has been made of all fatalities to restrained children in Great Britain since 1972 from Police and Coroners' records. This paper describes the results of this study for children aged up to four years. Some accidents have been simulated by sled tests as an aid to understanding the mechanism of injury. In most cases, the severity of the accident was such that it was unlikely that an improved restraint design could have prevented the fatality. This report summarizes the cases studied and draws conclusions about the few cases where the protection afforded by the restraint was impaired.

Introduction

Child restraints in the UK for children are specified by the maximum mass of the child for which they are designed. The two lowest mass groups correspond approximately to children aged up to 1 year and to children aged 9 months to 4 years. Restraints for these groups can be approved to an appropriate British Standard (1,2,3) or ECE Regulation (4). These standards all include a dynamic test of the restraint but the ultimate evaluation of a restraint system is how it performs in road accidents.

The performance of child restraints in accidents is being monitored in three ways. Most child restraints in the UK are sold with a questionnaire with the request that this be completed and returned to the TRRL in the event of an accident in which a child was in the car whether restrained or not. This provides only limited information on each accident, the accidents are often of low severity and the accident distribution of the responses is likely to be biassed but this method does provide a reasonable sample size (currently in excess of 5000 child car occupants). The second way is as part of the detailed accident investigation programme of TRRL. However, in the current investigation where the total sample size is 2720 car occupants, only 51 were children aged under 5 years of whom only 10 were restrained and only 1 of these was injured at AIS2 or greater.

The third way is to investigate accidents in which children were killed while restrained in order to evaluate their performance in the most extreme conditions. In particular, any possible failure of the restraint to provide protection in circumstances where this should be practicable are sought with a view to the amendment of the relevant standard where appropriate.

This paper reviews all the reported accidents to children aged under five who were killed as car passengers while restrained in the UK from 1972 to 1984 together with three more investigated from 1985 and 1986. The rate is only two or three per year. Some sled tests, performed to help understand some of these accidents, also are reported.

Study Method

Road accidents in the United Kingdom that involve personal injury and are reported to the police are recorded on a standard form and are collated to produce a single data base known as Stats 19. Many minor accidents will not appear on the data base but most serious and all fatal accidents should appear. This data base was used to identify the accidents in which a restrained child car occupant was killed. The police records of these cases were examined for details of the accident circumstances and the corresponding coroner's reports were reviewed for the detailed information on the injuries suffered by the child. In most cases there was sufficient information and photographs to enable a judgement to be made on the cause of the fatality with some confidence. In a few cases, the occurrence of a fatality to a restrained child occupant became known through other sources prior to the Stats 19 data availability and a rapid follow up including a vehicle inspection was possible.

This procedure has been followed for all restrained fatal child car occupants under the age of 13 years since 1972, although this paper refers only to children aged under 5 years.

Results

A brief summary of every accident studied to date is given in the table in the appendix. The first observation is the small number of fatalities to restrained children in each year.

There were, on average, between two and three fatalities to restrained children per annum between 1972 and 1984, the largest number in a year being six for both 1982 and 1983. This compares with an average of 30 per annum for all car occupants aged under 5 and about 2500 for all car occupants regardless of age. It would be expected that this number would increase as the wearing rate increases. In this period, the restraint use by children of this age group has increased from 17 per cent in 1974 (5) to 36 percent in 1984 (6). It is difficult to obtain a measure of exposure, but in 1984 there were 104 children aged less than five years in cars involved in fatal accidents where the child was at least slightly injured. The Stats 19 data base does not record the presence of uninjured occupants.

General Description of Sample

The sample comprises all 30 fatalities to restrained child car occupants aged less than five years for the years 1972-84 together with three cases from the years 1985 and 86.

Age Distribution. A high proportion (45 per cent) of the child fatalities were aged between 12 and 23 months. Six (18 per cent) were younger and twelve (36 per cent) were aged 2-4 years. This should not be taken to imply that younger children are at more risk of injury in these restraints but that the usage rate is higher for very young children and reduces as the children get older. In the TRRL questionnaire sample, where there is no selection by injury, 40 per cent of the restrained under five sample were in the age group 12-23 months.

Direction of Impact. Table 1 shows the main types of impact in this sample. 'Front' contains all cases where the main damage area was restricted to the front of the vehicle, although the main direction of force may be angled. Two categories of side impact were given because a large proportion of these cases involved intrusion into the passenger compartment in a direction close to a frontal impact. The proportion of frontal impacts among the cases, while still the most frequent, is much lower than is usually quoted for fatalities. (See, for example ref. 7). The proportion of side impacts (perpendicular and angled) is almost as high as for frontal impacts and rear impacts feature significantly also. This is undoubtedly due to the facts that the children were restrained and mostly were in the rear seats. Only two of the 33 were in front seats.

About one quarter of the accidents involved an under-run with a goods vehicle or public service vehicle (bus or coach). This is where the main load paths of the goods vehicle are substantially above those of the car leading to direct intrusion to the passenger compartment. This compares with about 8 per cent for all fatally injured car occupants, (8), suggesting that this type of impact is more important for restrained young children.

Impacted Object. Table 2 shows the object struck by the case vehicle. Three of the children (9 per cent) were involved in impacts with medium goods vehicles (3500-7500 Kg gross weight) and 9 (27 per cent) were involved in impacts with heavy goods vehicles (over 7500 Kg gross weight). This latter compares with an average of 15 per cent for all fatally injured car occupants over the years 1974-84 (5). The higher rate of involvement of goods vehicles in the child fatality accidents suggests that these might have been more

Table 1. Category of impact.

Impact type	No.	Per Cent
Front	11	33
Side (perpendicular) *	4	12
Side (angled) *	6	18
Rear	7	21
Overturn	2	6
Other	3	9
	33	
Involving under run	8	24
Involving fire	1	3
Involving drowning	1	3

* see text

Table 2. Impacting objects involved in restrained child car occupant fatalities.

Objects impacted	No. of children
cars	15
vans/light goods vehicles	2
(medium/heavy goods) (vehicles/buses)	12
Pole/tree	2
Nothing	2
	33

severe accidents on average than for all fatal car occupants. This is probably associated with the facts that the children were mainly in the rear seat and, for the period 1974-82, most of the 'all car occupants' sample would have been unrestrained.

Restraints Used. During this period, the main restraint available for babies, up to 10 kg, was the carry-cot restraint. This is two straps designed to hold an ordinary carry-cot or pram top transversely across the rear seat of a car. Rear facing infant carriers became available in the United Kingdom in 1985. Neither of these restraint types were used in this sample. For children of 9 to 18 Kg, (about 9 months to 4 years old) the conventional child restraint is the child chair. Until the last few years the design was of a hard shell seat attached to the car structure by two top straps and two lower straps. A five point child harness comprising lap strap, two shoulder straps and a crotch strap is attached to the shell or through the shell to the attachment straps. More recently frame seats have become available where the hard shell seat is fixed to a tubular metal framework which sits on the vehicle seat. The advantage of this type is that it only requires two lower straps to attach it to the car structure. Forward movement of the top of the chair is restricted by the reaction of the seat cushion on the frame. The disadvantage is that it is difficult to achieve the same forward movement control this way as for the four point designs where it is controlled by the direct attachment of an anchor strap. Some designs of frame seat can also use an adult seat belt as an attachment, and the diagonal strap can give added forward movement control. Child harnesses, (a lap strap and two shoulder straps), are available for children of 18 Kg. to 36 Kg (about 4-10 years) but these are sometimes used by younger children.

Table 3. Restraints used by fatally injured child care occupants 1972-1986

Restraint type	No.
Child seat - 4 point	22
Child seat - 2 point (frame)	4
Child harness	4
Adult belt	1
Non-approved child seat	2

Two child restraints, which were not approved to either the appropriate British Standard on the ECE Regulation, were in the sample and both of these were in accidents prior to 1978.

Table 3 shows the types of restraint used in this sample.

Three of the four 2-point (frame) seats were involved in frontal impacts compared with only 4 of the 22 4-point seats. However, this number is so small that no definite conclusions can be drawn from this observation alone.

Fatal Injuries.

Table 4. Frequency of fatal injuries by body area.

Body Area	No.
Head	24
Neck	10
Chest	2
Abdomen	3
Drowned	1

NOTE—more than one body area may have received fatal injuries in any one child.

The number of fatal injuries to different body areas that were reported are shown in Table 4. These figures may not be completely accurate owing to the method of collection but the differences are sufficiently great to indicate with confidence that the head is the body area most frequently seriously injured. Neck injuries are also of concern.

Injury Causes: Child Seats. In most of these cases, the accidents were very serious and it would have been unlikely that other child restraint designs could have prevented the fatality. There was one instance (77/1) where a non-approved "hook-over" seat was involved in a frontal accident and was thrown, complete with occupant, into the front of the passenger compartment and the child suffered a fatal head injury. One child was drowned while restrained in a child seat when the car was submerged. There is no suggestion that the restraint affected the outcome; two unrestrained children in the same car also were drowned. In one severe frontal impact (80/2) the child was found in the front footwell with a fatal neck injury and head injuries. One of the first witnesses reported that the child seat harness was undone when found. It seems quite possible that, in this case, the child was not properly restrained prior to the accident.

In another frontal impact (84/3), with a velocity change of about 25 mile/h, a child in a frame (2-point) seat with a four-point harness (no crotch strap) suffered a fatal fracture of C1-C2 with rupture of the anterior ligaments, laceration under the chin and other neck injuries. These injuries are commensurate with hyper-extension of the neck. It seems likely that the absence of a crotch strap and a rather loose lap strap enabled the 16 month old child to slip under the lap strap, finally being restrained under the chin by the buckle. Reconstruction of this accident on a test sled, with a 9 Kg dummy using the same model of child seat with straps adjusted to that found after the accident, reproduced just such submarining. A second case relating to the use of a crotch strap (85/3) involved a severe frontal impact followed by rotation and a child restrained in a four-point child seat. The top straps of a four-point child seat are intended to be attached to the car structure some way behind the top of the child seat so that they help to restrain the car seat backrest in a frontal impact. In this case the top straps were wrapped round the backrest which was of the split type, joined to the lower straps and attached to the lower seat belt anchorages. The backrest latch failed in the impact, allowing the seat back and child seat to rotate forwards. This, and the rotation of the car, caused most of the deceleration forces of the child to be taken by the left lap strap which failed at the adjuster. The child was ejected and found in the footwell with a fractured neck, damaged cervical cord, fractured ribs, lacerated liver and other chest injuries. Photographs at the scene taken after the accident show the crotch strap unattached at the buckle but the lap strap attached. As the child was not in the seat after the accident, there seems no reason for the rescuers to release the crotch strap. It seems unlikely, therefore that the crotch strap was attached prior to the accident. Reconstruction of the accident conditions on a sled showed that the dummy was ejected if the left lap strap was severed and the crotch strap missing, but retained if the crotch strap was attached.

Subsequently, the British Standard for these child restraints (BS3254-Ref 2) has been amended to mandate for the presence of a strong crotch strap.

Accident 84/7 involved the use of a two-point reclining child seat. The father had attached a non-standard top tether to this seat to aid stability. The accident involved a severe frontal impact to a tree and the child suffered a fracture-dislocation between C3 and C4 (neck) together with bruising under the chin, at the base of the tongue and at the front of the cervical spine. The injuries suggest a submarining problem similar to that in case 84/3 except that the frame seat in this case has a five-point harness, including a crotch strap. The 10 month old child in 84/7 was reported as being still strapped in after the accident, although the crotch strap was not specifically mentioned. A reconstruction of these accident conditions on a sled failed to produce submarining whether the crotch strap was attached or left disconnected. Neither did the test results suggest that the addition of the top tether increased the chest acceleration significantly. The cause of this fatal injury is not clear.

Case 82/3 was probably the most severe frontal impact seen in the study, with a velocity change of 35 miles/h or more. The child suffered a fatal brain contusion with no evidence of head contact and six fractured ribs. This child seat, which was a four point attachment design, was inspected in the damaged vehicle. It was correctly fitted to the vehicle and the strap adjustments were appropriate for the size of child. A sled test at 35 miles/h gave a very much higher chest and head accelerations than a test at 30 miles/h with the same model child seat. However, this case appears to be very unusual and it is possible that the child's brain was particularly susceptible to this type of injury.

Case 86/x concerned a two point frame seat in the rear seat of a car involved in a frontal impact with a velocity change of about 25 miles/h. These accident conditions should easily be survivable by a child in a child restraint but this occupant suffered a fatal neck injury. The child seat in this instance was inspected and the strap adjustment positions were noted. Reconstruction of this impact showed that, if the child seat was positioned prior to the impact in such a way that the attachment straps were very slack, fairly high chest accelerations could be generated. Also the maximum forward excursion of the head of the dummy was quite high. Comparison with the space available inside the particular vehicle suggests that it was highly likely that the head of the child would have made contact with the backrest of the front seat, although there was no injury nor evidence on the car seat to confirm this. This leaves two possibilities for the cause of the fatal injuries; high chest acceleration resulting in high head acceleration fracturing a weak neck, or head contact whilst the torso was rotating forwards, causing hyper-extension injuries to the neck. Whichever the mechanism, the importance of correctly adjusted straps is clear.

The last accident reported (86/y) involved a side swipe with intrusion inwards and rearwards of the B-pillar. The child was in a two point frame seat and died when his head made contact with the deformed B-pillar. The nature of the impact suggested that the velocity change would have been perhaps 15-20 mile/h. It was considered possible that the relative forward movement of a child in a 2-point frame seat, compared with a four-point seat, may be greater at lower speeds. Sled tests to evaluate this did not confirm this. In this accident, the rear seat back of the hatchback failed and it rotated forwards under the load of some luggage. This extra loading on the child seat undoubtedly resulted in greater forward movement, permitting the head contact. It is unlikely that a fully restrained child seat in this instance could have prevented head contact with the deformed B-pillar, but it might have reduced the severity of the injury.

For the remaining fatalities to child seat occupants some consideration was given to possible countermeasures, although it is not clear whether some of the fatalities could actually have been prevented. In two cases (72/2b) and (83/10) it seems likely that fatal head injuries were caused in rear impacts by impact of the child's head, through the child seat back, with the hard raised edge at the top of the rear seat backrest. This used to be a common feature in such estate cars and hatchbacks. Energy absorbing padding in the head area of the child seat might help also, but children using adult belts or child harnesses could still be vulnerable. Under-run guards, front and rear, or other means of achieving better compatibility between goods vehicles and cars would have been helpful in six cases. In a further six cases, better side impact protection in the car might have avoided some fatalities. A stronger roof structure could have proved helpful in three cases.

Injury Causes: Child harnesses. The four cases of fatalities to children in child harnesses had different causes. The first case (72/1) shows typical submarining injuries to the abdomen. It is known that child dummies can submarine in child harnesses but this is the only fatal case showing this feature. The harness had been fitted only the previous day and it is possible that it had not been correctly adjusted with the lap strap down low on the thighs. The second case (72/2a), a severe rear impact, was a fatal injury to the head caused by contact with the hard ridge at the top of the rear seat backrest, as occurred in two child seat cases. The third case (81/1) involved an angled sideswipe into the driver's side of the hatchback vehicle which deformed the B-pillar inwards and tore off the rear door. The child was in the rear seat on the struck side in a harness which was incorrectly mounted. The top attachment straps were run immediately down behind the backrest, providing no backrest restraint. This gave poor restraint and the child was ejected through the open doorway. The final case (81/4) involved an angled side swipe into the rear of a goods vehicle (milk truck). The child's head made direct contact with the intruded load platform. In this case an under-run guard might have helped.

Injury Causes: Adult Belt. There is only one case in this fatal sample in which the child was using an adult lap and diagonal belt. This was a severe frontal impact, with intrusion of the A-pillar and facia. The child was in the front passenger seat and behind him was an unrestrained adult. Rear loading by this adult caused the internal injuries (spleen, lungs) and probably the excessive forward movement leading to the head contact. Improved intrusion resistance and the use of rear seat belts would probably have prevented this fatality.

Injuries: Remedial measures. Of all the cases in this sample, it is judged that in three the fatal injuries could have been avoided by an improvement in the design and performance of the child restraint; in five, correct adjustment and fitting of the restraint could have avoided the fatality and in 18 cases, design changes to the car or struck vehicle would have been necessary. In the remaining 7 no obvious practical remedial measure could be suggested.

Dynamic Tests

Some aspects of a few of the fatal cases were reproduced in sled tests as an aid to understanding the mechanism of the injuries. The specific conclusions relating to those cases have been incorporated in the discussion of the injury causes in the previous section. This section describes the test results and the general conclusions from these tests. The measurements of head forward excursion in the tables are given with respect to a point on the test seat which is the intersection between the undeformed top surface of the seat cushion and the front surface of the backrest cushion. The test seat used was that specified in British Standard BS AU202 (1) and in ECE Regulation 44 (4).

Case 82/3. In this case a 15 month old child suffered brain injuries in a high energy frontal impact while restrained in a four point child seat incorporating a five strap harness. Table 5 shows the peak chest acceleration and maximum head forward excursion for tests at 30 mile/h and 35 mile/h using this model of child seat, adjusted as found after the accident using a TNO P3/4 child mannekin.

The British Standard to which this restraint is approved includes a dynamic test at 30 mile/h. The restraint survived the 35 mile/h impact but it can be seen that the chest acceleration increased dramatically over the value at 30 mile/h.

With such a low head excursion, even at 35 mile/h, these test results confirm the injury evidence that head contact was unlikely to have occurred except on rebound.

Case 84/3. This involved a child in a 2-point seat with an integral four point harness (no crotch strap). In the frontal impact, with an estimated velocity

Table 5. Test results with child seat involved in case 82/3.

Impact Speed	Maximum Chest Acceleration (g)	Maximum Head Excursion (mm)
30 mile/h	43	313
35 mile/h	95	380

change of 25 mile/h, the child suffered a fractured neck and other ligament and soft tissue injuries at the front of the neck. The accident was reconstructed twice on the sled with the same model child restraint and with the lap and shoulder straps adjusted as found after the accident. On both occasions the dummy slipped under the lap strap and was caught under the chin by the lap strap, confirming this as the likely mode of injury.

Case 84/7. This case involved a frontal impact into a tree with an estimated velocity change of 25 mile/h. The child was restrained in a two point frame seat with an integral five point harness and this was in its reclined position. The child's father had attached a non-standard top tether to aid stability. Pictures of the child seat after the accident show it to be slightly twisted to the left. The child suffered neck injuries typical of submarining, but this model of child seat has a crotch strap.

Three impact tests were performed with this model of child seat and a top tether. For the first (84/7-1), the harness straps were adjusted as seen in the post accident photographs and the attachment straps tight. In the second test (84/7-2) the right hand attachment strap was loosened by 100 mm to reproduce the final position seen in the photographs and the crotch strap was left unattached. In the third test (84/7-3), both attachment straps and the top tether were loosened by 100 mm. The results are shown in Table 6.

No suggestion of submarining could be observed in these tests. Neither were the chest accelerations unduly high.

The head excursion figures show that, under these conditions, head contact with the front seats of the car was unlikely to have occurred.

Case 85/3. This involved a four point attachment child seat incorporating a five point harness. The top attachment straps were wrapped round the backrest. This failed in the accident causing excentric loading to the lap strap which failed at the left adjuster. The child was ejected from the restraint. The photographs after the impact suggest that the crotch strap may not have been attached and it was this aspect that was investigated in the sled test. Two tests were performed at 30 mile/h with this model child seat in which the left hand lap strap was cut. In the first test the crotch strap was attached and the dummy swung round but

Table 6. Test results with child seat involved in case 84/7.

Run	Impact Speed mile/h	Peak head acceleration (g)	Maximum head excursion (mm)
84/7-1	25	31	342
84/7-2	25	35	349
84/7-3	25	50	419

was retained in the child seat. In the second test in which the crotch strap was not attached, the dummy was ejected, after loading the straps in a way which would have produced just the injuries suffered by the accident victim.

Case 86/x. In this case, a child in a two point frame seat suffered a fatal neck injury without evidence of head contact. The attachment straps were found after the accident to be adjusted so that the front lower rail of the child seat frame reached the front of the car seat cushion. The child harness straps were also rather loose and their adjustment positions were measured. A number of tests were performed to assess the effect of strap adjustment on the performance of two and four point attachment child seats and on the pre-impact position of the seats.

Test results are shown in Table 7. Model A was the child seat involved in the accident, model B was another 2 point child seat, both reclining seats, and model C was a 4 point child seat. The impact speed for all tests was 25 mile/h.

The results show that the addition of slack into the restraint system can markedly increase the chest acceleration, particularly for the 2 point frame seats. The peak chest acceleration for the tests reproducing the accident conditions are high for both 2 point models; both exceeding the 55g limit imposed by ECE Regulation 44. The maximum forward head excursions also increase with slack with values of about 600 mm for the accident conditions. The top of the front seat backrest of the case vehicle, measured in the same way from the back seat intersection point, is 420 mm when the front seat is fully back and 790 when it is fully forward. Although the backrest is angled so that this distance is greater lower down, it seems quite possible that the child's head could have made contact. No injury or contact evidence for this was found in the accident.

Case 86/y. This involved a glancing impact to the side of the case vehicle, deforming the B-pillar in-

Table 7. Results of tests under the conditions of case 86/x.

Child Seat Model	Strap Adjustment: Anchor Straps	Strap Adjustment: Child Harness	Pre-impact position of child seat	Peak chest acceleration (g)	Max,head excursion (mm)
B (2 point)	tight	with standard 1" slack	Flat on test cushion, against backrest. Reclined.	33	412
B	as accident	"	"	40	473
B	tight	as accident	"	60	462
B	as accident	as accident	"	105	609
B	"	"	as above but not reclined.	98	590
A (2 point)	"	"	as above, not reclined (attachment straps slack).	44	600
A	"	"	Tilted so that lower front rail of frame as far back as possible and rear upper rail raised. Attachment straps taut. Not reclined.	75	598
C (4 point)	tight	"	tight against backrest, supported on upper straps.	43	392
C	lower straps as above, upper straps lengthened by 100 mm.	"	resting on test seat cushion. Top straps slack.	55	427

wards and rearwards. These conditions suggest a relatively low deceleration for the struck vehicle. The child was in a 2 point frame seat on the struck side. The fatal injury was caused when the child's head struck the deformed B-pillar. The suggestion was made that the relative forward head movement in 2 point frame seats compared with 4 point seats might be greater in low severity impacts. To assess this, comparative tests were made with 2 and 4 point seats at three impact severities. The results, shown in Table 8 indicate that the changes both in peak chest acceleration and in maximum forward head excursion were similar for the two styles of child seat. Model A in Table 8, which was that in use in the accident, does show the greatest head excursion at both speeds tested.

Discussion

Perhaps the first remark that should be made is to note the small number of fatalities to restrained children in this age group, on average two to three per annum. This is during a period when the restraint use ranged from 17 per cent to 36 per cent. As the average number of child car occupants in this age group killed per annum was 30, this suggests a child restraint efficacy of about 75 per cent for fatalities. Many of the countermeasures being considered or being introduced for the benefits of car occupants generally (front and rear underrun guards for goods vehicles and improved side impact protection) will be of benefit also in reducing fatalities to restrained children.

In view of the prominence of head injuries in these cases, particular attention should be paid to protecting the child's head, both by avoiding excessive forward movement in the restraint and by padding areas of the car likely to be struck by the head in front, side and rear impacts.

As regards the design of child restraints, the results suggest that the absence or non-use of a crotch strap can lead to submarining or ejection in some circumstances.

Incorrect fitment or adjustment was not found to be a particularly frequent cause of fatality but nevertheless was seen to be important in those cases where it was observed. The dynamic tests showed the importance of tight adjustment of attachment straps, especially for the two point attachment frame seats.

Table 8. Affect of impact severity on chest acceleration and head excursion with 4-point seats and 2-point frame seats.

Impact Speed (miles/h)	2 point Frame Seat (A)		2 point Frame Seat (B)		4 point Seat (C)	
	Peak Chest Acc. (g)	Max. head excursion (mm)	Peak Chest Acc. (g)	Max. head excursion (mm)	Peak Chest Acc. (g)	Max. head excursion (mm)
15	12	360	13	223	19	243
20	17	406	21	302	20	267
25	-	-	29	369	36	309

Conclusions

1. In the thirteen years 1972 to 1984, only 30 children aged under 5 years were killed while restrained as car passengers and 374 were killed while unrestrained.
2. The head of the child is the body area which most frequently suffers fatal injuries, almost always through direct contact with the interior of the car or the impacting object.
3. Correct adjustment of the child harness and the attachment straps is important for the performance of child restraints.
4. The absence of a crotch strap in a child seat harness can, under certain circumstances, lead to ejection under the lap strap (submarining).
5. In most cases, remedial action would have to be applied either to the vehicles structure of the occupied car (improved protection from side impacts and rear seat back strength) or the striking vehicle (front and rear underrun guards for goods vehicles).

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Appendix. Summary Table of Accidents involving fatal injuries to restrained children aged 0-4 years, 1972-86

Accident Number	Child Age	Seat Position	Restraint	Impact Details	Probable Cause of Death
72/1	4 years	Rear seat (nearside)	Child 4-point harness	Frontal impact, car-car	Abdominal injuries from lap strap. Restraint fitted to car on the previous day-may have been incorrectly adjusted.
72/2a	2 1/2 years	Rear seat (nearside)	Child 4-point harness	Severe rear impact by coach	Brain injury; probably head impact to rigid member at top of rear seat backrest of estate car.
72/2b	9 months	Rear seat (offside)	Child seat, 4-point	Severe rear impact by coach	Brain injury; probably head impact, through the child seat to the rigid member at the top of the rear seat backrest.
72/3	2 years	Rear seat	Child seat, 4-point	Side impact by another car, followed by overturning, landing upside down on fence with roof torn off.	Severe head injury. Contact of head with deformed roof or with fence.
73/1	2 years	Front seat	Non-approved seat. Adult belt round child.	Car hit rear tail of heavy goods vehicle. Under-run. Car roof crushed down over front seat.	Severe head injury. Child's head probably contacted deformed roof of car or the goods vehicle tail.
74/2	15 months	Rear seat (nearside)	Child seat, 4-point	Struck on nearside by car, overturned and hit brick wall on nearside while inverted.	Multiple injuries, including fractured neck and ruptured spleen. Contact with intruding bodywork, impacting vehicle or brickwall.
77/1	11 months	Rear seat	Non-approved "hook-over" child seat	Frontal impact with a light goods vehicle.	Multiple skull fracture and brain injury. Seat and child flew forwards into front of passenger compartment.
78/1	11 months	Rear seat (offside)	Child seat, 4-point	Severe rear impact by a heavy goods vehicle. Considerable intrusion into rear of passenger compartment followed by fire.	Severe burns. Severe head injury prior to fire likely. One of two unrestrained children in the rear seat also died, the other seriously injured.
79/2	20 months	Rear seat (nearside)	Child seat, 4-point	Severe rear impact by another car.	Cerebral contusion, fracture-dislocation of neck(C7). Probable contact with front seat headrest as backrest of front seat collapsed rearwards.
79/3	4 3/4 years	Front seat	Adult 3-point belt	Frontal impact with another car. Some intrusion of facia and A-pillar.	Brain injury, lacerated spleen. Rear loading from unrestrained rear adult passenger, head contact with A-pillar or facia.
79/4	10 months	Rear seat (centre)	Child seat, 4-point	Stationary car over-ridden and crushed by heavy goods vehicle.	Fracture-dislocation of neck (C3) with spinal column compression from downward intrusion of car roof.

Accident Number	Child Age	Seat Position	Restraint	Impact Details	Probable Cause of Death
80/1	14 months	Rear seat (nearside)	Child seat, 4-point	Front of an agricultural tractor into side of case car causing intrusion and rearward displacement of doors and B-pillar. Angled side impact.	Multiple skull fracture, multiple cerebral contusions and haemorrhages. Contact of head with intruding bodywork.
80/2	19 months	Rear seat (centre)	Child seat, 4-point(?)	Frontal impact of car into a tree.	Fracture dislocation of neck (C1/2). Child found in front passenger foot-well. Harness reported as being undone after the accident. Child may have been ejected from harness or may not have been restrained prior to the accident.
81/1	4 years	Rear seat (offside)	Child 4-point harness	Side swipe from front of another car into the rear half of the offside	Multiple head and torso injuries Ejected. The harness was incorrectly mounted round the folding rear seat backrest. Child probably ejected through open rear door.
81/4	3 years	Rear seat (nearside)	Child harness	Angled impact to nearside of passenger compartment with rear corner of platform of a milk delivery truck	Skull fracture and brain lacerations. Direct impact of head to rear platform of milk truck.
82/1	16 months	Rear seat (nearside)	Child seat, 4-point	Severe side impact to nearside passenger compartment by another car.	Skull fracture and brain laceration. Head contact with intruding side structure and possibly other car.
82/2	6 months	Rear seat (offside)	Child seat, 4-point	Stationary car hit by goods vehicle and pushed under load platform of another goods vehicle. Roof structure highly deformed downwards.	Skull fracture and brain injury. Head contact with intruding roof structure.
82/3	15 months	Rear seat (nearside)	Child seat, 4-point	Severe frontal car-car impact. ($\Delta V \sim$ 35-40 mile/h)	Brain contusion without skull fracture. No evidence of head contact from external injury.
82/5	3 1/2 years	Rear seat	Child seat, 4-point	Car overturned and fell into river.	Drowned. Only minor injuries sustained from intruding roof. (2 other children, unrestrained, also drowned in this car).
82/7	22 months	Rear seat (nearside)	Child seat, 4-point	Side impact to nearside passenger compartment, struck by another car.	Widespread brain haemorrhage without skull fracture. Fracture-dislocation of the neck (C1/2) with spinal cord damage. Head contact with intruding side structure or impacting car.
82/8	16 months	Rear seat (nearside)	Child seat, 4-point	Car rolled down a bank, landing on its roof.	Brain injury (skull fracture with subgaleal, extra- and subdural haemorrhage and cortical laceration). Contact with roof which had crushed as far as the top of the child seat.
83/2	1 year	Rear seat (centre)	Child seat, 4-point	Very extensive crushing between two heavier cars. Rear impact while stationary.	Brain injury (skull fracture with torn dura, subarachnoid and subdural haemorrhage). Lacerated liver.

Accident Number	Child Age	Seat Position	Restraint	Impact Details	Probable Cause of Death
83/3	23 months	Rear seat (centre)	Child seat, 4-point	Angled sideswipe into offside of passenger compartment, by another car.	Multiple skull fracture and severe brain injury. Head contact with intruding side structure. Straps somewhat loose and possible additional loading from unrestrained adult in nearside rear seat.
83/7	2 years	Rear seat (offside)	Child seat, 4-point	1/4 overlap frontal into a medium goods vehicle resulting in direct intrusion into the drivers seating area (offside).	Skull fracture and major brain injury. Contact of head to grossly distorted roof or B-pillar.
83/8	4 years	Rear seat (nearside)	Child seat	Rear impact by coach while stationary in a line of stationary vehicles. Under-run with considerable crush.	Skull fracture and major brain injury. Head contact with intruding rear bodywork and possible coach front. Head of child may have been above the child seat backrest.
83/9	22 months	Rear seat (centre)	Child seat, 4-point	1/4 overlap of car across the front face of a heavy goods vehicle. Under-run, resulting in considerable intrusion along the whole offside of the car and removal of roof. Angled side impact.	Multiple skull fracture and brain injury. Fractured neck with complete separation of C2/3. Direct contact of head with front of goods vehicle or intruding roof.
83/10	16 months	Rear seat (nearside)	Child seat, 4-point	Rear impact by another car followed by roll-over.	Head injury either from impact to hard rail at top of rear seat backrest through child seat shell or with roof structure which collapsed in roll-over.
84/2	4 years	Rear seat (nearside)	Child seat, 4-point	Side impact into nearside passenger compartment by another car.	Brain injury. Head contact with intruding side structure of car.
84/3	16 months	Rear seat (centre)	Child seat, 2-point (no crotch strap)	Frontal impact with another car (ΔV approx. 20-25 mile/h)	Neck injury (Fracture-dislocation of C1/2). Likely cause - submarining resulting in lap strap applying load under chin.
84/7	10 months	Rear seat (offside)	Child seat, 2-point (with non-standard top-tether added)	Frontal impact with tree.	Neck injury (complete dislocation between C3 and C4 and evidence of heavy loading under chin and at front of neck). Cause not clear but possibly submarining, despite presence of crotch strap.
85/3 *	17 months	Rear seat (centre)	Child seat, 4-point (incorrectly mounted and crotch strap may not have been used)	Severe frontal impact to side of heavy goods vehicle followed by a spin (clockwise).	Neck injury (dissociation of atlanto-axial joint, extra dural haemorrhage and contusion to spinal cord.) Lung contusion and lacerated liver. Top attachment straps fitted to same anchor point as lower straps allowing excessive forward movement when car seat backrest failed, uneven loading of child restraint straps caused lap strap to fail and child was ejected from restraint. Photographs at the scene suggest crotch strap was not fastened prior to impact.

Accident Number	Child Age	Seat Position	Restraint	Impact Details	Probable Cause of Death
86/x *	19 months	Rear seat (nearside)	Child seat, 2-point	Frontal impact to the side of another car, followed by anti-clockwise rotation.	Neck injury (complete separation of C1 and C2 and macerated spinal cord). Cause unclear, but seat attachment straps were somewhat loose and crotch and lap straps were at maximum length.
86/y *	2 years	Rear seat (offside)	Child seat, 2-point	Sideswipe to offside of passenger compartment by a light goods vehicle resulting in inward and rearward intrusion of B-pillar and body panels.	Fatal head injury due to direct contact of child's head with deformed B-pillar. Car rear seat backrest failed.

*Note. Not all accidents in 1985 and 1986 recorded here.

Accident Data Analysis and Accident Prevention Measures—Especially for Two-Wheeled Vehicles

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Abstract

Due to greater traffic density and congestion, traffic accident in Japan has been increasing in past five years. We can mention three main factors in this increasing trend by analyzing accident data in Japan. These factors are:

1) Motor vehicle occupant deaths have been increased slightly and continue to make a large proportion of the number of total deaths.
2) The elders deaths have been increasing continuously.
3) Moped riders and motorcyclists deaths have been increased dramatically, which is the biggest factor of the above trend.

Therefore, strong countermeasures are needed such as follows:

1) Promotion of seat belt use including compulsory measures,
2) Systematic traffic safety education for elderly people,
3) Comprehensive traffic safety measures for mopeds and motorcycles.

Especially for safety measures for mopeds and motorcycles, we investigate comprehensively and make some proposals.

Trends of Traffic Accidents in Japan

The number of casualties from traffic accidents in Japan had increased from year to year, reaching almost 1 million in 1970. Then, the Government promoted comprehensive traffic safety policies based on the Fundamental Traffic Safety Program, the number of deaths were reduced nearly by half in 1974, the number of injuries to 60% in 1972.

However, the trend has reversed after the year, and the number of deaths has exceeded 9,000 for past five years, and the number of injuries to 700 thousand in last year (Fig. 1).

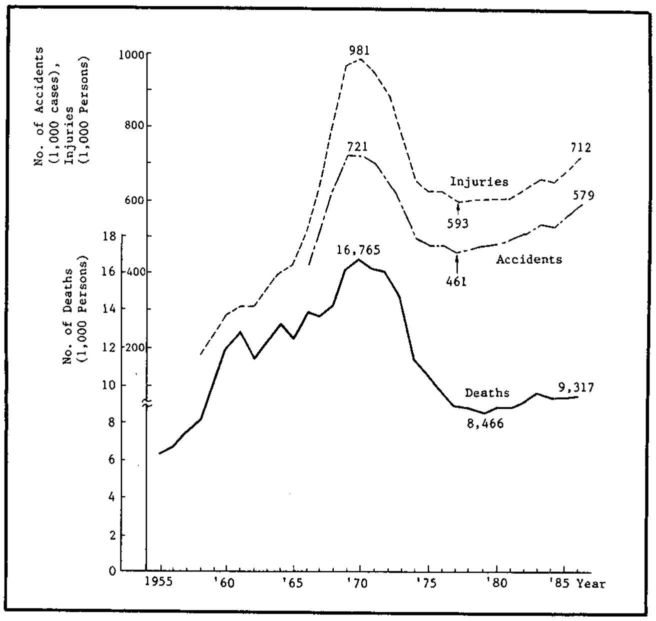

Figure 1. Trends of traffic accidents

The main factors in trends of traffic accidents are quantitative expansion of road traffic, such as increase in ownership and vehicle-kilometers, the qualitative changes such as diversification of vehicles and drivers, and the progress of traffic safety measures.

Between 1955 to 1965, the income level of people in Japan had risen by the economic expansion, and the number of motorvehicles increased annually by 20% on average. On the other hand, accident rate (the number of deaths per motorvehicle) kept declining annually by 10% on average, that was far below the rate of annual increase in ownership, in other words, traffic safety measures were insufficient in comparison with rapid expansion of road traffic, and thus the number of deaths increased.

In contrast, between 1965 and 1975, annual rate of economic growth kept at low level, the annual rate of increase in ownership went below 10% in average, and that in the vehicle-kilometers made a drastic decline, especially in 1974, from the effect of Oil Shock. In comparison, the accident rate made an annual decline of 15% on average, far greater than that of annual increase in ownership, and the number of deaths made a turn to a dramatic decline. This decline were affected by comprehensive traffic safety policies promoted in this period.

Since 1975, the rate of increase in motorvehicle ownership kept at the level lower than 5% according to the slow economic growth, the decline in the accident rate kept at about 3%, and thus the number of deaths has been increasing again.

There are various factors in movement in the accident rate. Diversified qualitative changes, and greater traffic density and congestion can be such causes. While quantitative expansion of traffic continues to progress, the number of two-wheeled vehicles has reached nearly 30% of total vehicles, and the elderly drivers rapidly increased due to the expanding population of this age group, thus enhancing the diversification of vehicles and drivers. On the other hand, various traffic safety measures have not necessarily produced satisfactory effect because of the less degree of accident preventative measures taken, as a result of depressed public investment (Fig. 2, Fig. 3).

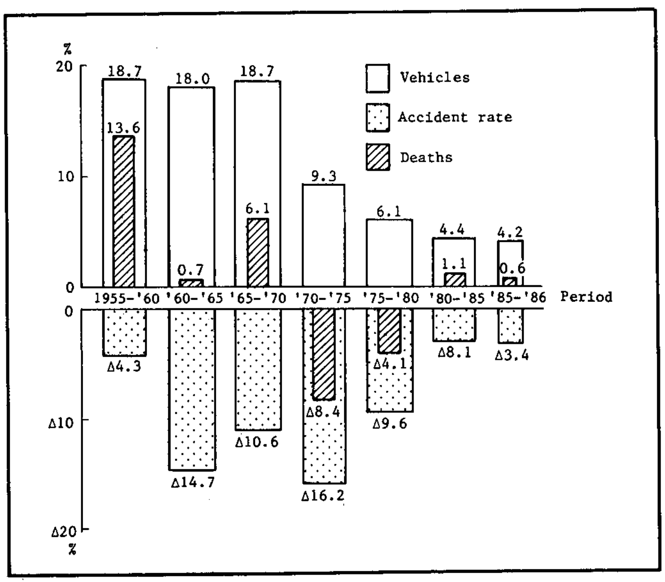

Figure 2. Changes in annual increase or decrease rate in deaths, etc.

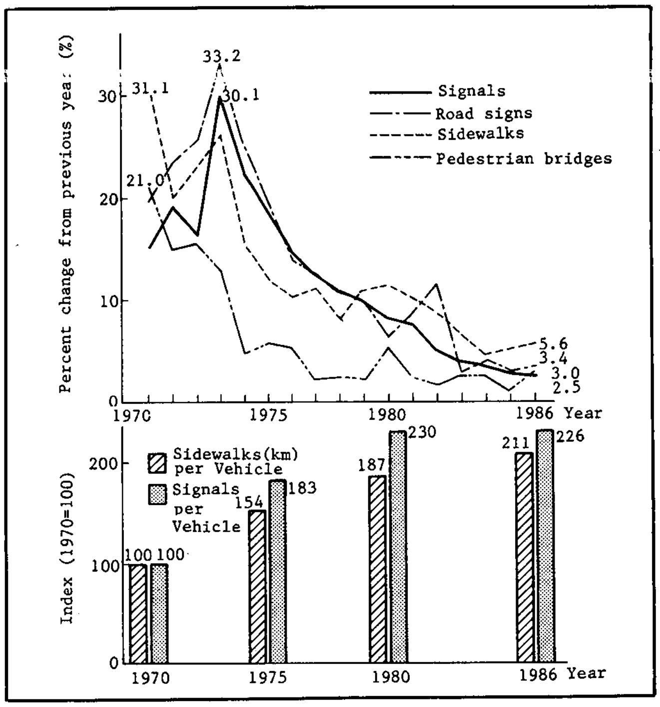

Figure 3. Changes in traffic safety provisions

Special Characteristics of Recent Traffic Accidents in Japan and the Future Problems of Traffic Safety Measures

Social and economic losses of traffic accidents in Japan are estimated to nearly 10% of the national government budget (excluding national government bonds), or 4 trillion yen per year, equivalent to almost 1% of GNP in Japan.

Corresponding to the quantitative expansion and qualitative changes in road traffic, following three characteristics are found in the recent traffic accidents (Fig. 4, Fig. 5).

1) Many occupant casualties,
2) Rapid increase in traffic accidents involving elderly people,
3) Rapid increase in traffic accidents involving two-wheeled vehicles,

Promotion of Full Use of Seat Belts

The number of occupant deaths has been kept at high level, making 1/3 of the total.

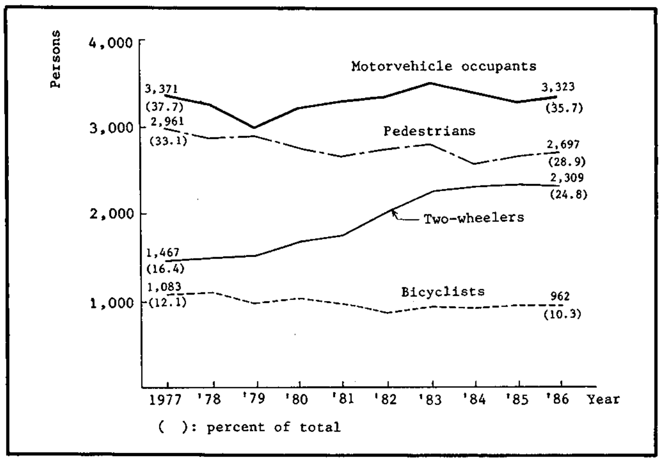

Figure 4. Changes in traffic accident deaths by accident type

In order to reduce the occupant casualties, seat belts are very effective at present. From our estimate, the total deaths will exceed 10,600 in 1990, providing that the quantitative and qualitative elements follow roughly past. However, with the promotion for full use of seat belts, the number is estimated to decline by 2,000 when the rate of seat belt use are risen up to 90%, (Fig. 6).

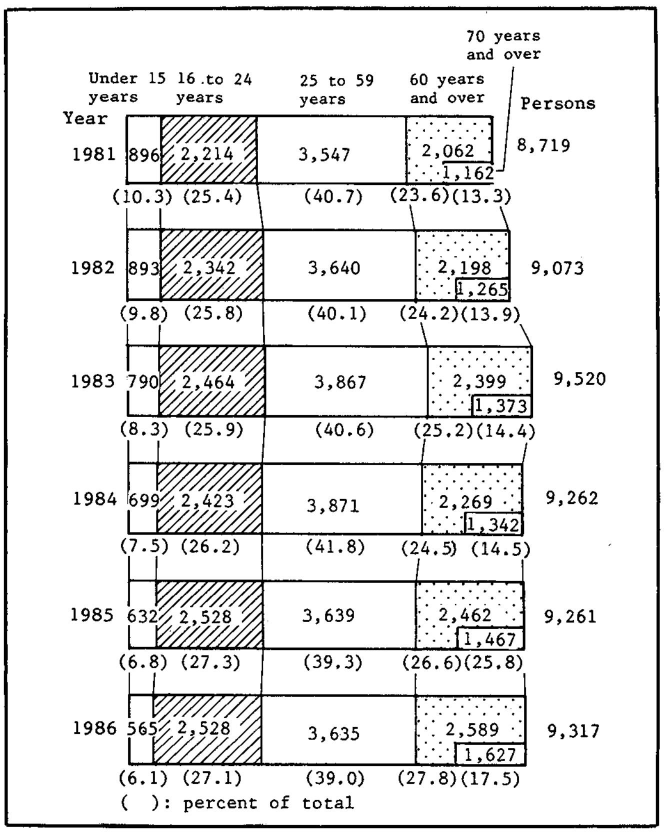

Figure 5. Changes in traffic accident deaths by age group

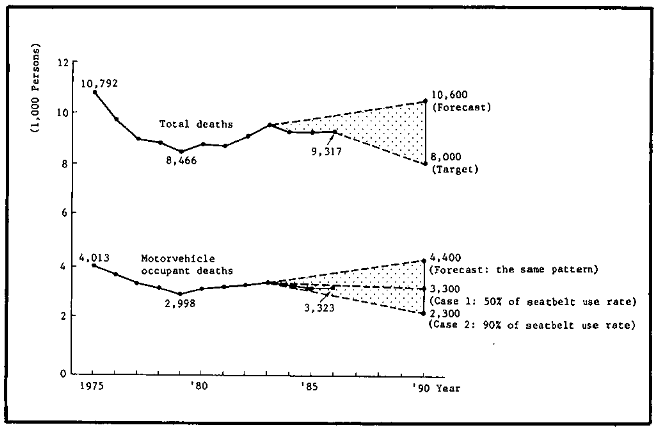

Figure 6. Forecast of decrease in deaths by the promotion of seat belt use

By analyzing the number of occupant casualties in 1986, the death rate (the number of deaths/the number of casualties) was 0.43% (424/97,824) for seat belt users; and 1.11% (2,857/258,227) for non-users, proving the effectiveness of seat belt with the 2.6 times better survival chance

In Japan, seat belt use has become compulsory, and in one point administrative measure has been charged for the violation since November, 1986, and the rate of seat belt use improved to 95%. The number of occupant deaths, after this month, made a down turn, from the previous years' up-trend (Fig. 7).

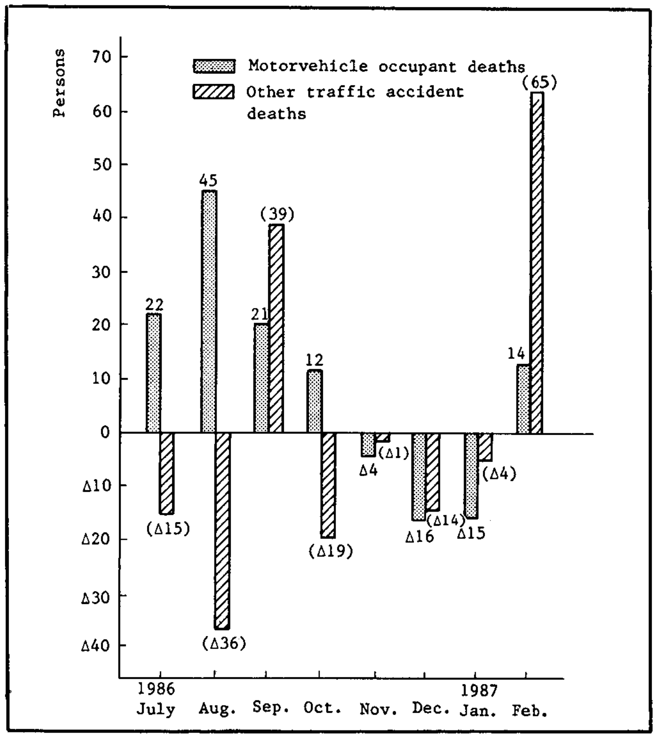

Figure 7. Changes in the number of occupant deaths from the same month in the previous year.

Our society will become more and more automobile oriented, and the number of occupant deaths will increase, considering that the rate of the number of people per vehicle is 1.4 in the U.S. (2.7 in Japan), where the number of occupant deaths accounts for 70% of total deaths. Therefore, active campaign for the proper use of seat belts, including backseat belts and safety seat for children, should be promoted.

Promotion of Traffic Accident Prevention Measures for the Elderly

As the population of the elderly (60 years and over) becomes larger, the number of traffic accident deaths of the elderly has increased by 25% during the past five years. This accounts for nearly 30% of the total traffic deaths, and more than 50% of pedestrian and bicyclists deaths. Recently, the number of traffic deaths of those 70 years and over is drastically increasing. Also, the rapid increase in the number of drivers deaths is noticed as the number of elderly drivers increases (Fig. 5).

Elderly people bear much higher risk for traffic accidents, because of their deterioration of physical functions, and the death rate in such accident is also extremely high, with people of 70 years and over at the highest (Fig. 8).

The number of pedestrian deaths in 1986, having the elderly as its highest risk group, involved 1,057 deaths while crossing the street, including 220 while walking on pedestrian's crossing, accounting for almost 3/4 of 1,399 total traffic deaths.

For fatal accidents in which elderly drivers were the first party, in 1986, collisions upon meeting at intersections occured in 539 cases (23.0%), followed by head on collisions. Compared to other age groups, there are fewer accidents with pedestrians, whereas there are larger number of collisions upon meeting at intersections.

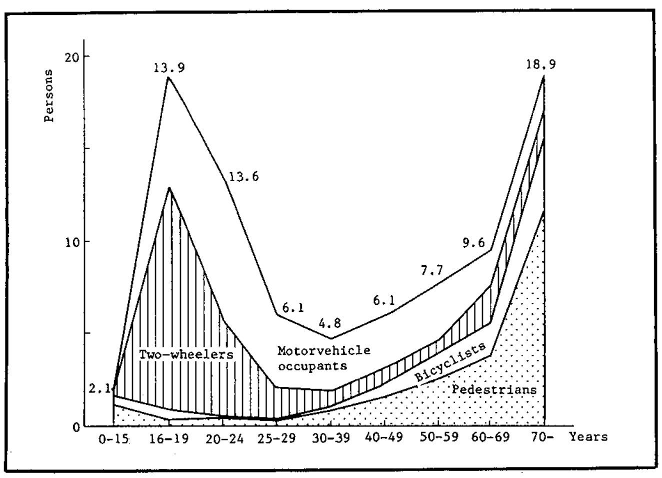

Figure 8. Traffic accident deaths per 100,000 persons by age group and accident type

In future, considering the dramatic increase in population of 70 years and over and the expansion of places for various activities participated by elderly people, traffic accident picture in Japan is expected to look similar to some European countries where the population of elderly people is much larger. Incidentally, in Sweden where the age group of 65 years and over accounts for 17.5% of the total population, far larger compared to our 10.3%, the number of traffic accident deaths, in this age group, accounts for 27.7% of the total traffic accident deaths, far higher than 21.1% in Japan. There are presently some measures being executed for Traffic Safety Education for the elderly. However, it is now very necessary to consider a countermeasure including measures for elderly drivers.

Preventive Measures for Accidents With Two-Wheeled Vehicles—Especially for the Younger Generation

The number of deaths of two-wheelers (moped and motorcycle riders) has been increasing every year, making 60% leap over the past ten years, accounting for nearly 1/4 of the total number of traffic accident deaths. The number of deaths in the age group between 16 and 24 years old was particularly large, and it accounted for nearly 60% of the total number of two-wheelers deaths. For motorcycles, it accounted for 3/4 of the total (Fig. 4).

Also, the number of traffic accident deaths accounts for almost a half of the total accident deaths. That of the youth group comes in the first place among all causes for death, including sickness. Particularly, two-wheelers deaths account for half of the total number of traffic accident deaths.

Looking at the types of fatal accidents where two-wheeled vehicles are the first party, single vehicle accidents, such as colliding into the structure or over-turning, accounts for 45.1%, far larger compared to motor vehicles at 20.0% in Japan.

Also, looking at the traffic violations, maximum speed violations account for the largest portion of 38.0%, compared to 22.5% in motorvehicles. 54.3% of traffic violations in motorcycles involve maximum speed violations. By contrast, in case of mopeds, major violations are reckless riding, and not stopping at inter-sections.

Two-wheeled vehicles are in some way difficult to operate, and the types of accidents such as overturning attributable to such special characteristics are so many. Because two-wheeled vehicles may easily overlooked by others, and two-wheelers are hardly protected physically, accidents would often invite very serious results. In 1986, the number of deaths per 10,000 vehicles was 3.5 persons, concentrating particu-

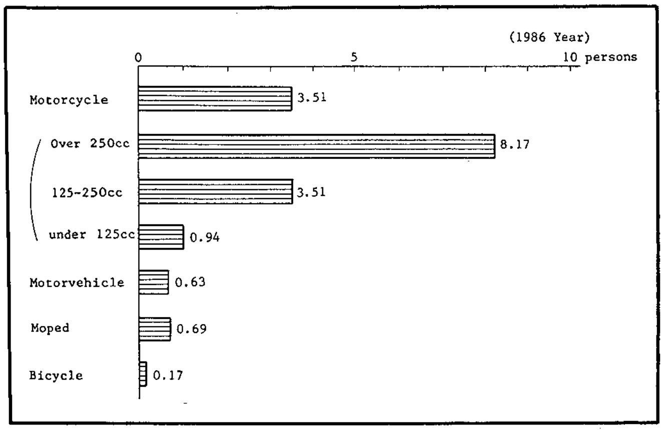

Figure 9. Comparison in deaths per 10,000 vehicles

larly in the category of motorcycles with engine capacity more than 250 cc (Fig. 9).

National government, local public bodies, and private traffic safety organizations promote following safety measures for two-wheeled vehicles.

1) Improvement of Road Traffic Safety Facilities

In addition to improvement of safety facilities such as modifications of traffic crossings, the provision of two-step stopping lines, the promotion of two-step right-turns for mopeds, and the execution of traffic regulations to separate the paths for two-wheeled vehicles and motor vehicles, are being taken.

Also, the control of illegal motorvehicle parking of highly dangerous and annoying nature is being administered.

2) Traffic Safety Education and Publicity Activities

Greater effort has been put in to fulfill the need for better education for two-wheelers, riding school curriculums are now programmed to contain such practices and study subjects which place emphasis on traffic safety awareness and on enhancing the ability to deal with dangers and trouble. Besides, there are seminars held for new license aquisitors, such as Safety Skill Training for moped riders, and Safe Riding Seminars for new motorcyclists. There are also such safety education programs as Seminar for two-wheelers at the time of license renewal, and Safe Riding Education for inexperienced riders.

On the other hand, Family Meetings for Traffic Safety are suggested. Also, especially at high school, education programs to promote better understanding on special characteristics of two-wheeled vehicles and ways to prevent traffic accidents, particularly through extra-curricular home-room sessions and school events, are being executed.

Also, prevention of two-wheeled vehicle accidents are taken up as primary items of emphasis during the National Traffic Safety Campaign, and guidance is given through on-the-street campaign and home visits made by private organizations.

3) Support for Good Traffic Order

Prohibition of taking rear seat occupants for inexperienced motorcyclists, compulsory two-step right turns for mopeds, compulsory helmet wearing for mopeds are put into effect.

Furthermore, controls over such violations of exceeding speed limit, driving without a license, and ignoring traffic lights are taken especially for young riders.

Generally, comprehensive policies should be promoted in a concerted effort by national government, local public bodies, and private traffic safety organizations.

For this reason, Accident Preventive Measures Promotion Council for Two-Wheeled Vehicles (organized by members of the concerned ministries and agencies) was established in 1986. In order to assist their work, some research and model projects are now being promoted. In this case, the important subjects are to make young motorcyclists aware of social responsibilities and to foster their attitude for safe riding.

So, we must promote traffic safety education at home and school from early age, at work places, and throughout local communities, for safety awareness among people.

Crash Protection Offered by Safety Belts

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Abstract

This paper analyzes the national records of highway accident experience, for the purpose of evaluating the protection offered by safety belts to car occupants, as a function of seating position, belt type and casualty type. In response to persistent concerns, this investigation emphasizes the need for and the application of controls, necessary to deal with confounding effects and with other limitations inherent in large scale accident data bases. The entire inventory, all eleven years 1975 to 1985, of the Fatal Accident Reporting System (FARS) is the data source for fatal car accidents. Data for all crash severities, whether fatal or not, are supplied by the National Accident Sampling System (NASS), 1979 to 1985. Three methods are used to analyze the data: "Explicit Control for Crash Severity", "Subject/Control Occupant Pairs", and "Overall Evaluation of Casualty Reduction". The analysis of the NASS data produces familiar results for safety belt effectiveness, i.e. values scattered around a nominal 50%. A parametric analysis is conducted to determine the sensitivity of safety belt effectiveness estimates to inaccuracies in reporting restraint system status. The FARS data are analyzed by methods that utilize "Subject/Control Occupant Pairs", exposed to similar crash conditions. Results from these analyses indicate the following fatality prevention effectiveness: (a) about 50% for the driver restrained by a Lap & Shoulder belt; (b) about 40% for the front seat outboard passenger restrained also by a Lap & Shoulder belt; (c) a decline of each of the above by about 20 percentage points, when the Lap and Shoulder belt is replaced by a Lap Only belt; and (d) an effectiveness in the range 15% to 25%, with a rather large error, for rear seat passengers, who are restrained exclusively by Lap belts. A similar method is applied for the determination of the fatality prevention effectiveness of ejection avoidance. It is found that this effectiveness ranges between 70% and 80%, irrespective of seating position. This is very important given that there are between 6,500 and 8,000 people per year, roughly 2/3 of them car occupants and 1/3 occupants of light trucks and vans, that are killed ejectees. This is the population that may benefit from the high effectiveness of ejection avoidance.

Introduction

Car occupant restraint use and effectiveness, as a function of occupant, vehicle, and crash attributes, have been issues in the United States for over twenty years. Recently however, with the onset of national campaigns for increased seat belt use, and especially with the onset of mandatory seat belt use for front seat occupants, increased attention is being paid to the performance and use rate of safety belts as a function of occupant's seating position and belt type.

The purpose of the work reported here is to perform an analysis of the up to date large scale national data, concerning on the road accident experience of restrained car occupants, and to evaluate this experience in comparison to that of unrestrained occupants.

In response to persistent concerns, this investigation emphasizes the need for and the application of controls, necessary to deal with confounding effects and with other limitations inherent in large scale accident data bases.

The entire inventory, all eleven years 1975 to 1985, of the Fatal Accident Reporting System (FARS), reference 1, is the data source for fatal car accidents. Data for all crash severities, whether fatal or not, are supplied by the National Accident Sampling System (NASS), 1979 to 1985, reference 2.

Methods of Analysis

Two methods are used to analyze the data: "Explicit Control for Crash Severity", and "Subject/Control Occupant Pairs". In order to deal with differences in the crash severity distributions, between restrained and unrestrained occupants, explicit control for crash severity is applied to the injury rates calculated from the NASS data. Specifically, injury rates for unrestrained and for restrained occupants are calculated in each and every interval of crash severity.

These, crash severity specific, rates are calculated as the count of injured divided by the count of all involved, under identical conditions. At the same time, a common frequency distribution of all (restrained and unrestrained) occupants is determined from the basic data, over the crash conditions of interest.

The crash severity specific rates, each weighted with its appropriate frequency from the common distribution, are then combined to yield an overall but controlled injury rate, first for the unrestrained occupants and then for the restrained.

The relative difference of these two controlled rates is a measure of the restraint effectiveness, free as much as possible of the confounding effects due to the differences in crash conditions between restrained and unrestrained occupants.

In certain accident files, for example in the FARS, it is difficult, if not impossible, to determine casualty rates for restrained and unrestrained car occupants, under comparable crash conditions. This is due to the absence of information for the characterization of these conditions.

One way to circumvent this difficulty is the application of a "Subject/Control Occupant Pair" method, a method that is gaining increasing acceptance in analyses of this type conducted recently, see references 3, 4, and 5.

In this method the rate of casualties for one category, (subject category), of car occupants, say unrestrained drivers, is measured by reference to the casualties of another category, (control category), say unrestrained front seat passengers. This is done with the additional provision that both categories are exposed to identical or very similar crash conditions. Such a rate may be thought of as the relative odds of casualty and is expressed, in this example, as unrestrained driver fatalities per unrestrained passenger fatality.

A similar rate may be determined for restrained drivers, i.e. restrained driver fatalities per unrestrained passenger fatality. Note that the denominator in both instances is unrestrained passenger fatalities, paired with unrestrained driver fatalities in the first instance, while paired with restrained driver fatalities in the second instance.

The denominator commonality acts as the normalizing factor that controls against variability in the crash conditions and ensures a fairly common reference for the casualties appearing in the numerators. Commonality in crash conditions is obtained by the fact that both the subject and the control occupant categories, whether restrained or unrestrained, are selected as pairs, each pair involved in the same crash, thus each pair experiencing the same or very similar crash conditions.

In the two methods discussed above, we determine and analyze casualty rates for specific car occupant populations, first unrestrained and then restrained. Subsequently we apply these results in order to estimate safety belt effectiveness.

Results from the NASS Concerning Safety Belt Effectiveness

The NASS data were analyzed through an explicit control for crash severity, similar to that used in reference 6. For this purpose injury rates, (injured occupants per involved occupant), are calculated for each available crash severity control interval. These, crash severity specific, injury rates are then combined into an overall injury rate, controlled for crash severity, as explained earlier. The results are summarized in Table 1.

The data were treated for three car occupant seating positions: the driver, the outboard passenger in the front seat, and the rear seat passengers. Consequently two types of safety belts were considered: lap and shoulder for the front seat occupants, and lap only for the rear seat occupants.

Four different casualty thresholds were addressed: MAIS 3+, i.e. occupants with maximum injury severity of AIS 3 or higher, irrespective of outcome; MAIS 2+; Hospitalization; and Transport to a Hospital for treatment, irrespective of subsequent hospitalization. Because of sparsity, fatalities were not addressed individually.

The results in Table 1 are familiar. The shown values for the safety belt effectiveness are scattered around a nominal value of 50%, as has been known from past analyses of these NASS data, or the data from the National Crash Severity Study (NCSS), or the data from State accident records.

A pattern worth noting is that the effectiveness for the front passenger seat is about five to ten percentage points lower than the corresponding effectiveness for the driver. This pattern will be observed in many more instances, to be considered later, and is well beyond statistical errors.

The scatter observed in the effectiveness values concerning various casualty thresholds for the rear seat occupants is consistent with the larger statistical errors corresponding to these determinations, as a result of sample sparsity associated with rear seat restrained occupants.

Effects of Inaccuracies in the Reporting of Restraint Status

The purpose of this section is to develop quantitative estimates of injury rates, for restrained and unrestrained car occupants, and of belt effectiveness,

Table 1. Summary of results from determinations of seat belt effectiveness for car occupants, as a function of seat belt type, seating position, and casualty threshold, according to NASS data.

Seat Belt Type	To Prevent	Seat Belt Effectiveness % — Front Seat — Driver	Front Seat — Passngr	Rear Seat Passngrs
Lap & Shldr	MAIS 3+	58.2	51.0	----
Lap	MAIS 3+	----	----	74.4
Lap & Shldr	MAIS 2+	54.9	40.8	----
Lap	MAIS 2+	----	----	27.1
Lap & Shldr	Hospitlztn	57.2	48.1	----
Lap	Hospitlztn	----	----	62.5
Lap & Shldr	Transport	39.0	39.2	----
Lap	Transport	----	----	35.1

as a function of parametrically treated over- or under-reported restraint status.

Let Ar and Cr represent the all involved and injured, respectively, car occupant populations that are accurately characterized as restrained in crashes. Let Au and Cu be the corresponding accurate characterizations for the unrestrained. It follows that the accurate injury rates are given by:

$$Rr = Cr / Ar \tag{1}$$

$$Ru = Cu / Au \tag{2}$$

Furthermore, it follows that the accurately characterized values of safety belt use rate and effectiveness are given by:

$$u = Ar / (Ar + Au) \tag{3}$$

$$e = (Ru - Rr) / Ru \tag{4}$$

Suppose now that Br and Dr are inaccurate counts of restrained occupants, all involved and injured, i.e. the inaccurate counterparts of Ar and Cr, respectively. Similarly suppose that Bu and Du are the inaccurate counterparts of Au and Cu. The injury rates, the belt use rate and the effectiveness, based on these inaccurate counts, are given by relations that parallel (1) to (4) as follows:

$$Qr = Dr / Br \tag{5}$$

$$Qu = Du / Bu \tag{6}$$

$$v = Br / (Br + Bu) \tag{7}$$

$$f = (Qu - Qr) / Qu \tag{8}$$

Two important parameters shall be used to describe the restrained occupant count inaccuracies:

$$m = Br / Ar \tag{9}$$

$$n = Dr / Cr \tag{10}$$

Values for these parameters are higher than one for over- and lower than one for under-estimates of restrained occupant counts.

The objective of this section is to arrive at expressions of the affected belt use rate and effectiveness, v and f given by (7) and (8) respectively, as a function of the accurate values, u and e given by (3) and (4), and the two parameters, m and n, of count inaccuracies given by (9) and (10).

In order to accomplish this we need analytical expressions for Bu and Du, similar to those shown in (9) and (10). These expressions may be derived with the help of the following two constraints:

$$Br + Bu = Ar + Au \tag{11}$$

$$Dr + Du = Cr + Cu \tag{12}$$

Relation (11) states that the sum of restrained and unrestrained occupants that have been injured remains the same, irrespective of the accuracy of the distribution between restrained and unrestrained. Similarly, relation (12) states that the count of all occupants involved in car crashes is the same, irrespective of accuracy in the distribution between restrained and unrestrained.

A combination of relations (9) and (11) yields relation (13), while when (10) and (12) are combined, relation (14) is obtained, as shown below:

$$Bu = Au - (m - 1) * Ar \tag{13}$$

$$Du = Cu - (n - 1) * Cr \tag{14}$$

Relation (7), with use of (9) and (11), is transformed into:

$$v = m * u \tag{15}$$

This relation is one of the objectives of this section as it expresses the safety belt use rate, the apparent rate based on inaccurate counts, in terms of the accurate rate u and the inaccuracy parameter m. Relation (15) is also intuitively evident.

With similar transformations we arrive at the last objective of this section, i.e. at an expression for the apparent belt effectiveness, f, based on inaccurate counts, as a function of the accurate values of belt use rate and effectiveness, u and e, and of the inaccuracy parameters m and n:

$$f = 1 - (1 - e) * F \tag{16}$$

$$\text{where } F = (n/m)*[(1-u)-(n-1)]/[(1-u)-(1-e)*(n-1)] \tag{17}$$

Relations (15) and (16) are easily evaluated numerically. Results from such evaluations are summarized in Table 2, for four combinations of assumed true values of seat belt use rate and effectiveness.

This table shows the apparent values v and f of belt use rate and effectiveness as a function of combinations of the inaccuracy parameters m and n. It is evident that the apparent effectiveness is much more sensitive to inaccuracies than the belt use rate. This sensitivity is also illustrated in Figures 1 and 2.

Both figures are drawn for an assumed true effectiveness of 50% and an assumed true belt use rate of 20%. In Figure 1, the apparent effectiveness is plotted versus inaccuracy parameter m, which is given values between .9 and 1.5, i.e. from 10% under-reporting to 50% over-reporting of restrained occupants, irrespective of injury.

In this figure, inaccuracy parameter n is kept constant at a value of one, implying no inaccuracy in the reporting of restrained injured occupants. In Figure 2, the independent variable is inaccuracy parater n, which assumes values between 0.7 and 1.1, i.e. from 30% under-reporting to 10% over-reporting of restrained injured occupants. Here, parameter m is

Table 2. Apparent values of safety belt use rate, v in %, and effectiveness, f in %, resulting from shown true values, u and e, as a function of parameters m and n, representing inaccuracies of restrained occupant counts.

		u = 20 % e = 50 %		u = 20 % e = 40 %		u = 10 % e = 50 %		u = 10 % e = 40 %	
m	n	v	f	v	f	v	f	v	f
0.9	0.7	18	63	18	57	9	63	9	57
0.9	0.8	18	56	18	48	9	56	9	48
0.9	0.9	18	47	18	37	9	47	9	38
0.9	1.0	18	38	18	25	9	38	9	26
0.9	1.1	18	27	18	11	9	28	9	13
1.0	0.7	20	71	20	66	10	70	10	65
1.0	0.8	20	64	20	58	10	64	10	58
1.0	0.9	20	58	20	50	10	57	10	49
1.0	1.0	20	50	20	40	10	50	10	40
1.0	1.1	20	41	20	29	10	42	10	29
1.1	0.7	22	77	22	73	11	76	11	72
1.1	0.8	22	72	22	67	11	71	11	66
1.1	0.9	22	66	22	60	11	66	11	59
1.1	1.0	22	60	22	52	11	60	11	52
1.1	1.1	22	53	22	43	11	53	11	43
1.2	0.7	24	82	24	79	12	81	12	77
1.2	0.8	24	78	24	74	12	77	12	73
1.2	0.9	24	74	24	69	12	72	12	67
1.2	1.0	24	69	24	63	12	68	12	61
1.2	1.1	24	63	24	55	12	62	12	54
1.3	0.7	26	86	26	84	13	85	13	82
1.3	0.8	26	83	26	80	13	82	13	78
1.3	0.9	26	80	26	76	13	78	13	74
1.3	1.0	26	76	26	71	13	74	13	69
1.3	1.1	26	72	26	66	13	70	13	64
1.4	0.7	28	89	28	86	14	88	14	86
1.4	0.8	28	87	28	85	14	86	14	83
1.4	0.9	28	85	28	82	14	83	14	80
1.4	1.0	28	82	28	79	14	80	14	76
1.4	1.1	28	79	28	75	14	77	14	72
1.5	0.7	30	93	30	91	15	91	15	90
1.5	0.8	30	91	30	90	15	89	15	87
1.5	0.9	30	89	30	87	15	87	15	85
1.5	1.0	30	88	30	85	15	85	15	82
1.5	1.1	30	85	30	82	15	83	15	79

kept constant at a value of one, implying no inaccuracy for the count of all restrained occupants, irrespective of injury.

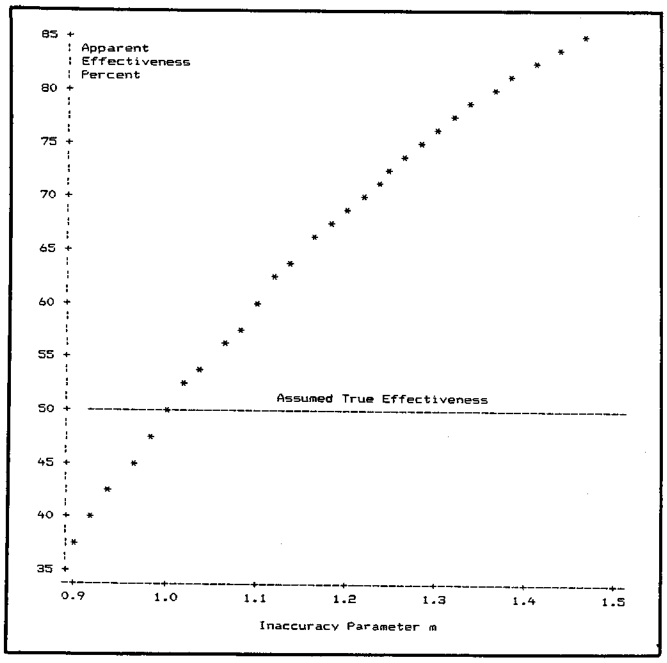

Figure 1. Apparent effectiveness versus inaccuracy parameter m, at constant n = 1.0, corresponding to an assumed true effectiveness of 50% and a belt use rate of 20%.

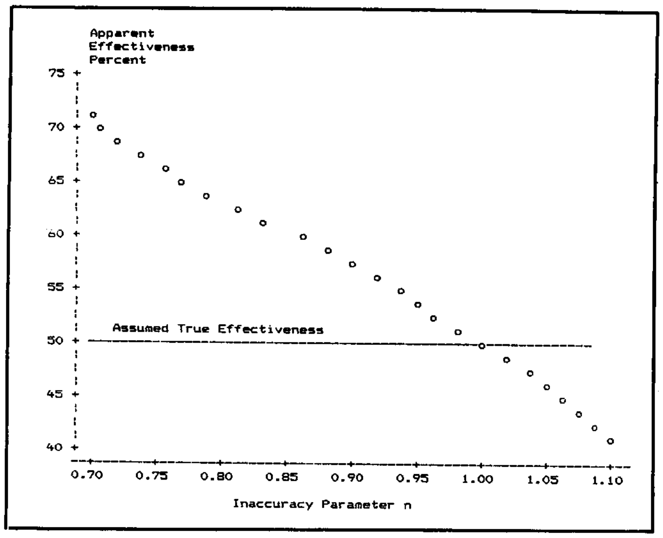

Figure 2. Apparent effectiveness versus inaccuracy parameter n, at constant m = 1.0, corresponding to an assumed true effectiveness of 50% and a belt use rate of 20%.

Figures 1 and 2 illustrate the sensitivity of the apparent belt effectiveness to errors in the reporting of belt use. This sensitivity is one reason that belt effectiveness estimates may vary between accident files, and from year to year for a given file. The figures provide a basis for bounding the errors in effectiveness caused by belt use reporting errors.

Measures of Safety Belt Effectiveness from the FARS Data

In the "Subject/Control Occupant Pair" method, the relative odds are determined for one category of fatalities, (subjects), with respect to another category of fatalities, (controls). This leaves out of consideration the large majority of all occupants, irrespective of injury, which is also the primary source of bias in the characterization of restraint status.

We proceed to apply the mentioned method to the entire inventory of the FARS data. In such applications the FARS is filtered so that the cases that are retained are cars where a pair of subject/control occupants, say a driver and a front outboard passenger, are involved in the same car and crash, with at least one of the pair members being a fatality.

Cars where both members of the said pair survive are eliminated from further consideration. In order to minimize any questions raised by child restraint classification, all the car occupants under consideration are over 5 years old.

Two types of safety belts are distinguished: Lap & Shoulder and Lap Only. The distinction is accomplished through the following filters. All front seats of cars of model years post-73 are assigned a Lap & Shoulder belt. Front seats of earlier model years and all rear seats, irrespective of model year, are assigned a Lap Only belt.

From the occupant restraint point of view, the cars in the processed file fall into four categories:

UU cars, i.e. cars where both the driver and passenger are unrestrained;
Ur cars, i.e. cars with an unrestrained driver and a restrained passenger;
RU cars, i.e. cars with a restrained driver and an unrestrained passenger; and
RR cars, i.e. cars with both driver and passenger restrained.

Note that the order in the UU, UR, RU, and RR designations is always: Driver, Passenger. In each car of each of the four subfleets mentioned above, there is at least one fatality. Next, fatality counts are taken as follows:

DDU: the driver fatality count in UU cars;
PUU: the passenger fatality count in UU cars;
DRU: the driver fatality count in RU cars; and
PRU: the passenger fatality count in RU cars.

From these basic counts the relative odds of fatality, or relative fatality rates may be calculated by:

$$DUPU = DUU / PUU \tag{18}$$

$$DRPU = DRU / PRU \tag{19}$$

Where the notation: DUPU stands for Driver Unrestrained, Passenger Unrestrained; and DRPU for Driver Restrained, Passenger Unrestrained. Note that in both instances the unrestrained passenger fatality counts are the control counts.

The relative difference of the odds or rates shown above, between unrestrained and restrained drivers, is a measure of the restraint effectiveness, EDPU, in preventing driver fatalities, using unrestrained passenger fatalities as the control population, i.e.

$$EDPU = (DUPU - DRPU) / DUPU \tag{20}$$

Paired counts of fatalities (DUU, PUU) and (DRU, PRU) are shown in Table 3, as a function of FARS year 1975 to 1985. This table also gives the fatality odds, DUPU and DRPU, determined according to relations (18) and (19), as well as the safety belt effectiveness, EDPU, resulting from relation (20).

The subject occupant for this entire table is the driver, unrestrained and restrained; the unrestrained passenger is the control in both cases; the belt type is Lap and Shoulder. The bottom row in this table gives counts for all years combined. It also gives weighted mean values of fatality odds and of the resulting effectiveness.

Table 3. Driver/passenger paired counts, driver fatality odds, and safety belt effectiveness for drivers, based on FARS data. Lap & shoulder belt; control: unrestrained front passenger.

FARS Year	Paired Counts DUU	PUU	DRU	PRU	Fatality Odds DUPU	DRPU	Eff. % EDPU
75	459	517	17	35	0.89	0.49	45.3
76	764	785	18	35	0.97	0.51	47.2
77	1155	1191	17	35	0.97	0.49	49.9
78	1723	1742	37	46	0.99	0.80	18.7
79	2116	2106	29	55	1.00	0.53	47.5
80	2397	2442	30	52	0.98	0.58	41.2
81	2695	2679	33	68	1.01	0.49	51.8
82	2554	2565	28	61	1.00	0.46	53.9
83	2394	2500	38	68	0.96	0.56	41.6
84	2605	2624	51	111	0.99	0.46	53.7
85	2621	2580	82	198	1.02	0.41	59.2
All	21483	21731	380	764	0.99	0.50	49.7

The weighted mean value of the Lap & Shoulder belt effectiveness for the driver is 49.7%. A standard error of 3.2% may be determined for this mean, from the year to year variation of the effectiveness values. This weighted mean and standard error refer to a determination where the unrestrained passenger fatality counts are used as the control.

A very similar trail may be followed for a determination of driver Lap & Shoulder belt effectiveness, that uses restrained, instead of unrestrained, passenger fatality counts as the control. The result of this determination is a weighted mean of 47.6%, and a standard error of 4.0%, resulting from the year to year variation of the effectiveness.

There is no statistically significant difference between the effectiveness values determined as described above, based either on unrestrained or on restrained control passenger fatalities. The weighted mean value of the two, with an appropriately compounded stan-

Table 4. Summary of safety belt effectiveness determinations by the "subject/control pair" method, as a function of occupant seating position and belt type, in fatal car crashes, 1975-1985.

Belt Type	Seating Position Subject	Control	Wted Mean Effectiveness % & Std. Error: Control Unrestr.		Control Restrained		Unrstr., Restr. Wted Mean	
Lap & Shldr	1	3	49.7	3.2	47.6	4.0	48.9	2.5
Lap & Shldr	3	1	39.3	3.4	41.7	3.6	40.4	2.5
Lap Only	1	3	28.9	5.6	33.8	10.5	30.0	4.9
Lap Only	3	1	20.9	13.0	14.9	7.0	16.2	6.2
Lap Only	4	1,3	20.5	8.4	50.9	15.4	27.5	7.4
Lap Only	6	1,3	10.0	15.1	30.7	22.0	16.6	12.4
Lap Only	4,6	1,3	18.0	7.3	44.3	12.6	24.6	6.3

Seating Positions

1: Driver;
3: Front Seat Right Passenger;
4: Rear Seat Left Passenger;
6: Rear Seat Right Passenger;

dard error, may be used as the final answer for the driver Lap & Shoulder belt effectiveness. This is 48.9%, with a standard error of 2.5%.

This procedure offers the additional advantage of further spreading any potential biases. The results discussed above appear in the first row of the summary Table 4. This table summarizes the results obtained from all other possible determinations of safety belt effectiveness, as a function of belt type, subject occupant seating position, and control occupant restraint status.

The overall weighted means and their standard errors appear in the last column. In all cases, rows or columns, the results are obtained by the method described above, and are weighted means and standard errors, resulting from the year to year variations. In certain instances, such as those designated by composite seating positions 1,3 or 4,6, individual results are averaged across seating positions in order to deal with count sparsity that introduces unnecessarily large errors. The results summarized in Table 4 indicate the following fatality prevention potential: (a) about 50% for the driver restrained by a Lap & Shoulder belt; (b) about 40% for the front seat outboard passenger restrained also by a Lap & Shoulder belt; (c) a decline of each of the above by about 20 percentage points, when the Lap and Shoulder belt is replaced by a Lap Only belt; and (d) an effectiveness in the range 15% to 25%, with a rather large error, for rear seat passengers, who are restrained exclusively by Lap belts.

The Fatality Odds of Unrestrained Car Occupants

The principal conclusions from the evaluation of the FARS results discussed above may also be reached by an alternative analysis of essentially the same data. Car occupants from the FARS are considered again in controlled pairs, under the same general constraints.

The difference here is that seating position is not resolved, in favor of emphasizing the difference in fatality odds between unrestrained and restrained occupants that participate in the same car crash, irrespective of seating position.

In the actual application of this approach we address the FARS car population, with controlled occupant pairs, imposing all the filters considered in the preceding section. In addition, we filter so that we address cars where only one of the two occupants in the pair is restrained, and only one is killed, irrespective of restraint status or seating position.

Let (UK,RS) represent the count of cars where the unrestrained occupant was killed, while the restrained counterpart of this occupant survived. In this notation, U stands for Unrestrained, K for killed, R for Restrained, and S for Survivor.

Similarly, let (RK,US) represent the count of cars where the restrained occupant was killed, while the unrestrained counterpart survived. We shall consider the ratio of these two counts as a measure of the fatality odds for unrestrained occupants, i.e. as the odds of the unrestrained occupant being killed, while the restrained survives, as opposed to the converse. Specifically:

$$\text{Fatality Odds Ratio} = (\text{UK,RS}) / (\text{RK,US}) \qquad (21)$$

Results from such an evaluation are shown in Table 5, for a driver, front passenger pair. The restraint, when in effect is the Lap and Shoulder belt. This table presents the counts and fatality odds shown in relation (21), as a function of the FARS year. The bottom row sums the counts for all years and gives the weighted mean ratio of the fatality odds, as 2.3.

A standard error of 0.13 is assigned to this ratio based on its year to year variation. The mean and standard error for this case is shown as the first entry in Table 6, which summarizes the results for two additional cases: a driver/front passenger pair under a Lap Only belt, and a pair of rear seat occupants also under a Lap Only belt.

The Fatality Prevention Effectiveness of Ejection Avoidance

Ejection avoidance (occupant retention) is singled out for particular attention, as a fatality prevention countermeasure, because of: (a) the high incidence of ejection in fatal car crashes; (b) the high incidence of

Table 5. Determination of fatality odds based on fatality outcome in controlled pairs of unrestrained-restrained car occupants, as a function of FARS year.

Occupant Pair: (Driver, Front Seat Right Passenger);
(UK,RS) = Unrestrained Killed, Restrained Survives;
(RK,US) = Restrained Killed, Unrestrained Survives;
Fatality Odds = (UK,RS)/(RK,US);
Belt Type: Lap and Shoulder.

FARS Year	Outcome of Pairs (UK,RS)	(RK,US)	Fatality Odds
75	41	22	1.9
76	40	17	2.4
77	49	22	2.2
78	53	38	1.4
79	64	31	2.1
80	65	39	1.7
81	88	35	2.5
82	85	34	2.5
83	99	50	2.0
84	148	64	2.3
85	280	94	3.0
All	1012	446	2.3

Table 6. Summary of determinations of fatality odds as a function of safety belt type and subject-/control pair. Weighted means and std. errors; FARS 1975-1985.

Belt Type	Average of	Mean	S.E.
Lap & Shldr	Driver, Seat 3	2.35	0.13
Lap Only	Driver, Seat 3	1.60	0.22
Lap Only	Seat 4, Seat 6	1.57	0.21

fatality among car ejectees; and (c) the very high effectiveness that ejection avoidance has in preventing fatality, as will be seen shortly.

This section addresses the fatality prevention effectiveness of occupant retention, irrespective of the specific countermeasure(s) used to achieve such retention. This subject is examined quantitively, based on the FARS data analyzed in a fashion similar to that used in connection with the safety belt effectiveness.

Specifically in this section, we shall perform analyses of fatality odds, as a function of ejection status, and evaluations of the fatality prevention effectiveness of occupant retention, by simple transformations of analyses performed earlier, as a function of occupant restraint status.

Here however, the occupants under consideration, whether subjects or controls are unrestrained, since we are seeking the benefits of ejection avoidance, irrespective of restraint status.

For the purpose of this analysis, consider the data shown in Table 7. This table is structured like Table 3, discussed earlier. Paired counts of fatalities (DDN,PNN) and (DEN, PEN) are shown in Table 7, as a function of FARS year 1975 to 1985.

In this notation, D and P stand for a Driver and a Passenger, respectively; NN stands for cars where neither the subject driver nor the control passenger were ejected; EN stands for cars where the subject driver was ejected, but the control passenger was not.

Table 7. Car driver/passenger paired counts, driver fatality odds, and fatality prevention effectiveness of driver retention (non-ejection), based on FARS data. Control: retained (non-ejected) front passenger.

FARS	Paired Counts				Fatality Odds		Eff. %
Year	DNN	PNN	DEN	PEN	DNPN	DEPN	EDPN
75	1252	1524	306	84	0.82	3.64	77.4
76	2467	2835	438	147	0.87	2.98	70.8
77	3586	4051	644	196	0.89	3.29	73.1
78	3873	4271	706	187	0.91	3.78	76.0
79	3813	4191	671	200	0.91	3.36	72.9
80	3844	4190	673	190	0.92	3.54	74.1
81	3878	4089	687	175	0.95	3.93	75.8
82	3227	3525	643	175	0.92	3.67	75.1
83	3051	3408	598	166	0.90	3.60	75.1
84	3109	3374	598	145	0.92	4.12	77.7
85	3164	3605	639	159	0.88	4.02	78.2
All	35264	39063	6603	1824	0.90	3.62	75.1

Table 7 also gives the fatality odds, DNPN and DEPN, determined according to relations (22) and (23), as well as the fatality prevention effectiveness, EDPN, resulting from relation (24).

$$\text{DNPN} = \text{DNN} / \text{PNN} \tag{22}$$

$$\text{DEPN} = \text{DEN} / \text{PEN} \tag{23}$$

$$\text{EDPN} = (\text{DEPN} - \text{DNPN}) / \text{DEPN} \tag{24}$$

In this notation, DNPN stands for the fatality odds of non-ejected driver subjects, when controlled by paired non-ejected passengers, (acronym for Driver Non-Ejected, Passenger Non-Ejected); DEPN stands for the odds of ejected driver subjects, when controlled also by non-ejected passengers. In the last relation, EDPN is the resulting effectiveness of fatality prevention, through ejection avoidance. Note that relations (22) to (24) are very similar to (18) to (20).

The bottom row in Table 7 gives counts for all years combined. It also gives weighted mean values of fatality odds and of the resulting effectiveness. The weighted mean value of this effectiveness for the driver is 75.1%. A standard error of 0.7% may be determined for this mean, from the year to year variation of the effectiveness values.

This weighted mean and standard error refer to a determination where the non-ejected passenger fatality counts are used as the control. A very similar trail may be followed for a determination of driver fatality prevention effectiveness, that uses ejected, instead of non-ejected, passenger fatality counts as the control.

The result of this determination is a weighted mean of 73.5%, and a standard error of 1.0%, resulting from the year to year variation of the effectiveness. This mean differs non-significantly from the one determined earlier with non-ejected controls. The two may be appropriately combined to yield 74.6% with a standard error of 0.6%.

The results discussed above appear in the first row of the summary Table 8. This table summarizes the results obtained from all other possible determinations of the fatality prevention effectiveness, offered by ejection avoidance, as a function of subject occupant seating position, and control occupant seating position and ejection status.

The overall weighted means and their standard errors appear in the last column. In all cases, rows or columns, the results are obtained by the method described above, and are weighted means and standard errors, resulting from year to year variations. In certain entries, such as those designated by composite seating positions 3,4 or 3,4,6 etc., individual results are averaged across seating positions in order to minimize errors and distil the results.

A review of the results of Table 8 shows a remarkable consistency in high effectiveness values for

Table 8. Summary of fatality prevention effectiveness of car occupant retention (non-ejection), by the "subject/control pair" method, as a function of occupant seating position, in fatal car crashes, 1975-1985.

Seating Position		Wted Mean Effectiveness % & Std. Error					
		Control Non- Ejected		Control Ejected		Non-Ejctd, Ejctd Wted Mean	
Subject	Control						
1	3	75.1	0.7	73.5	1.0	74.6	0.6
3	1	67.1	1.1	69.1	0.8	68.4	0.6
1	4	60.0	2.0	70.1	2.0	65.1	1.4
4	1	74.1	1.8	65.4	1.9	70.0	1.3
1	6	69.2	2.1	74.4	2.3	71.6	1.6
6	1	77.1	1.3	72.5	2.5	76.1	1.2
3	4	63.1	1.8	74.2	2.1	67.8	1.4
4	3	78.6	1.6	69.3	2.5	75.9	1.3
3	6	57.6	2.2	64.7	3.8	59.4	1.9
6	3	72.7	2.4	67.3	2.5	70.1	1.7
4	6	70.9	3.5	67.8	2.3	68.7	1.9
6	4	67.3	1.5	70.4	3.5	67.8	1.4
1	3,4	74.6	0.6	65.1	1.4	73.1	0.6
3	1,4	68.4	0.6	67.8	1.4	68.3	0.6
4	1,3	70.0	1.3	75.9	1.3	73.0	0.9
6	1,3	76.1	1.2	70.1	1.7	74.1	1.0
1	3,4,6	73.1	0.6	71.6	1.6	72.9	0.6
3	1,4,6	68.3	0.6	59.4	1.9	67.5	0.6
4	1,3,6	73.0	0.9	68.7	1.9	72.2	0.8
6	1,3,4	74.1	1.0	67.8	1.4	72.0	0.8

Seating Positions

1: Driver;
3: Front Seat Right Passenger;
4: Rear Seat Left Passenger;
6: Rear Seat Right Passenger;

fatality prevention by ejection avoidance, virtually irrespective of occupant seating position. In the last four entries, which show distilled results, we see an effectiveness between 72% and 73% for all car occupant seating positions, front or rear, except for that of the front passenger, which shows an effectiveness about five percentage points lower. Similar results are obtained in reference 7.

In addition to the benefits that car occupants derive from ejection avoidance, we addressed the benefits to occupants of light trucks and vans. We did this because there is a much higher incidence of ejections in occupants of light trucks and vans, and because these vehicles are often involved in more severe crashes than cars are. For this purpose we followed an approach identical to that described above. We limited, however, the analyses to front seat outboard occupants, i.e. to drivers as subjects, with front passengers as controls, and vice versa. We addressed four categories of light trucks and vans: Pickups, Vans, Other (Utility and Multipurpose Vehicles), and All combined.

The results of these analyses are summarized in Table 9, which is structured in a fashion identical to that of Table 8, discussed earlier. The final results may be reviewed in the last column of the table.

It is evident that the fatality prevention effectiveness of ejection avoidance for occupants of light trucks and vans is not only consistent across the board, but also higher than that found in the case of car occupants. This effectiveness exceeds 80% for the driver, and is close to 78% for the front passenger.

Table 9. Reference to occupants of light trucks and vans. Summary of fatality prevention effectiveness of occupant retention (non-ejection), by the "subject/control pair" method, as a function of occupant seating position, in fatal car crashes, 1975-1985.

Type of Vehicle	Occupant's Seating Position		Wted Mean Effectiveness % & Std. Error					
	Subject	Control	Control Non-Ejected		Control Ejected		Non-Ejctd, Ejctd Wted Mean	
Pick Up	1	3	81.4	1.3	79.9	0.9	80.4	0.7
Pick Up	3	1	77.0	1.6	78.7	2.1	77.6	1.3
Van	1	3	79.2	3.8	79.3	5.1	79.2	3.0
Van	3	1	78.3	4.2	78.2	3.6	78.2	2.7
Other	1	3	85.0	3.5	84.3	2.7	84.6	2.1
Other	3	1	77.6	9.2	78.6	4.5	78.4	4.0
All	1	3	81.7	1.2	80.3	0.9	80.8	0.7
All	3	1	77.1	1.5	78.8	1.9	77.8	1.2

Other Vehicle: Utility, Multi-Purpose, On/Off the Road.

Seating Positions
1: Driver;
3: Front Seat Passenger.

The Fatality Odds of Ejected Occupants

An alternative approach may be followed in analyzing the same basic FARS data, but in such a way as to emphasize more the large difference in fatality odds between ejected and non-ejected occupants, under very similar crash conditions. This approach parallels the one followed earlier, in connection with occupant restraint status.

We address specific pairs of unrestrained occupants, in cars where only one of the two is ejected, and only one is killed, irrespective of ejection status or seating position. Let (EK,NS) represent the count of cars where the ejected occupant was killed, while the non-ejected counterpart of this occupant survived. In this notation, E stands for Ejected, K for Killed, N for Non-Ejected, and S for Survivor.

Similarly, let (NK,ES) represent the count of cars where the non-ejected occupant was killed, while the ejected counterpart survived. We shall consider the ratio of these two counts as a measure of the fatality odds for ejected occupants, i.e. as the odds of the ejected occupant being killed, while the non-ejected survives, as opposed to the converse. Specifically:

$$\text{Fatality Odds Ratio} = (EK,NS) / (NK,ES) \qquad (25)$$

Results from an evaluation of this relation are summarized in Table 10, for car occupant pairs in all combinations of seating position. The values shown in this table are weighted means and standard errors, resulting from the year to year variations of the FARS 1975 to 1985 data.

It is evident that the odds of fatality for car ejectees are about six times higher by comparison to non-ejectees subjected to very similar crash conditions, irrespective of seating position. Note that the non-

Table 10. Fatality odds of ejected versus non-ejected car occupants, evaluated from the FARS 1975-1985 experience of shown pairs. Weighted mean values and standard errors due to year to year variation.

Average of	Mean	S.E.
Driver, Seat 3	5.6	0.11
Driver, Seat 4	5.6	0.40
Driver, Seat 6	6.4	0.31
Seat 3, Seat 4	6.1	0.60
Seat 3, Seat 6	5.0	0.20
Seat 4, Seat 6	5.9	0.64

ejectees here, indeed all occupants under consideration are unrestrained, since we seek to isolate the benefit of ejection avoidance, irrespective of how occupant retention is achieved.

The fatality odds for light truck and van ejectees are even higher, than those of car ejectees, as the summary of results shows in Table 11.

The Potential Benefits of Ejection Avoidance

In order to evaluate the potential benefits of the very high effectiveness of ejection avoidance, we have drawn two tables of the target populations from the FARS. These are Tables 12 and 13, the first addressing car occupant fatalities, and the second, fatalities of light truck and van occupants.

Shown in these tables are the distributions of occupant fatalities per year, as a function of ejection status and occupant seating position. The annual values are averages of the four recent FARS years: 1982 to 1985. The ejection occurrence designations follow the FARS classification.

Under "Both", we include the sum of complete and partial ejections, which is what we addressed in the analyses of this paper. Seating positions are self explanatory. Under "Other & Unknown" we include the very few counts of occupants that have no appropriate seating position.

Table 11. Fatality odds of ejected versus non-ejected occupants of light trucks and vans, evaluated from the FARS 1975-1985 experience. Weighted mean values and standard errors due to year to year variation.

Body Type	Average of	Mean	S.E.
Pickup	Driver, Seat 3	7.7	0.6
Van	Driver, Seat 3	8.0	1.4
Other	Driver, Seat 3	9.6	1.3
All	Driver, Seat 3	7.8	1.6

Table 12. Distribution of annual fatalities of car occupants, by ejection status and seating position. Average of FARS years 1982 to 1985.

Occupant's Seat	Ejection Occurrence					
	None	Complete	Partial	Both	Unknown	Total
Driver	11441	2895	703	3599	341	15381
Front Right	3807	957	191	1147	58	5013
Rear	1298	350	49	398	20	1716
Front Other	307	87	21	108	68	483
Other & Unknown	304	131	9	140	19	463
All	17156	4419	973	5391	507	23054

Specifically, the count of car occupants, that are classified as ejected but have been riding in trailing units, vehicle exterior, or in other non enclosed areas is about 2 per year. A similar count for occupants of light trucks and vans is about 98 per year, when averaged over the FARS years 1982 to 1985.

A review of the data in Table 12 shows that out of about 23,000 car occupants killed per year, 20% to 25%, or between 4,500 and 5,500 per year are ejectees, depending on the specific characterization of the ejection. For light truck and van occupants, Table 13 shows that out of the approximately 6,000 killed per year, 35% to 40%, or between 2,100 and 2,400 per year are also ejectees.

Thus there are between 6,500 and 8,000 people per year, roughly 2/3 of them car occupants and 1/3 occupants of light trucks and vans, that are killed ejectees. This is the population that may benefit from the high values, 70% to 80%, determined earlier for the fatality prevention effectiveness of ejection avoidance.

Ejection avoidance may be achieved by various means with various degrees of success. The restraining action of safety belts is virtually 100% effective in preventing ejection.

Other countermeasures against ejection are related to door retention, and to the resistance of car windows and other glazed surfaces. Last but not lease, rollover resistance can be an important countermeasure, given that car rollover promotes ejections.

As a closing remark for this section it should be noted that the proportional rates quoted above for killed ejectees are averages over all cars or all trucks

Table 13. Distribution of annual fatalities of light truck and van occupants, by ejection status and seating position. Average of FARS years 1982 to 1985.

Occupant's Seat	Ejection Occurrence					
	None	Complete	Partial	Both	Unknown	Total
Driver	2500	1271	266	1536	67	4103
Front Right	627	389	62	451	8	1085
Other & Unknown	426	340	23	364	24	814
All	3553	1999	351	2350	99	6002

Table 14. Estimates of contribution made by ejection avoidance to the overall fatality prevention of safety belts as a function of occupant's seating position and belt type.

Occupant	Belt Type	e %	e2 %	(N2/N) %	(N2/N)*(e2/e) Percent
Driver	Lap & Shldr.	49	73	24	36
Front Pass.	Lap & Shldr.	40	68	23	39
Driver	Lap Only	30	73	24	58
Front Pass.	Lap Only	16	68	23	97
Rear Pass.	Lap Only	25	72	23	66

and vans, respectively. A resolution of cars by market class, or of light trucks and vans by functional classification, will reveal highly non uniform proportional rates.

Much higher rates are generally associated with smaller cars, or with certain categories of light trucks and vans. This results from correspondingly higher rates of rollovers, which are crashes that promote ejections.

Contribution of Ejection Avoidance to the Overall Fatality Prevention of Safety Belts

Safety belts prevent ejections, in addition to providing protection against occupant impacts with vehicle interior surfaces. In order to estimate the contribution of ejection avoidance to the overall fatality prevention effectiveness of safety belts, we make the assumption that safety belts, irrespective of type, provide virtually 100% ejection control, irrespective of occupant's seating position.

The overall fatality prevention effectiveness of seat belts may be approximated by:

$$e = (N1/N)*e1 + (N2/N)*e2 \qquad (26)$$

where: subscript 1 relates to non-ejection events, subscript 2 relates to ejection, e1 and e2 are the respective effectiveness values, and (N1/N), (N2/N) are the respective occupant fatality proportions in each of the above modes.

It is evident from the above approximation that the contribution made by ejection avoidance to the overall fatality prevention of safety belts is given by:

$$(N2/N)*(e2/e)$$

This contribution may be evaluated for car occupants, as a function of occupant's seating position and safety belt type, on the basis of results appearing in Table 12 for values of (N2/N); Table 8 for values of e2; and Table 4 for values of e.

The results of this evaluation are shown in Table 14, where the last column tabulates the contribution under consideration. Errors for these values may be estimated on the basis of the errors reported in Table 4. A similar procedure may be followed for occupants of light trucks and vans.

Conclusions

The FARS data, when analyzed under controlled crash conditions, indicate the following fatality prevention effectiveness of safety belts: (a) about 50% for the driver restrained by a Lap & Shoulder belt; (b) about 40% for the front seat outboard passenger restrained also by a Lap & Shoulder belt; (c) a decline of each of the above by about 20 percentage points, when the Lap and Shoulder belt is replaced by a Lap Only belt; and (d) an effectiveness in the range 15% to 25%, with a rather large error, for rear seat passengers, who are restrained exclusively by Lap belts.

Ejection avoidance, as effected by safety belt use, irrespective of occupant seating position and belt type, is a very significant contributor to the fatality prevention effectiveness quoted above. In lap belt applications, ejection avoidance is the major contributor to fatality prevention.

Ejection avoidance deserves particular attention in-crash protection because of: (a) a large fraction of occupant fatalities are associated with ejection, (24% in cars; 40% in light trucks and vans), and (b) the remarkably high effectiveness of ejection avoidance in preventing fatalities (70% to 80%, irrespective of occupant seating position and motor vehicle category).

This high effectiveness of ejection avoidance is confirmed in analyses of the fatal accident experience of car and light truck occupants, under controlled crash conditions. Furthermore, it is independent of the confounding effects introduced by inaccuracies in the reporting of safety belt use rates.

Ejection avoidance as a general countermeasure is relevant to between 6,500 and 8,000 people per year, roughly 2/3 of them car occupants and 1/3 occupants of light trucks and vans, that are killed ejectees. This is the population that may benefit from the high effectiveness of ejection avoidance.

Safety belt use is the most straightforward but not the only way to provide ejection avoidance for motor vehicle occupants. Other countermeasures against ejection are related to door retention, and to the resistance of car windows and other glazed surfaces. Last but not least, rollover resistance can be an important countermeasure, given that car rollover promotes ejections.

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Technical Session Three

Biomechanics and Dummy Development

Chairman: Dr. Dominique Cesari, France

On the Examination of Biases in the Abbreviated Injury Scale

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Abstract

The Abbreviated Injury Scale is a catalog of injury lesion descriptions which are indexed by anatomical region and ranked by a numerical scale which is defined to represent "injury severity." It was developed by a panel of experts using a consensus process and did not utilize any quantitative restrospective analyses of trauma data bases. The common notion of the individual AIS "injury severity" values, which range from 1 to 6, is that they are indicative of the "threat to life" that the individual lesion would present to a living individual and the higher this numerical value, the higher the risk of death would be. It is also implied that all lesions ranked at a specific AIS level would have the same "severity" (threat-to-life) regardless of their anatomical location.

This study examines both of the above notions of the Abbreviated Injury Scale. This is accomplished by analyzing over five hundred cases recorded in the National Accident Sampling System which have been stratified by AIS levels of the two most severe injuries and the anatomical location of each injury. Both the analysis techniques used and the results that assign both an absolute survival rate to each AIS level/body region combination as well as how the compensatory effects of multiple injuries of various severity levels and anatomical locations affect survival are discussed.

Introduction

The Abbreviated Injury Scale (AIS)[1,2,3] has become one of the widest used schemes for classifying traumatic injuries and their severity due, in part, to its universal use in various automotive crash investigation programs (NCSS and NASS) sponsored by the National Highway Traffic Safety Administration. Since its inception and throughout all of its subsequent revisions, the format of the AIS remains as a catalog of injury lesion descriptions which are classified by anatomical location and ranked by a numerical scale ranging from one to six which is defined to represent "injury severity." The earlier versions of the AIS, 1976 and before, included, parenthetically, qualitative descriptions of what the various levels of the AIS severity ratings represent, i.e., 3: Severe (Not Life-Threatening), 4: Serious (Life-Threatening), 5: Critical (Survival Uncertain), and 6: Maximum (Currently Untreatable). The later revisions, 1980 and 1985, have eliminated this practice. The reason given was "that they may be misleading to the coder in suggesting that the AIS is based upon the single criterion of threat to life" and that threat to life "is only one of several criteria used for developing AIS codes"[2]. Further investigation of the provided references[4,5] revealed other categories of traumatic consequences that may have been considered, e.g., permanent impairment, energy dissipation, treatment period, but they did not reveal how these additional criteria are considered in the assignment of a single AIS severity level nor did it reveal any additional definitive statements of what, in the context of the presently used numerical scale, the AIS injury severity ranks really represent. It appears, therefore, that, by default alone, both the committee of experts who developed the individual rank values as well as the users of the AIS system have and are operating under the common notion that the AIS severity ranking is solely a measure of the threat to life and, in addition, that all lesions rated with a specific AIS severity ranking have the same threat to life. It is these two common notions that will be explored in the following sections.

Investigations into the relationship between the AIS rankings assigned to lesions and the threat-to-life that those lesions present to the person experiencing them are not new. In a pioneering effort, Baker, et al.[6] developed the Injury Severity Score (ISS), a numerical process which combines the highest three AIS rankings from each of three individual body areas into a single numerical rank, and demonstrated that this ISS value had strong correspondence with mortality, morbidity, clinical stay, and other indicators that vary as a consequence of a victim's number and severity of injuries. While the ISS demonstrated some successful correlation, it was based on two very strong, a priori assumptions that the authors neither justified nor verified. These being: 1. Only the highest AIS injury severity ranking from any one anatomical region should be used, and 2. The highest three individual

AIS rankings shall be combined as the sum of their squares to form the ISS.

These assumptions lead to some questions and potentially conflicting conclusions. One conflicting conclusion being that if, as the authors of the ISS assert, secondary injuries do effect outcome, then why should an injury that has an AIS ranking greater than all other injuries except the primary (highest) one be ignored solely because it is in the same anatomical area as the primary AIS injury? Other unanswered questions are: Why is the sum of the squares the most appropriate functional method to combine the rankings of multiple injuries into a single value? Why not the sum of the cubes, or some other form? What is the criteria for making that judgment? Subsequent studies[7,8] have eliminated the above issues by partitioning injury cases by either the two or three highest AIS injuries regardless of anatomical location and examining how the fatality rate changes in each of the classifications.

All of these studies, regardless of their technique, show that, while the AIS ranking system is said to consider more than threat-to-life, the common interpretation of the AIS ranking system being a measure of the threat-to-life is not an invalid one.

Method

The approach followed in this effort further expands on the efforts of Eppinger[7] and Ulman[8], by not only stratifying the case injury data by the two highest AIS rankings occurring in the body but by additionally stratifying the cases by the anatomical locations of the lesions considered. That is, each AIS three, four, and five is further classified by whether it occurred in the head, chest, or abdomen (this results in 9 possible region/AIS combinations). When all possible combinations and permutations of these nine combinations are considered, 45 individual classifications are developed.

The National Accident Sampling System data base for the combined years of 1982 through 1985 was then queried to determine the number of fatals and nonfatals in each of the 45 injury categories. This allowed the calculation of a survival rate for each of the 45 injury combinations (number of survivors divided by total number of cases in each category) to be made and the matrix shown in Figure 1 to be generated. The development of this data was not a straight forward census of fatals and survivors in each body region/AIS combination in the crash data files because of the stratified sampling system employed by NASS. Since this system only investigates a portion of the total crashes occurring in any investigative geographical area and those that are investigated are chosen by a predetermined sampling process that varies depending on the type and severity of the crash event, the completed data file resulting from the investigation of all chosen events not only contains the detailed information obtained by the investigators from each event but also contains a weighting factor that represents how often this type of event was sampled. In order to create a data file wherein each event has the same probability of occurring, each crash event in the raw data file is multiplied by its associated sampling factor and a new file, designated the weighted file, is created. It is from this weighted file that the number of fatals and survivors for each body region/AIS combination was obtained and the resulting survival rate calculated. (It must be noted that because of the limited number of actual events in many of the multiple injury categories and because of the sampling and weighting strategies employed in NASS, estimation of the accuracy of any of the calculated survival rates is not possible). Since the weighting factors varied considerably among the 45 different body region/AIS combinations, it was felt that it was more appropriate to use the actual count of events in each of the 45 categories in the raw file when generating the sum of squared error parameter used in the optmization procedure described below.

In order to address both issues relating to the AIS classification scheme under study, i.e., the relationship of the AIS classification to the threat-to-life and the equality of the threat-to-life evaluation in different anatomical areas, it became necessary to assume a mathematical form of how the hazards of multiple injuries combine. Therefore, it was assumed that the ultimate survival rate associated with a particular multiple injury class is equivalent to the product of the survival rates associated with each individual body region/AIS injury combination. That is,

$$SR_{AIS1,AIS2} = SR_{AIS1} * SR_{AIS2}$$

PRIMARY (HIGHEST) INJURY

SECONDARY INJURY	H5	C5	A5	H4	C4	A4	H3	C3	A3
H5	0.54 6								
C5	0 1	0 12							
A5	0 3	0.14 26	0.51 17						
H4	0.38 17	1 1	NO DATA	0.45 29					
C4	0.13 13	0.37 17	0.09 12	0.1 10	0.5 11				
A4	0.58 9	0.1 11	0.76 16	0.77 8	0.18 27	0.71 18			
H3	0.59 19	0.24 4	0 2	0.75 43	0.76 3	0.84 6	0.62 30		
C3	0.81 9	0.81 8	1 1	0.8 4	0.8 16	1 8	0.83 16	0.95 37	
A3	0.69 5	0 1	0.86 5	NO DATA	0.18 2	1 15	0.87 10	0.78 18	1 13

Figure 1. Survival rate/number of cases (NASS '82-'85)

where:

SR_{AIS1}—Survival rate associated with highest injury

SR_{AIS2}—Survival rate associated with second highest injury

$SR_{AIS1,AIS2}$—Survival rate associated with combined injuries

This proposed algorithm for combining the effects of multiple injuries has a logical basis for its form. This can be demonstrated by examining several extremes conditions as well as a more nominal condition. First, if the second highest injury presents no hazard to life, it should not reduce survival rate at all. This the algorithm does. Secondly, if the primary injury is extremely severe and its associated survival rate is near zero, the presence of a second injury should not have a great influence nor should it make the survival rate less than zero. This, the algorithm also handles. And, in the third case of two injuries which, each by themselves have a specific survival rates associated with them, the algorithm assumes the influence of the second highest injury would have no effect on the percent expected to die as a consequence of the first injury and would reduce the percentage of those surviving by the survival rate associated with the risk of the second injury. Therefore, if a unique survival rate associated with each body region-AIS level combination is known, the resulting survival rate for any combination of two injuries can be calculated. It was with this hypothesized model that the injury rate data from NASS was analyzed. It is recognized that other forms of multiple injury/survival rate models are possible and could even, ultimately, explain more of the data. Such models were not pursued because, without an assessment of the accuracy of the survival rates obtained form the NASS data, evaluation of the relative performance of models is not very meaningful.

The analysis was accomplished by developing and employing a weighted, least squared error methodology which assumes a survival rate for each of the nine possible body region/AIS combinations, calculates a combined survival rate for each of the 45 multiple injury combinations, subtracts the calculated survival rate from each observed survival rate and squares this difference, multiplies each of these differences by the number of field observations from which each observed survival rate was derived, and sums all 45 squared, weighted errors to form a weighted, sum of square error parameter. This process was then put under the control of an numerical optimizer[9] to search for the best combination of the 9 individual survival rates that would minimize this overall summed, weighted, squared error. The results of this optimization effort are listed in Table 1 and a comparison of the actual and predicted survival rates are shown in Figure 2.

Table 1. Calculated optimum survival rates

(Body Region/AIS Rank)		
Head 5 = 0.65	Chest 5 = 0.36	Abdomen 5 = 0.69
Head 4 = 0.74	Chest 4 = 0.45	Abdomen 4 = 0.88
Head 3 = 0.84	Chest 3 = 0.99	Abdomen 3 = 0.98

Discussion

Straight forward examination of the results presented in Table 1 suggest the conclusions reached by others[6,8,10], that the AIS severity rankings have a relationship with the threat-to-life posed by a particular lesion, are supported by this study. That is, in each major body area, the survival rate descreases monotonically with increasing AIS rank. However, this study also indicates that the survival rate due to injuries in the various body regions having the same AIS ranking, either 3,4, or 5, may vary significantly. This is seen, for example, by noting that the calculated average survival rate attributable to an AIS 5 injury in the head is 65%, while in the chest and abdomen the respective survival rates are 36% and 69%. A similar result is observed with the calculated survival rates for the AIS 4 injuries. Here the chest has a calculated survival rate of 45%, while the Head has 74% and the abdomen has 88%.

It must be noted here that the observed bias in survival rate among the various body regions is the combined averaged effect of those injuries occurring in the NASS data file in each particular body region-

PRIMARY (HIGHEST) INJURY

SECONDARY INJURY	C5	C4	H5	A5	H4	H3	A4	A3	C3
C5	0 0.127								
C4	0.37 0.161	0.5 0.205							
H5	0 0.232	0.13 0.295	0.54 0.424						
A5	0.14 0.246	0.09 0.313	0 0.45	0.51 0.48					
H4	1 0.26	0.1 0.333	0.38 0.48	NO DATA 0.51	0.45 0.54				
H3	0.24 0.301	0.76 0.383	0.59 0.55	0 0.585	0.75 0.62	0.62 0.72			
A4	0.1 0.315	0.18 0.48	0.58 0.58	0.76 0.61	0.77 0.65	0.84 0.75	0.71 0.78		
A3	0 0.349	0.18 0.443	0.68 0.64	0.86 0.68	NO DATA 0.72	0.87 0.86	1 0.86	1 0.96	
C3	0.81 0.35	0.8 0.45	0.81 0.65	1 0.686	0.8 0.73	0.83 0.88	1 0.87	0.78 0.97	0.95 0.98

Figure 2. Actual survival rate/predicted survival rate

AIS rank. It was not possible, with the limited number of observations in each category, to determine if the bias was due to one particular lesion having a severely incorrect ranking or due to all lesions having about the same survival rate bias.

It should also be noted that the survival rate data derived from NASS may be somewhat confounded by the presence of third and fourth injuries. Again, the limited quantity of data in the NASS files prevented sorting by the three highest injuries and obtaining a sufficient number of cases in each category to calculate a survival rate. Therefore, a sampling strategy was employed that considered only those cases where the third highest injury severity was less than or equal to and from the same body region as the second or the third highest injury was less than the severity of the second highest injury and from any other body region. No method could be developed that could assess the influence that this strategy had on the accuracy of the file determined survival rate.

Age has been shown to have a considerable effect on survival rate. Evans[11] using the Fatal Accident Reporting System and an analysis technique called the "double pair comparison method" estimates a four fold increase in fatality probability between a 20 year old and 80 year old victim given all other crash conditions equal. An examination of the data used to develop the individual survival rates in Figure 1 was therefore undertaken to see if age was contributing in some systematic manner to the calculation of the survival rate. This examination revealed similar age distributions among both the fatal and non-fatal groupings of the various individual categories. As a result, it was concluded that age was not a confounding factor in this analysis.

Examination of residual errors resulting from the differences between the survival rates predicted by the model and the survival rates from the data, however, did reveal some interesting observations. Specifically, it was observed that the survival rate model predicts a survival rate that is on the average 21% higher when any thoracic AIS 4 or 5 injury was combined with an abdominal injury of any AIS rank (indicating actual event more life threatening) while the model predicts a 22% lower rate (actual event less life threatening) when both the primary and secondary injuries are in the thorax. All other averaged residual errors did not show this great a disparity and were at least half of these values, i.e., thoracic 4 and 5 with any head AIS—predicts 9% high; any head with any abdominal injury—predicts 1% high, head with any head—predicts 0.08% high, and any abdominal injury with any abdominal injury—predicts 6% low. This suggests that the influence of multiple injuries may be more complex than the multiplicative survival rate relationship assumed and used in this analysis and that additional effort must be devoted to this subject if a better understanding is to be gained.

Conclusions and Recommendations

As a result of conducting the above analysis the following conclusions and recommendations are offered:

1. A definitive statement explaining what is meant by the various AIS injury severity ranks used in the AIS system is not available and, therefore, prevents examination of the accuracy of the classifications, causes confusion in the use of files incorporating the AIS classifications, and brings into question the meaningfulness of recent AIS revisions. Such a definition effort should be pursued.
2. The common notion that the AIS ranking scale is a system that primarily ranks the threat-to-life a lesion presents to a victim has been reconfirmed by this study. This study also suggests that the threat-to-life posed by lesions of equal AIS rank, but residing in different anatomical regions, is not the same. Therefore, studies that combine like ranked AIS injuries from different anatomical areas and consider their associated threat-to-life equal, should do this with considerable caution.
3. A relationship between the simple multiple injury model assumed in this study and the major synergistic effects of multiple injuries has been demonstrated. A residual error analysis indicates that second order effects, regulated by other anatomical distribution considerations, have additional influences on survival rate. Therefore, considerably more effort should be devoted to obtaining, assembling and analyzing additional field data if the hazards of individual injuries and the effects of multiple injuries are to be better understood.

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Priorities of Automobile Crash Safety Based on Impairment

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Introduction

The determination of benefits from, and the setting of priorities for automotive crash injury protection programs requires the rating of the severity of injuries sustained during a crash being studied. Typically, data bases of accident statistics are used to reveal relationships between some vehicle component, such as the steering assembly, and the injuries sustained. The injuries are often rated in terms of their severity with a numerical scale, such as the Abbreviated Injury Scale (AIS)[4]. Injury severity scales such as AIS are generally based on the threat to life represented by an injury. However, the threat to life is not necessarily the best measure of the severity of an injury.

While reduction of deaths due to automobile crashes is a goal of crash safety research, it is also important to reduce serious injury due to crashes. Injuries exist which do not represent a large threat to life, but which have drastic consequences to the individual injured. Often an individual will never fully recover, leaving them with some permanent or long lasting impairment. In contrast, an injury may represent a serious threat to life, but with quick medical intervention may be readily treatable with a full and quick recovery expected. Consider the following examples from a study of crash victims admitted to the Washington Hospital Center Shock-Trauma Unit. In one case a driver suffered a serious AIS 4 abdominal injury, while in another case the driver sustained an AIS 2 head injury. The AIS 4 abdominal injury, while a serious injury representing a real threat to life, had a good prognosis for full recovery after the victim was transported to a shock-trauma unit, and in fact was discharged approximately one month after the crash, and was able to resume a full, normal life. In contrast the AIS 2 head injury did not improve with time, and resulted in a permanent disability.

Clearly, both types of injuries are serious and deserve attention. However, many methods of prioritizing crash safety issues would rank the head injury as a much less serious problem, if a problem at all, because of its low AIS. To address this problem of injuries with serious impairment consequences but a small threat to life being ranked as less important than injuries that are a serious threat to life, but which are readily treatable with little or no long term consequences, Hirsch et. al.[2] developed an impairment severity ranking scheme. Hirsch's system allows one to relate an Occupant Injury Code (OIC) to periods of impairment at 4 levels of severity in 6 functional areas for 3 periods of time.

In this paper Hirsch's system will be used with accident data from the National Accident Sampling System (NASS) from 1982, 1983, 1984, and 1985. The coupling of NASS data with Hirsch's impairment severity rating scheme will be used to determine the relative importance of various vehicle components (such as windshield, instrument panel, etc.) as sources of impairment in both frontal and lateral collisions. A similar analysis will determine the relative importance of the different body regions as sources of impairment.

Previous Work

One of the first attempts to use a large file of accident data with sophisticated estimates of the societal consequences of injury was by Malliaris, et al.[1]. Malliaris termed the economic consequences of an injury "Harm", and sought to find where the greatest Harm lay and what the potential was for reducing Harm. Malliaris used the National Crash Severity Study and an NHTSA societal cost study from 1975. The severity of an injury was based solely on the AIS level. Differences due to body region or age of the individual were ignored. In addition, because AIS is primarily a threat to life scale, Harm was unable to evaluate injuries (such as the head injury discussed in the introduction) which result in a serious impairment, but which are not a great threat to life.

Hirsch, et al.[2] developed a system for evaluating the consequences of an injury. This system will be used in the current study, and will be discussed in detail shortly. The Hirsch study made it possible to determine the likely consequences of an injury for the remainder of the person's life, given an OIC, an AIS and an age.

Following the publication of Hirsch's system Carsten and O'Day[3] used this system to prioritize crash injury safety programs. They analyzed Hirsch's study and added to it in several areas. The primary aim of Carsten and O'Day was to combine Hirsch's system with NASS data from 1980 and 1981 (which was all that was available at the time). In addition, they did research on the relative importance of the different functional impairment categories. When reporting their findings they wished to simplify the impairment results. Whereas Hirsch reported impairments at 4 different severity levels within each of 6 different categories for 3 periods of time, Carsten and O'Day developed a system for translating impairments in different functional categories into a single whole body impairment. They then based their results and analyses on this single whole body impairment, which they called the Injury Priority Rating (IPR), for the three different periods of time that Hirsch used. Among the results of their study was that the head, face, and neck are responsible for 60% of the total IPR to passenger car occupants; in excess of one-third of all driver IPR comes from impacts with a 12 o'clock direction of force (straight head on); that 84% of driver IPR from collisions with a 12 o'clock direction of force comes from crashes with a delta V in excess of 20 mph; that Harm and IPR were agreed on the relative priority with respect to the directions of force, but when ranking the priority of the different body regions, IPR gave a higher priority to head, face, and neck injuries than Harm did.

Impairment Severity Rating Scheme

Hirsch's system uses a 4 level impairment scale (1-minor, 4-severe) in 6 different functional areas (mobility, cognitive/psychological, cosmetic, sensory, pain, and daily living) for 3 lengths of time (short term, medium term, and long term). For each available OIC which results in an AIS between 2 and 5 the resulting impairment is defined in each functional area for each period of time. The impairment which results is dependent on the age of the individual injured, with age divided into 4 groups (less than 16, 16-45, 46-65, and over 65), and on the AIS of the injury. For short term impairment (less than a year) the rating is in terms of duration at each impairment level, while medium term (years 2 through 5) and long term (more than 5 years) are defined in terms of a single severity level within each functional category. As an example of short term impairment, an AIS 2 concussion has an OIC of HWKB. If we assume an age of 30, putting the individual in the second age group (16-45), Hirsch's system predicts the following:

Category	*Period of Time*
1) Mobility	
Level 4	1 month
Level 3	5 months
Level 2	6 months
Level 1	0
2) Cognitive/psychological	
Level 4	2 months
Level 3	6 months
Level 2	4 months
Level 1	0

3) Neither Cosmetic, Sensory, nor Pain have any impairment for this injury.

4) Daily Living has 1 month at level 1.

Thus, in the first month following the injury the individual is predicted to have a severe (Level 4) mobility impairment, and for the first two months following the injury to have a severe cognitive/psychological impairment. As the injuries heal, the resulting impairment severity is reduced so that after 8 months (2+6) the cognitive impairment is at Level 2.

At the end of the year following the injury continuing through the fifth year after the injury, the impairment severity rating scheme predicts that the individual would have a level 1 mobility impairment, and a level 1 cognitive impairment. There would be no long term impairment resulting from this injury.

There are several points worth noting about this system. The impairment consequences were developed by a panel of three physicians, but only a single physician was used to code each injury. The system may therefore be biased by the individual physician's experience or preconceptions. One must be careful to approach the functional categories without preconceived notions. For example, brain injuries are responsible for high and long levels of mobility impairment. On the other hand, facial injuries result in short periods of cosmetic impairment, while lower leg injuries may result in long periods of cosmetic impairment (often due to the need for prostheses, or unnatural gaits). An ankle laceration may result in a period of mobility impairment (through all levels) lasting less than 7 weeks, but a permanent cosmetic impairment.

The Carsten-O'Day work attempted to reduce this complex system to a single total body impairment. This paper will not reduce the impairment to a single number but will instead present the various forms of impairment at the various levels for the various times. As part of their work to derive a single whole body impairment, Carsten and O'Day compared the importance of the impairment categories. The results of their analysis show that a given severity level in one impairment category is not equal to the same severity level in another impairment category. For example, a level 4 cognitive impairment resulted in a 95% whole body impairment under Carsten-O'Day's system, while a level 4 cosmetic impairment resulted in only a 10% whole body impairment. Thus, when considering the results discussed here, do not equate the consequences of different impairment categories that have the same severity level.

Method of Analysis

In the study discussed here Hirsch's impairment severity rating scheme was combined with NASS data from 1982, 1983, 1984, and 1985 to develop priorities for automotive safety programs. This study should not be used alone to prioritize such progams. All vehicle occupants who died as a result of the crash were excluded, because impairment does not deal with deaths. In addition, the Hirsch study may need further development before it can be fully used and understood. Early in the analysis performed the daily living impairment category yielded counter-intuitive or nonsensical results (such as no impairment in any other category, but a level 4 daily living impairment). Because of this, it was decided to exclude the daily living category from further consideration until it could be better understood.

Two types of priority ratings were developed. The first was based on vehicle components, and the second based on body regions. In each case the aim was to determine the percentage of the total impairment resulting from a particular crash scenario which was due to a particular body region or vehicle component. Cases were selected from NASS on an injury by injury basis where the age of the occupant was known, the AIS was between 2 and 5 and the occupant was not killed, and the occupants were in the front seat of a passenger car. All collision speeds were considered. These data were then divided into frontal collisions (impact directions of 11, 12 or 1 o'clock), and lateral collisions (impact directions of 2, 3, 4, 8, 9, or 10 o'clock). The vehicle components considered were, for frontal collisions, windshield, steering assembly, instrument panel, roof rails and A pillar, and the floor. For lateral collisions the vehicle components considered were side interior surfaces, side window glass, side hardware, the floor, and A, B pillars and roof rails. Side hardware consists of such things as the arm rest, window cranks, door opener, etc. The same body regions were used for both frontal and side collisions. The body regions studied were head and face, neck, thorax (back and chest), abdomen, pelvis (including the hip), upper extremity, and lower extremity. In some cases the NASS data contained an OIC which the Hirsch system did not rate. In these cases the injury was ignored, but other injuries that the occupant sustained were included. In other cases, the Hirsch system has several different impairments for the same OIC, AIS, and age group. In these cases, the average of the impairment ratings for that OIC, AIS, and age group were used.

Multiple Injuries

Occupants in automobile crashes often suffer more than one single injury with an AIS between 2-5. In determining priorities a problem arises from these multiple injuries. If the impairment from each injury is totaled we are saying five different injuries to one individual is no different, from total societal impairment view, than 5 different individuals each with one of these injuries. This view would also imply that if an occupant suffers two injuries the impairments are additive (i.e. if one injury results in a level 4 mobility impairment for 1 month, and another injury results in a level 4 mobility impairment of 2 months, than an injury basis impairment system would say that the individual had a level 4 mobility impairment for 3

Table 1. Example showing multiple injury algorithm

Impairment Category	Injury 1	Injury 2	Result
Mobility			
Level 1	2 Months	7 Months	5 Months
Level 2	6 Months	1 Month	3 Months
Level 3	1 Month	3 Months	3 Months
Level 4	1 Week	1 Month	1 Month

months). The approach used in this analysis assumes that with a particular impairment category (such as mobility, or cosmetic) the individual's impairment is the greatest level at a given time after the injuries were sustained. Table 1 and Figure 1 illustrate an individual who suffers two injuries which result in short term mobility impairments and how the individual's total impairment is determined with this algorithm.

This method is used to determine the resulting impairment to an individual who suffers multiple injuries. Multiple injuries create a second problem. When considering vehicle components as contributors to an injury, and if a particular component is isolated as a significant injury source, it is incorrect to assume that a 100% effective countermeasure would prevent all resulting impairment. Multiple injuries are often due to multiple injury sources. Thus, someone who hits their head on the windshield and the A pillar may not benefit from a 100% effective countermeasure for the A pillar because they would still suffer an injury from the windshield contact. The approach taken for this analysis started with a computation of the total impairment in all categories at all levels for a given impact direction (frontal or lateral). The previously discussed algorithm for assigning an impairment level to an individual with multiple injuries was used for all individuals within NASS included in the study. After the total societal impairment was calculated, the analysis was repeated excluding injuries due to a particular vehicle component or a particular body region. In this manner, it was possible to see what the total societal impairment would be if there was a 100% effective countermeasure for a particular vehicle component, or to prevent injury to a particular body region. With this type of analysis, an individual with multiple injuries could still have some impairment after removing their injuries due to a body region or vehicle component.

When interpreting the results the following should be noted. The smaller the total impairment after a component or body region has been removed, the

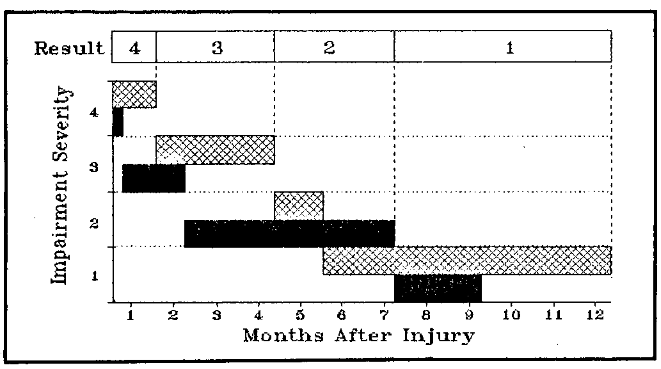

Figure 1. Example of multiple injury algorithm

higher priority it should receive. Because of multiple injuries there may be little or no reduction in the total societal impairment after removing a vehicle component or body region. In fact, there can be an increase in lower levels of impairment because of the multiple injury algorithm. For example, refer to Table 1. Assume that Injury 2 is caused by some vehicle component of interest, and will now be excluded from the total. The resulting impairment is now calculated as level 1-2 months; level 2-6 months; level 3-1 month; and level 4-1 week. While the level 3 and level 4 impairments have been reduced, the level 2 impairment has increased from 3 months to 6 months.

Results

Tables 2-9 show the results of the analyses performed. Tables 2-5 show the data for frontal collisions, while Tables 6-9 show results for lateral collisions. Each table represents the average of the 4 NASS years studied (1982, 1983, 1984, and 1985). The values reported from the accident data were multiplied by a scaling factor so that they are representative of the numbers encountered throughout the United States. Tables 2, 4, 6, and 8 are based on short term impairment, while Tables 3, 5, 7, and 9 are based on medium term impairment. In each table the number shown represents the percentage of the original total impairment for that category at that level that remains after all injuries due to the body region or vehicle component are removed. Thus, the lower the number, the greater the potential benefit from an injury counter-measure. For example, in Table 4 (short term impairment from frontal collisions based on body regions) it can be seen that after head and face injuries are removed from consideration, there is no level 4 cognitive impairment remaining. In contrast, 97% of the cosmetic level 1 impairment remains after removing all head and face injuries.

The values in these Tables are a function of several factors. An injury with drastic impairment consequences, but which seldom occurs will not account for a major portion of the total impairment. For similar reasons, a common injury with little impairment consequences also will not account for a major portion of the total impairment. As an aid in evaluating how commonly a vehicle component or body region is involved, at the base of each column is the average number of cases per year from NASS that were used. This number represents the number of cases left after the body region or vehicle component of interest has been removed. Thus, from Table 4 it can be seen that for frontal collisions there were an average of 470 cases per year in the NASS data. An average of 248 cases remained after injuries to the head and face were excluded. This means that 222 cases had injuries to only the head and face, with no other body region involved. It does not mean the 248

Table 2. Short term impairment

Frontal Collisions
Vehicle Components

Average of NASS 1982, 1983, 1984, 1985

		All Frontal (Days)	w/o Windshield Only	w/o Steering Assembly Only	w/o Roof Rails and A Pillar	w/o Instrument Panel	w/o Floor
Mobility	1	3851555	84%	87%	97%	82%	93%
	2	7855088	64%	91%	92%	90%	98%
	3	4323841	57%	93%	90%	94%	100%
	4	1443217	64%	93%	87%	92%	100%
Cognitive	1	1423513	64%	85%	94%	93%	99%
	2	4058703	54%	94%	90%	94%	100%
	3	6022842	55%	93%	88%	95%	100%
	4	2320142	57%	94%	87%	95%	100%
Cosmetic	1	5072054	99%	94%	99%	62%	86%
	2	2422463	95%	66%	100%	84%	98%
	3	1563482	86%	71%	98%	86%	87%
	4	842927	89%	73%	98%	74%	97%
Sensory	1	2835204	92%	61%	98%	91%	98%
	2	404596	75%	76%	92%	91%	100%
	3	150070	82%	74%	83%	95%	100%
	4	167125	74%	72%	88%	96%	100%
Pain	1	3296170	96%	76%	100%	77%	82%
	2	913841	88%	80%	97%	84%	85%
	3	219574	94%	79%	99%	79%	90%
	4	95638	92%	74%	98%	81%	95%
Number considered		470	371	414	454	407	448

remaining did not have a head or face injury, only that if they had such an injury, it was not included when determining the impairment.

It is tempting when examining Tables 2-9 to assess the percentages due to each item analyzed. This can lead to incorrect analyses due to multiple injury problems. For example, from Table 4 consider head/face injuries, and thoracic injuries and the resulting level 1 mobility impairments. While it is true that a 100% effective head/face injury countermeasure would reduce level 1 mobility impairment by 34% (because 56% of such impairment exists after all

Table 3. Medium term impairment

Frontal Collisions
Vehicle Components

Average of NASS 1982, 1983, 1984, 1985

	Category	All Frontal (Occupants)	w/o Windshield Only	w/o Steering Assembly Only	w/o Roof Rails and A Pillar	w/o Instrument Panel	w/o Floor
Mobility	1	40337	58%	91%	90%	93%	100%
	2	427	57%	93%	80%	100%	100%
	3	468	96%	100%	62%	95%	100%
	4	50	50%	100%	100%	100%	100%
Cognitive	1	36278	54%	93%	91%	94%	100%
	2	936	75%	96%	47%	100%	100%
	3	518	97%	100%	65%	95%	100%
	4	25	0%	100%	100%	100%	100%
Cosmetic	1	16967	96%	87%	100%	61%	90%
	2	4785	103%	53%	100%	93%	98%
	3	1156	50%	73%	93%	100%	100%
	4	279	58%	92%	100%	100%	100%
Sensory	1	6186	95%	60%	97%	94%	98%
	2	120	100%	100%	81%	100%	100%
	3	43	100%	100%	65%	100%	100%
	4	304	53%	93%	100%	100%	100%
Pain	1	2070	85%	70%	83%	92%	97%
	2	2024	93%	27%	100%	93%	96%
	3	0	NA	NA	NA	NA	NA
	4	0	NA	NA	NA	NA	NA
Number considered		470	371	407	454	407	448

Table 4. Short term impairment.

Frontal Collisions
Body Regions

Average of NASS 1982,1983,1984,1985

		w/o Head and Face	w/o Neck	w/o Thorax (Back&Chest)	w/o Abdomen	w/o Upper Extremity	w/o Pelvis (w/ Hip)	w/o Lower Extremity
Mobility	1	56%	99%	97%	99%	91%	99%	75%
	2	20%	100%	99%	100%	97%	99%	92%
	3	3%	100%	100%	100%	100%	100%	99%
	4	11%	100%	98%	100%	100%	97%	97%
Cognitive	1	14%	98%	91%	100%	100%	100%	100%
	2	0%	100%	100%	100%	100%	100%	100%
	3	0%	100%	100%	100%	100%	100%	100%
	4	0%	100%	100%	100%	100%	100%	100%
Cosmetic	1	97%	99%	103%	102%	83%	97%	45%
	2	78%	100%	87%	72%	85%	100%	94%
	3	61%	100%	86%	87%	102%	100%	74%
	4	72%	99%	91%	93%	75%	100%	75%
Sensory	1	67%	98%	82%	67%	97%	100%	100%
	2	4%	100%	98%	100%	100%	100%	100%
	3	6%	96%	99%	100%	100%	100%	100%
	4	10%	99%	91%	100%	100%	100%	100%
Pain	1	80%	99%	79%	100%	96%	99%	52%
	2	69%	100%	91%	96%	94%	88%	73%
	3	85%	99%	87%	96%	92%	97%	70%
	4	75%	100%	87%	93%	88%	77%	88%
Number considered		248	469	449	459	443	460	391

Total Cases - 470

head/face injuries are removed from consideration), it is not correct to say that thoracic injuries account for only 3% (100-97) of all mobility level 1 impairment. If an individual sustained a head injury and a thoracic injury, each with an associated level 1 mobility impairment, this individual would have been counted in both the column excluding head injury, and the column excluding thoracic injury. Even though a

Table 5. Medium term impairment.

Frontal Collisions
Body Regions

Average of NASS 1982,1983,1984,1985

	Category	w/o Head and Face	w/o Neck	w/o Thorax (Back&Chest)	w/o Abdomen	w/o Upper Extremity	w/o Pelvis (w/ Hip)	w/o Lower Extremity
Mobility	1	6%	100%	100%	100%	97%	100%	99%
	2	0%	100%	100%	100%	100%	100%	100%
	3	0%	100%	100%	100%	100%	100%	100%
	4	49%	100%	50%	100%	100%	100%	100%
Cognitive	1	0%	100%	100%	100%	100%	100%	100%
	2	0%	100%	100%	100%	100%	100%	100%
	3	0%	100%	100%	100%	100%	100%	100%
	4	0%	100%	100%	100%	100%	100%	100%
Cosmetic	1	87%	100%	102%	102%	75%	96%	55%
	2	103%	99%	70%	38%	100%	100%	100%
	3	0%	100%	100%	100%	100%	100%	100%
	4	9%	100%	91%	100%	100%	100%	100%
Sensory	1	82%	96%	77%	53%	100%	100%	100%
	2	0%	100%	100%	100%	100%	100%	100%
	3	0%	100%	100%	100%	100%	100%	100%
	4	8%	100%	92%	100%	100%	100%	100%
Pain	1	42%	97%	88%	100%	100%	100%	77%
	2	93%	100%	7%	100%	100%	100%	100%
	3	NA	NA	NA	NA	NA	NA	NA
	4	NA	NA	NA	NA	NA	NA	NA
Number considered		248	469	449	459	443	460	391

Total Cases - 470

Table 6. Short term impairment.

Side Collisions
Vehicle Components

Average of NASS 1982, 1983, 1984, 1985

		Total All Side (Days)	w/o Side Interior Surfaces	w/o Side Glass	w/o Side Hardware	w/o A, B Pillars Roof Rails	w/o Floor
Mobility	1	1287022	92%	94%	97%	88%	98%
	2	2937658	93%	90%	98%	89%	99%
	3	1627381	95%	88%	99%	87%	99%
	4	649104	91%	92%	96%	85%	100%
Cognitive	1	456730	94%	90%	98%	88%	100%
	2	1507317	96%	87%	100%	87%	100%
	3	2438695	96%	89%	99%	84%	100%
	4	982368	94%	89%	99%	85%	100%
Cosmetic	1	1321375	79%	97%	95%	97%	95%
	2	921043	73%	99%	89%	98%	99%
	3	605166	78%	100%	91%	99%	96%
	4	259758	83%	99%	93%	98%	100%
Sensory	1	1252127	73%	99%	87%	96%	99%
	2	119009	90%	97%	98%	83%	100%
	3	68812	95%	99%	99%	78%	100%
	4	18698	100%	96%	99%	89%	100%
Pain	1	1005591	65%	99%	94%	98%	95%
	2	402343	75%	99%	87%	93%	97%
	3	85698	80%	99%	93%	98%	97%
	4	45255	74%	99%	80%	98%	98%
Number considered		180	159	171	173	168	177

countermeasure may have prevented an injury, the individual may still have the same impairment due to another injury suffered during the same event.

Table 2 shows the short term impairment due to frontal collisions based on the following vehicle components as injury sources: windshield; steering assembly; roof rails and A pillars; instrument panel; and the floor. The windshield is injury source for much of the resulting impairment, particularly mobility, cognitive, and to a lesser degree sensory impairment. Surprisingly, the windshield is not a major contributor to lower levels of cosmetic impairment, nor is it the

Table 7. Medium term impairment.

Side Collisions
Vehicle Components

Average of NASS 1982, 1983, 1984, 1985

	Category	Total All Side (Occupants)	w/o Side Interior Surfaces	w/o Side Glass	w/o Side Hardware	w/o A, B Pillars Roof Rails	w/o Floor
Mobility	1	15430	96%	89%	98%	87%	100%
	2	69	100%	100%	100%	100%	100%
	3	389	75%	100%	100%	76%	100%
	4	0	NA	NA	NA	NA	NA
Cognitive	1	13916	97%	88%	99%	87%	100%
	2	647	92%	97%	100%	68%	100%
	3	389	75%	100%	100%	76%	100%
	4	0	NA	NA	NA	NA	NA
Cosmetic	1	3880	89%	98%	94%	98%	95%
	2	2623	72%	100%	85%	100%	99%
	3	99	100%	100%	100%	100%	100%
	4	26	100%	100%	100%	100%	100%
Sensory	1	2903	71%	100%	87%	99%	99%
	2	229	100%	100%	100%	74%	100%
	3	0	NA	NA	NA	NA	NA
	4	26	100%	100%	100%	100%	100%
Pain	1	845	90%	100%	89%	97%	100%
	2	689	74%	100%	88%	100%	100%
	3	0	NA	NA	NA	NA	NA
	4	0	NA	NA	NA	NA	NA
Number Considered		180	159	171	173	168	177

Table 8. Short term impairment.

Side Collisions
Body Regions

Average of NASS 1982, 1983, 1984, 1985

		w/o Head and Face	w/o Neck	w/o Thorax (Back&Chest)	w/o Abdomen	w/o Upper Extremity	w/o Pelvis (w/ Hip)	w/o Lower Extremity
Mobility	1	46%	100%	97%	97%	96%	99%	78%
	2	20%	99%	99%	99%	98%	97%	94%
	3	3%	100%	100%	100%	100%	100%	99%
	4	13%	100%	99%	99%	100%	94%	98%
Cognitive	1	15%	100%	91%	100%	100%	100%	100%
	2	2%	100%	100%	99%	100%	100%	100%
	3	0%	100%	100%	100%	100%	100%	100%
	4	0%	100%	100%	100%	100%	100%	100%
Cosmetic	1	98%	94%	104%	102%	84%	97%	65%
	2	88%	100%	86%	47%	94%	100%	99%
	3	87%	100%	82%	68%	100%	100%	70%
	4	89%	92%	88%	81%	84%	100%	67%
Sensory	1	80%	99%	84%	44%	99%	100%	100%
	2	5%	99%	98%	100%	100%	100%	100%
	3	20%	97%	99%	83%	100%	100%	100%
	4	9%	91%	100%	100%	100%	100%	100%
Pain	1	91%	97%	69%	99%	96%	97%	62%
	2	73%	99%	93%	92%	98%	69%	88%
	3	94%	95%	90%	93%	96%	89%	80%
	4	89%	100%	91%	90%	95%	50%	93%
Number considered		100	179	175	171	174	169	165

Total Cases - 180

leading cause of cosmetic impairment at higher levels. The steering assembly and instrument panel are major contributors to cosmetic impairment. Roof rails and A pillars are responsible for some serious cognitive and sensory impairment but are not the leading impairment source in either case. The steering assembly is the leading cause of sensory impairment, and together with the instrument panel is the cause of

Table 9. Medium term impairment.

Side Collisions
Body Regions

Average of NASS 1982, 1983, 1984, 1985

	Category	w/o Head and Face	w/o Neck	w/o Thorax (Back&Chest)	w/o Abdomen	w/o Upper Extremity	w/o Pelvis (w/ Hip)	w/o Lower Extremity
Mobility	1	6%	100%	100%	99%	99%	100%	96%
	2	0%	100%	100%	100%	100%	100%	100%
	3	0%	100%	100%	100%	100%	100%	100%
	4	NA	NA	NA	NA	NA	NA	NA
Cognitive	1	1%	100%	100%	99%	100%	100%	100%
	2	0%	100%	100%	100%	100%	100%	100%
	3	0%	100%	100%	100%	100%	100%	100%
	4	NA	NA	NA	NA	NA	NA	NA
Cosmetic	1	89%	92%	103%	101%	80%	96%	53%
	2	100%	100%	77%	26%	100%	100%	100%
	3	0%	100%	100%	100%	100%	100%	100%
	4	0%	100%	100%	100%	100%	100%	100%
Sensory	1	89%	98%	80%	38%	100%	100%	100%
	2	55%	100%	100%	45%	100%	100%	100%
	3	NA	NA	NA	NA	NA	NA	NA
	4	0%	100%	100%	100%	100%	100%	100%
Pain	1	93%	100%	64%	97%	100%	100%	52%
	2	100%	100%	0%	100%	100%	100%	100%
	3	NA	NA	NA	NA	NA	NA	NA
	4	NA	NA	NA	NA	NA	NA	NA
Number considered		100	179	175	171	174	169	165

Total Cases - 180

most pain impairment. Examination of the number of cases analyzed shows that the leading injury source was the windshield, following by the instrument panel, followed closely by the steering assembly. Neither roof rails and A pillars, nor the floor are major contributors to the number of cases of short term impairment from frontal collisions.

Table 3 shows a similar analysis for medium term impairment due to frontal collisions. As before, the windshield is the leading cause of mobility and cognitive medium term impairment. Roof rails and A pillars take on a much greater importance as causes of mobility and cognitive impairment. Both the instrument panel and the floor appear to have little influence on medium term impairment. Cosmetic impairment is due to the windshield and the steering assembly. Particularly for higher levels of cosmetic impairment, the windshield is the cause of half of the problems. The steering assembly is the major cause of medium term pain impairment.

Table 4 repeats the analysis for short term impairment due to frontal collisions, but now based on body regions. The message of Table 4 is that the overwhelming majority of impairment, particularly mobility, cognitive, and sensory, is due to injuries to the head and face. Indeed, 100% of the cognitive impairment is due to head injuries (as would be expected). Cosmetic impairment is due to the head and face, thorax, abdomen, and lower extremity. The lower extremity is the source of much of the cosmetic impairment (more than half of the level 1 cosmetic impairment). The lower extremity is also the source of much of the pain impairment resulting from frontal collisions. Surprisingly, neither the lower extremity, the pelvis, nor the upper extremity are major contributors to short term mobility impairment. Examination of the number of cases involved reveals that approximately half of all individuals injured in frontal crashes suffer injuries to only the head and face. A significant number of individuals also suffer injuries to only their lower extremities. No other body region seems to represent a major problem area in terms of the number of cases. It is worth repeating that a high percentage in the thorax column, for example, (meaning that removing thorax injuries does not significantly reduce the resulting impairment) does not mean that the thorax is not injured frequently. It may mean that when the thorax is injured, other body regions are also injured.

A similar distribution of body regions is found in Table 5, medium term impairment due to frontal collisions based on body regions. As with short term impairment, the head and face is the overwhelming majority of the problem. As might be expected all medium term cognitive impairment results from head and facial injuries, as well as almost all medium term mobility and sensory impairment. Interestingly, while the thorax does not account for any of the lower level mobility impairment, it does account for half of the level 4 mobility impairment. The head and face are responsible for virtually all level 3 and 4 medium term cosmetic impairment, but the thorax and abdomen account for almost all level 2 medium term cosmetic impairment, and the upper and lower extremities are responsible for the majority of the level 1 medium term cosmetic impairment. The head and face are responsible for more than half of the level 1 medium term pain impairment, with the lower extremity accounting for approximately 25% of level 1 medium term impairment. The thorax is responsible for virtually all medium term level 2 pain impairment due frontal collisions, and the thorax also accounts for a significant percentage of the level 1 pain impairment. There were no cases of level 3 or level 4 medium term pain impairment from frontal collisions in any of the 4 NASS years used.

The same type of analyses just described were repeated for lateral collisions. The vehicle components analyzed were side interior surfaces (interior door panels), side glass (the window in the door), side hardware (armrests, window cranks, door openers, etc.), the floor, and A, B pillars and roof rails. Based on these vehicle components, Table 6 shows the short term impairment due to lateral collisions. Examination of Table 6 reveals that there is no clear leading single cause of impairment in any category, implying that injuries due to lateral collisions might be the result of multiple injury sources. Of the vehicle components examined, the A, B pillars and roof rails are the leading cause of short term mobility impairment from lateral collisions, with side glass and side interior surfaces the next leading causes. For cognitive impairment the leading cause is also the A, B pillars and roof rails, with side glass a close second. The leading cause of short term cosmetic impairment resulting from lateral collisions is side interior surfaces, with the second leading cause being side hardware. Of the components studied short term sensory impairment resulting from lateral collisions at levels 2-4 is caused by A, B pillars and roof rails, and level 1 is caused by side interior surfaces. Short term pain impairment resulting from lateral collisions is caused by side interior surfaces. The floor does not appear to be a leading cause of impairment in lateral collisions.

Vehicle components as a cause of medium term impairment resulting from lateral collisions is shown in Table 7. Note that there were no injuries in any of the NASS years used which resulted in medium term impairments at cognitive level 4, sensory level 3, or pain level 3 or 4. Of the components examined the A, B pillars and roof rails were responsible for much of the mobility and cognitive impairments. Side interior surfaces were responsible for 25% of the level 3 mobility impairment, and 25% of the level 3 cognitive impairment. Side interior surfaces and side hardware were responsible for much of the cosmetic and pain impairment.

Examination of Tables 6 and 7 reveals that significant amounts of impairment result from something other than the components studied. The components selected were those that in the opinion of the authors represented the major injury sources in lateral collisions. Two possibilities exist to explain the low percentage of impairment caused by these components. One possibility is that there were multiple injury sources in many of the lateral collisions. Thus, removing one component would still leave many injuries resulting from other vehicle components. This is particularly true when the fact that side hardware, side interior surfaces, and side glass are all part of the door is considered. The second possible explanation is that significant amounts of impairment result from other vehicle components. To evaluate this second possibility, the distribution of injury sources from lateral collisions in the NASS 1984 data set was considered. This analysis was done on an injury basis rather than an occupant basis. In 36% of the cases studied the accident investigator was unable to determine the source of the injury. In addition, in 21% of the cases injuries were due to components that were viewed as frontal collision injury sources. Among these components are the windshield (6.9%), the rear view mirror (3.3%), the steering assembly (4.6%), and the instrument panel (6.6%). For comparison, of the vehicle components studied in lateral collisions, side interior surfaces were the injury source in 16.1% of the injuries, side hardware in 4.3% of the cases, side window glass in 4.9% of the cases, and A, and B pillars and roof rails in 9.9% of the cases. While the side analysis was not repeated to determine the impairment from lateral collisions where these "frontal" components were the injury source, it is clear that any future analysis must include these "frontal" components.

Problems with selection of the wrong components do not exist in Tables 8 and 9 which represent the impairment resulting from lateral collisions based on body regions. The same body regions as were used for frontal impact are used again. Table 8 shows the short term impairment, while Table 9 shows the medium term impairment. Table 8 shows that, as with frontal impact, injuries to the head and face are responsible for the overwhelming majority of impairment. In part this is due to the frequency with which they are injured. On the average 180 cases per year are placed in NASS due to lateral collisions. Of these 180, 80 are injuries to only the head and face, with no other body region injured. Virtually all short term cognitive impairment results from head and face injuries, while the majority of the short term mobility and sensory impairment from lateral collisions is due to injuries to the head and face. Cosmetic impairment is a mix of lower extremity, abdomen, thorax, upper extremity, and head and face. Pain impairment arises from injuries to the pelvis, the lower extremities, and the head and face.

Table 9 shows the medium term impairment, based on body region, which results from lateral collisions. The dominance of the head and face as the most important body region is shown in that virtually 100% of the mobility and cognitive impairment, as well as 100% of the level 3 and 4 cosmetic, and level 4 sensory impairment are due to head and face injuries. The thorax and the lower extremities are responsible for most of the pain impairment, and the abdomen, and head and face are responsible for medium term sensory impairment. Almost half of all people injured in lateral impacts recorded in the NASS files suffered injuries to only the head and face.

Frontal Head and Face Injury Sources

The preceding analysis showed the great importance of the head and face as the body region most in need of protection. Were a program being developed to prevent head and face injuries, the first question would be which vehicle component should be redesigned. Such an analysis was performed to examine the cause of head and face impairment resulting from frontal crashes. The same "frontal" components used previously were selected, but the data set was now limited to head and face injuries. Tables 10 and 11 show the percentages of total head and face impairment from frontal crashes arising from the vehicle components of interest. These numbers are averages of the 4 NASS years examined. The greater the percentage numbers in Tables 10 and 11, the more important that vehicle component is as a source of impairment. In addition, Tables 10 and 11 also have the percentage of impairment due to head and face injuries caused by vehicle components not examined (in the column labeled Other). Table 10 shows the results for short term impairment, while Table 11 shows the results for medium term impairment.

Table 10 shows that approximately one third of all impairment from injuries to the head and face due to frontal collisions has an injury source other than windshield, steering assembly, roof rails and A pillar, instrument panel, or floor. The floor, as would be expected is not a significant contributor to impairment from head and face injury. Of the vehicle components examined, the windshield is the most important injury source for short term mobility and cognitive impairment. Roof rails and A pillars are also a significant source of short term mobility and cognitive impair-

Table 10. Short term impairment to head and face frontal collisions.

Average of NASS 1982, 1983, 1984, 1985

	Category	All Frontal (Days)	Windshield	Steering Assembly	Roof Rails and A Pillar	Instrument Panel	Floor	Other
	1	2092842	28%	15%	9%	5%	2%	41%
Mobility	2	6545104	45%	8%	10%	5%	1%	31%
	3	4215064	45%	7%	11%	5%	1%	32%
	4	1322693	41%	7%	14%	4%	1%	33%
	1	1265907	43%	7%	7%	6%	2%	35%
Cognitive	2	4058703	46%	6%	10%	5%	1%	33%
	3	6022842	45%	7%	12%	4%	1%	31%
	4	2319677	43%	6%	13%	4%	1%	33%
	1	292771	18%	47%	6%	6%	3%	20%
Cosmetic	2	632024	25%	46%	3%	10%	4%	12%
	3	667625	37%	31%	6%	8%	2%	15%
	4	276442	38%	30%	3%	5%	1%	22%
	1	1057599	21%	26%	6%	7%	3%	37%
Sensory	2	1170655	8%	8%	3%	2%	1%	78%
	3	141928	18%	23%	17%	4%	1%	36%
	4	155235	42%	15%	6%	1%	0%	35%
	1	561580	20%	49%	5%	8%	2%	16%
Pain	2	312217	37%	23%	11%	3%	1%	25%
	3	40605	28%	36%	7%	6%	2%	21%
	4	26363	30%	31%	12%	4%	3%	22%
Number considered		278	123	36	21	11	4	78

ment. Short term cosmetic, sensory, and pain impairment is due to the steering asembly, the windshield, and to a lesser extent roof rails and A pillars.

Medium term impairment from head and face injuries is shown in Table 11. As with short term impairment, injury sources other than those studied account for approximately one third of all impairment. The windshield continues as the leading source of mobility and cognitive impairment, but for the medium term roof rails and A pillars contribute greater percentages to mobility and cognitive impairment than they did for short term impairment. Medium term cosmetic impairment is due to the windshield and steering assembly. Roof rails and A pillars are a leading cause of medium term sensory impairment, though the windshield is the major source of level 4 medium term sensory impairment. The steering assembly and windshield are also significant contributors to level 1 medium term sensory impairment. Level 1 medium term pain impairment is

Table 11. Medium term impairment to head and face frontal collisions.

Average of NASS 1982, 1983, 1984, 1985

	Category	All Frontal (Occupants)	Windshield	Steering Assembly	Roof Rails and A Pillar	Instrument Panel	Floor	Other
	1	38362	45%	9%	10%	5%	0%	31%
Mobility	2	427	43%	7%	19%	0%	0%	30%
	3	468	3%	0%	38%	5%	0%	53%
	4	25	100%	0%	0%	0%	0%	0%
	1	36164	46%	6%	9%	6%	0%	32%
Cognitive	2	936	25%	4%	53%	0%	0%	18%
	3	518	3%	0%	34%	5%	0%	58%
	4	25	100%	0%	0%	0%	0%	0%
	1	2561	24%	46%	1%	19%	0%	10%
Cosmetic	2	22	0%	0%	100%	0%	0%	0%
	3	1156	50%	27%	7%	0%	0%	16%
	4	254	46%	9%	0%	0%	0%	46%
	1	1275	18%	21%	19%	2%	0%	40%
Sensory	2	120	0%	0%	19%	0%	0%	81%
	3	43	0%	0%	35%	0%	0%	65%
	4	279	51%	8%	0%	0%	0%	42%
	1	1220	20%	40%	28%	6%	0%	4%
Pain	2	143	100%	0%	0%	0%	0%	0%
	3	NA	NA	NA	NA	NA	NA	NA
	4	NA	NA	NA	NA	NA	NA	NA
Number considered		278	123	36	21	15	0	83

caused by the windshield, steering assembly, and roof rails and A pillars, while all level 2 medium term pain impairment is due to the windshield.

Discussion of Results

In the preceding analyses, there is often one component or body region which stands out as the major source of impairment for that impact direction and impairment term. For example, in frontal collisions, the windshield is the leading cause of both short and medium term impairment. In all situations studied the head and face were the most important body region to protect. No other body region comes close to the head and face as a source of impairment. In both cases, the windshield and the head and face, the predominance of these items is due largely to the frequency with which they are involved. Approximately half of all cases examined resulted in an injury to only the head and face. The percentage is even higher if cases where injury occurred to another body region as well as the head and face are considered. Thus, using impairment with NASS data to prioritize crash safety issues tends to rank issues in terms of how frequently an injury occurs, as well as how serious the injury is.

The analysis of vehicle components as injury sources revealed that many of the components not viewed as injury sources for a particular impact direction are important, and must be considered. In side impacts, it was found that the front windshield was the injury source in a greater percentage of the cases than the side window glass, or the side hardware. This partly explains the high incidence of other as the injury source in the analyses performed. However, another contributing factor was multiple injuries. Eliminating one injury source, such as the side glass did not eliminate all injuries. The implication of these two factors (non-traditional injury sources for a particular impact direction, and the multiple injury problem) may be that systems tests provide more reliable information than component tests provide. A systems tests is more likely to detect all injury sources, including those from non-traditional sources (such as the windshield in a side impact), and multiple injury sources.

One of the issues evaluated is which vehicle components are responsible for impairment resulting from automobile crashes. The windshield was revealed as the leading contributor to impairment resulting from frontal crashes. This study did not analyze the practicality of modifying a vehicle component in an attempt to reduce resulting impairment. It may be that windshield glass is already optimized to the maximum extent practical with respect to crash safety performance. However, the windshield is a worthy subject for further development of injury countermeasures. In addition, there are other vehicle components whose contribution to total impairment was smaller, but which can be readily modified to increase crash safety performance.

The A pillar and roof rail component is an interesting study. In frontal collisions the contribution to short term impairment of the A pillar and roof rail component, while not insignificant, is not major. When the medium term is considered, the importance of the A pillar and roof rail component is greater. This trend can also be found with A, B pillar/roof rails in side impacts. The explanation of this finding may be that injuries resulting from these vehicle components are not common, but tend to be severe. Thus, for short term impairment the A pillar/roof rail contribution to the total is overwhelmed by other more common but less severe injuries. Because these other injuries may be less severe, recovery is faster and less permanent impairment results. A pillar/roof rail impacts may be more severe resulting in injuries which, while they are less common, have a poorer recovery prognosis. Thus, individuals who have medium term impairment (implying their original injury was severe and had a poor recovery prognosis) may be more likely to have received their injury from an A pillar/roof rail impact. In contrast, the instrument panel contributes to short term cosmetic and pain impairment, but is not a significant source of any impairment for the medium term. This implies that injuries from the instrument panel have a good prognosis for recovery.

Improvements Needed to the Impairment Rating System

The system used for evaluating the impairment consequences of an injury was the result of Hirsch's first attempt to create such a system. The system made it possible, given an OIC, AIS, and age of an injured occupant, to determine the impairment consequences of the injury. This study revealed several shortcomings in the system, and several areas in need of further development. Specific problems such as several different impairment consequences for the same OIC, AIS and age combination, or OIC's missing from Hirsch's system have already been mentioned. It has also been mentioned that each injury was coded by only one physician, and thus may reflect that individual's own bias. These are problems which could be readily addressed by more development of an impairment rating scheme. Other problems found require more consideration.

The Daily Living category often yielded counterintuitive results. This problem may have developed because of poorly understood definitions of this impairment category on the part of either users or developers of the impairment severity rating scheme. In other instances the impairment ratings resulted in confusing results. For example, an ankle laceration injury was considered. This injury resulted in limited short term mobility impairment, but in severe, longer term cosmetic injury. At periods of time after the injury when there was no mobility impairment (or any other impairment other than cosmetic) remaining, the injury was still coded as having serious cosmetic impairment. While an ankle laceration might cause some disfigurement, it would be readily covered by clothing. If the injury does not impair mobility with an upset gait or some other such problem, it seems unlikely that serious cosmetic impairment would remain.

These examples are cited not as specific criticisms, but rather an examples of problems with Hirsch's system as developed. The need exists to improve the system. Particular problems such as multiple OIC's, or missing OIC's should be corrected. The use of several physicians to code each injury would remove possible bias, as well as illustrate cases where definitions are in need of clarification. In addition, attempts to validate the predictions of impairment would be useful. Two possible methods of doing this would be to include a prognosis variable in future NASS surveys, and a study by a Health Maintenance Organization (HMO), or similar organization with ready access to long term medical records of injured individuals.

Conclusions

Hirsch's impairment severity rating scheme was used with NASS data from 1982-1985 to prioritize crash safety issues in frontal and side impacts. Two types of priorities were developed, one based on body region, and one based on vehicle components. Hirsch's system, while in need of further development, allows one to use an OIC, AIS, and age to predict the impairment consequences of an injury.

In both frontal and side impacts, for both short term (less than a year) and medium term (1-5 years) periods, injuries to the head and face are the overwhelming source of impairment due to automobile crashes. In frontal collisions the windshield is the leading cause of impairment, with the steering assembly, and A pillar/roof rails also causing significant proportions of the resulting impairment. In lateral collisions, much of the impairment is caused by injury sources which have not traditionally been considered side impact problems such as the windshield and steering assembly. Because of non-traditional injury sources, and because of multiple injuries, systems tests may provide more reliable information on safety benefits than component tests provide. Of the vehicle components studied for lateral impacts, side interior surfaces, and A, B pillars and roof rails were the leading sources of impairment. The floor is not a significant source of injury in either frontal or lateral collisions.

A pillar and roof rails are more important sources of medium term impairment than short term impairment. This may be due to less common, but more severe injuries associated with these components. This trend is also observed with the steering assembly. While the instrument panel contributes to short term impairment, it is not a significant source of any medium term impairment.

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Disclaimer

The opinions expressed in this paper are those of the authors and are not necessarily those of the National Highway Traffic Safety Administration, United States Department of Transportation.

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Relationship Between Mechanical Input and Injury Severity in Side Impact Tests

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Abstract

Results from 58 side impact tests with Postmortem Human Subjects (PMHS) and 15 side impact tests with dummies are reported. Based on this test series the relationships between mechanical inputs and injury severity as well as mechanical inputs and dummy loadings are analyzed.

The test configuration was identical for all side impacts: 90° impact angle, Opel Kadett body in white, CCMC deformable element in front of the impacting barrier. The PMHS were subjected to test runs at impact speeds of 40 km/h, 45 km/h, 50 km/h and 60 km/h. The dummy tests (5 HSRI-SID, 5 Hybrid II, 5 APROD) were performed at 50 km/h.

Under these test configurations head injuries were observed in 8 cases at impact speeds of at least 45 km/h.

Rib fractures were the most frequent injuries. The number of rib fractures was primarily influenced by the age of the subject.

Due to the asymmetry of the abdomen the severity of abdominal injuries is strongly influenced by the impacted body side. In right-side impacts liver ruptures were observed in almost 50% of the 45 km/h collisions and in nearly all the 50 km/h impacts. During left-side impacts, liver ruptures occurred in several of the 50 km/h tests and in almost half of the 60 km/h collisions. Spleen ruptures were observed in 4 cases. Spinal injuries were observed in 43 tests. Severity was rated AIS 1 or 2 except in 3 cases.

Pelvic fractures occurring at test speeds of 45 km/h or higher were found in 4 tests.

For statistical analysis, logistic regression was used. The purpose of this evaluation was to find the relationship between mechanical and anthropometrical parameters on the one hand and injury severity on the other.

The prediction of thorax AIS probability was primarily influenced by age. The rib accelerations on the impacted as well as on the opposite side were found to be important.

As far as the prediction of abdomen AIS probability is concerned, age is of no importance. The body mass and the accelerations at the sternum and the 12th thoracic vertebra produced good prediction results.

Regarding the biofidelity of the dummies, some results of the thorax agreed well with the PMHS response.

Objective

The goals of the research project were

a) an analysis of the relationships between mechanical inputs in a 90° side impact and the injuries to a belt restrained occupant at the impacted side in order to describe the mechanics of injury and associated kinematics

b) the establishment of load limits as a contribution to the definition of protection criteria

c) the comparison of measured data for PMHS and dummies in a 90° side collision for the characterization of dummy biofidelity.

Procedure

In order to achieve these goals, side collision tests were conducted under defined conditions:

- 90° side impact with a movable deformable barrier (CCMC element) against a stationary Opel Kadett carbody
- Impact speeds of 40, 45, 50 and 60 km/h
- Belt restrained occupant at the impacted side
- Test objects PMHS and 50 percentile male dummies (HSRI SID, APROD, HYBRID II)

Test Conditions

The test parameters, impacting vehicle, impacted vehicle, impact zone, test objects, instrumentation, optical documentation and first results have already been described in /1,2,3,4/.

Test Groups and Test Matrix

Figure 1 provides an overview of the test groups according to test object, collision speed and direction of impact. Except for age, the PMHS were allocated to the test groups (speed/impact side) at random, i.e. according to availability, without regard to anthropometry. The goal for each group was to achieve the same distribution of age.

Statistical Evaluation

A principal objective of this research project was to derive the relationship between mechanical input, anthropometric data and the probability of injury severity of PMHS described by discrete AIS distribution.

Test Object	Collision Speed	Number of Tests Impact Side	
		Right	Left
PMHS	40km/h 45km/h 50km/h 60km/h	5 10 12 0	4 10 12 5
HSRI-SID HYBRID II APROD	50km/h	2 3 3	3 2 3
Total Number of Tests		35	38

Figure 1. Test matrix for all tests with impact speed and direction

For this goal, statistical evaluations were performed based on the test results using the logistic regression method /5/.

Test Results

Vehicle Deformation

Deformation profiles on the exterior of the impacted vehicle:

The permanent deformation to the vehicle side structures in the upper, central and lower series of measuring points with reference to the individual speed groups is shown in Figure 2. The deformation profiles for the individual speed groups correspond to the average deformation from several individual tests. It can be observed that deformation increases with impact speed at all three measuring-point levels.

Loadings on the Test Objects

A selection of the measured values for dummies and PMHS is shown in Figures 3 and 4. In making the selection, those variables were taken into account which were expected to play an important role as predictors in statistical evaluation.

Dummy Loadings

In a comparison of the range of, e.g., maximum accelerations of the dummies (Figure 3), varying degrees of reproducibility of the measured values at the individual measuring points are apparent. For example, the range of variation of the values for maximum accelerations at the second rib (ROSYM) of the HSRI-SID in an impact from the left has the remarkably low value of only 1 g (three tests). However, the same dummy exhibits a range of variation of 89 g with a mean value of 199 g at the 12th thoracic vertebra, 3 ms value in the Y direction (T12Y3) during impact from the left.

In a comparison between the dummies there is good partial agreement between the measured values, but also great variation, depending on the location of accelerometer.

On the whole, the standard deviations are lower in the HSRI-SID than in the other two dummies.

PMHS Loadings

The measurements relevant for statistical evaluations are given in Figure 4.

PMHS have a relatively great variation of maximum accelerations at the individual measuring points, attributable to the variations of anthropometry.

PMHS Injuries

- *Head injuries*

Head injuries were found in 8 of 58 tests. Of these 5 injuries were noted in left-side and 3 injuries in right-side impacts. Figure 5 presents data regarding the tests in which head injuries occurred. Head contact occurs in only three tests.

The low incidence of head injury is possibly attributable to pre-existing damage to the clivus, a weak

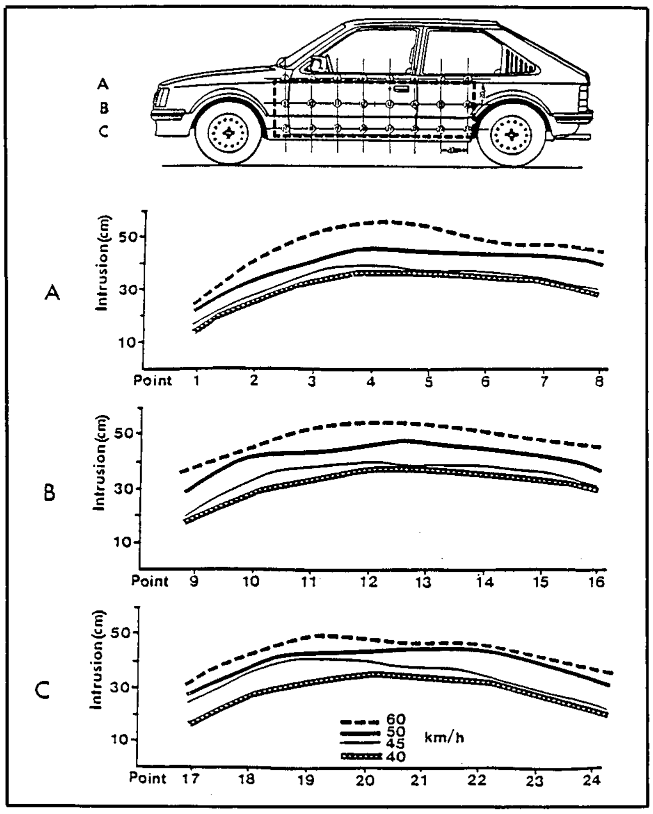


Figure 2. Permanent deformation of the impact zone in 3 horizontal planes (as shown in the illustration at top), arranged according to speed group (9 tests at 40 km/h, 14 tests at 45 km/h, 21 tests at 50 km/h and 3 tests at 60 km/h. In speed groups 40, 45 and 50 km/h, impacts from right and left, in 60 km/h group impacts from left only)

VARIABLE	LABEL	N	MEAN	MINIMUM VALUE	MAXIMUM VALUE	STANDARD DEVIATION
COLLISION TYPE=R	OBJECT=1					
ROSYM	ACC.UPP.RIB IMPACT SID.MAX. (G)	0	.	.	.	.
ROSY3	ACC.UPP.RIB IMPACT SID.3MS. (G)	0	.	.	.	.
RUSYM	ACC.LOW.RIB IMP.SIDE MAX. (G)	2	123.0	120.0	126.0	4.2
RUSY3	ACC.LOW.RIB IMP.SIDE 3MS. (G)	2	95.0	85.0	105.0	14.1
ROGYM	ACC.UPP.RIB OFFSID.IMP.MAX. (G)	2	107.0	105.0	109.0	2.8
ROGY3	ACC.UPP.RIB OFFSID.IMP.3MS. (G)	2	88.5	86.0	91.0	3.5
RUGYM	ACC.LOW.RIB OFFSID.IMP.MAX. (G)	2	132.5	129.0	136.0	4.9
RUGY3	ACC.LOW.RIB OFFSID.IMP.3MS. (G)	2	108.5	103.0	114.0	7.8
T1Y3	Y-ACC.THOR.SPINE 1 3MS. (G)	1	82.0	82.0	82.0	.
T12Y3	Y-ACC.THOR.SPINE 12 3MS (G)	1	59.0	59.0	59.0	.
KSY3	Y-ACC. HEAD IMP. SIDE 3MS (G)	2	66.5	50.0	83.0	23.3
KGY3	Y-ACC. HEAD OFFSIDE IMP. 3MS. (G)	2	34.5	28.0	41.0	9.2
COLLISION TYPE=R	OBJECT=2					
ROSYM	ACC.UPP.RIB IMPACT SID.MAX. (G)	2	112.0	96.0	128.0	22.6
ROSY3	ACC.UPP.RIB IMPACT SID.3MS. (G)	2	53.0	47.0	59.0	8.5
RUSYM	ACC.LOW.RIB IMP.SIDE MAX. (G)	2	181.0	104.0	258.0	108.9
RUSY3	ACC.LOW.RIB IMP.SIDE 3MS. (G)	2	109.0	66.0	152.0	60.8
ROGYM	ACC.UPP.RIB OFFSID.IMP.MAX. (G)	2	88.5	75.0	102.0	19.1
ROGY3	ACC.UPP.RIB OFFSID.IMP.3MS. (G)	2	55.0	50.0	60.0	7.1
RUGYM	ACC.LOW.RIB OFFSID.IMP.MAX. (G)	2	139.5	103.0	176.0	51.6
RUGY3	ACC.LOW.RIB OFFSID.IMP.3MS. (G)	2	132.0	91.0	173.0	58.0
T1Y3	Y-ACC.THOR.SPINE 1 3MS. (G)	2	48.0	41.0	55.0	9.9
T12Y3	Y-ACC.THOR.SPINE 12 3MS (G)	2	60.5	56.0	65.0	6.4
KSY3	Y-ACC. HEAD IMP. SIDE 3MS (G)	2	32.0	17.0	47.0	21.2
KGY3	Y-ACC. HEAD OFFSIDE IMP. 3MS. (G)	2	32.0	25.0	39.0	9.9
COLLISION TYPE=R	OBJECT=3					
ROSYM	ACC.UPP.RIB IMPACT SID.MAX. (G)	3	179.7	114.0	215.0	56.9
ROSY3	ACC.UPP.RIB IMPACT SID.3MS. (G)	3	95.0	76.0	123.0	24.8
RUSYM	ACC.LOW.RIB IMP.SIDE MAX. (G)	3	276.7	204.0	408.0	114.0
RUSY3	ACC.LOW.RIB IMP.SIDE 3MS. (G)	3	103.3	96.0	110.0	7.0
ROGYM	ACC.UPP.RIB OFFSID.IMP.MAX. (G)	3	103.7	75.0	132.0	28.5
ROGY3	ACC.UPP.RIB OFFSID.IMP.3MS. (G)	3	69.0	57.0	75.0	10.4
RUGYM	ACC.LOW.RIB OFFSID.IMP.MAX. (G)	3	112.7	87.0	144.0	28.9
RUGY3	ACC.LOW.RIB OFFSID.IMP.3MS. (G)	3	77.3	74.0	80.0	3.
T1Y3	Y-ACC.THOR.SPINE 1 3MS. (G)	3	58.3	55.0	64.0	4.
T12Y3	Y-ACC.THOR.SPINE 12 3MS (G)	3	83.3	70.0	105.0	18.
KSY3	Y-ACC. HEAD IMP. SIDE 3MS (G)	3	61.7	45.0	70.0	14.
KGY3	Y-ACC. HEAD OFFSIDE IMP. 3MS. (G)	3	25.7	17.0	30.0	7.

Figure 3. Dummy loadings according to impact speed (50 km/h), right side impact Object 1: HSRI SID, 2: Hybrid II, 3: APROD

VARIABLE	LABEL	N	MEAN	MINIMUM VALUE	MAXIMUM VALUE	STANDARD DEVIATION
COLLISION TYPE=L	OBJECT=1					
ROSYM	ACC.UPP.RIB IMPACT SID.MAX. (G)	3	123.7	123.0	124.0	0.6
ROSY3	ACC.UPP.RIB IMPACT SID.3MS. (G)	3	70.7	65.0	76.0	5.5
RUSYM	ACC.LOW.RIB IMP.SIDE MAX. (G)	3	105.0	95.0	110.0	8.7
RUSY3	ACC.LOW.RIB IMP.SIDE 3MS. (G)	3	84.0	83.0	85.0	1.0
ROGYM	ACC.UPP.RIB OFFSID.IMP.MAX. (G)	0	.	.	.	.
ROGY3	ACC.UPP.RIB OFFSID.IMP.3MS. (G)	0	.	.	.	.
RUGYM	ACC.LOW.RIB OFFSID.IMP.MAX. (G)	3	132.3	105.0	146.0	23.7
RUGY3	ACC.LOW.RIB OFFSID.IMP.3MS. (G)	3	112.7	84.0	127.0	24.8
T1Y3	Y-ACC.THOR.SPINE 1 3MS. (G)	3	85.0	82.0	89.0	3.6
T12Y3	Y-ACC.THOR.SPINE 12 3MS. (G)	2	199.5	155.0	244.0	62.9
KSY3	Y-ACC. HEAD IMP. SIDE 3MS. (G)	3	53.0	49.0	55.0	3.5
KGY3	Y-ACC. HEAD OFFSIDE IMP. 3MS. (G)	3	41.0	25.0	49.0	13.9
COLLISION TYPE=L	OBJECT=2					
ROSYM	ACC.UPP.RIB IMPACT SID.MAX. (G)	2	155.5	140.0	171.0	21.9
ROSY3	ACC.UPP.RIB IMPACT SID.3MS. (G)	2	83.0	77.0	89.0	8.5
RUSYM	ACC.LOW.RIB IMP.SIDE MAX. (G)	2	250.5	250.0	251.0	0.7
RUSY3	ACC.LOW.RIB IMP.SIDE 3MS. (G)	2	187.0	149.0	225.0	53.7
ROGYM	ACC.UPP.RIB OFFSID.IMP.MAX. (G)	2	130.0	112.0	148.0	25.5
ROGY3	ACC.UPP.RIB OFFSID.IMP.3MS. (G)	2	56.5	50.0	63.0	9.2
RUGYM	ACC.LOW.RIB OFFSID.IMP.MAX. (G)	2	100.5	90.0	111.0	14.8
RUGY3	ACC.LOW.RIB OFFSID.IMP.3MS. (G)	2	82.5	81.0	84.0	2.1
T1Y3	Y-ACC.THOR.SPINE 1 3MS. (G)	2	55.5	55.0	56.0	0.7
T12Y3	Y-ACC.THOR.SPINE 12 3MS. (G)	2	67.5	60.0	75.0	10.6
KSY3	Y-ACC. HEAD IMP. SIDE 3MS. (G)	2	50.0	36.0	64.0	19.8
KGY3	Y-ACC. HEAD OFFSIDE IMP. 3MS. (G)	2	27.0	23.0	31.0	5.7
COLLISION TYPE=L	OBJECT=3					
ROSYM	ACC.UPP.RIB IMPACT SID.MAX. (G)	2	181.5	171.0	192.0	14.8
ROSY3	ACC.UPP.RIB IMPACT SID.3MS. (G)	2	92.5	89.0	96.0	4.9
RUSYM	ACC.LOW.RIB IMP.SIDE MAX. (G)	2	362.0	357.0	367.0	7.1
RUSY3	ACC.LOW.RIB IMP.SIDE 3MS. (G)	2	185.0	149.0	221.0	50.9
ROGYM	ACC.UPP.RIB OFFSID.IMP.MAX. (G)	2	109.0	106.0	112.0	4.2
ROGY3	ACC.UPP.RIB OFFSID.IMP.3MS. (G)	2	79.5	73.0	86.0	9.2
RUGYM	ACC.LOW.RIB OFFSID.IMP.MAX. (G)	2	198.0	165.0	231.0	46.7
RUGY3	ACC.LOW.RIB OFFSID.IMP.3MS. (G)	2	115.5	96.0	135.0	27.6
T1Y3	Y-ACC.THOR.SPINE 1 3MS. (G)	2	70.0	63.0	77.0	9.9
T12Y3	Y-ACC.THOR.SPINE 12 3MS. (G)	2	108.5	108.0	109.0	0.7
KSY3	Y-ACC. HEAD IMP. SIDE 3MS. (G)	2	84.0	73.0	95.0	15.6
KGY3	Y-ACC. HEAD OFFSIDE IMP. 3MS. (G)	2	28.5	21.0	36.0	10.6

Figure 3a. Dummy loadings according to impact speed (50 km/h), left side impact Object 1: HSRI SID, 2: Hybrid II, 3: APROD

VARIABLE	LABEL	N	MEAN	MINIMUM VALUE	MAXIMUM VALUE	STANDARD DEVIATION
	VG=40					
ROSYM	ACC.UPP.RIB IMPACT SID.MAX. (G)	2	130.5	122.0	139.0	12.0
ROSY3	ACC.UPP.RIB IMPACT SID.3MS. (G)	2	66.5	64.0	69.0	3.5
RUSYM	ACC.LOW.RIB IMP.SIDE MAX. (G)	5	129.0	98.0	173.0	33.3
RUSY3	ACC.LOW.RIB IMP.SIDE 3MS. (G)	5	75.4	53.0	116.0	27.3
ROGYM	ACC.UPP.RIB OFFSID.IMP.MAX. (G)	5	63.0	51.0	86.0	14.9
ROGY3	ACC.UPP.RIB OFFSID.IMP.3MS. (G)	5	51.4	43.0	61.0	7.7
RUGYM	ACC.LOW.RIB OFFSID.IMP.MAX. (G)	5	61.8	44.0	82.0	14.3
RUGY3	ACC.LOW.RIB OFFSID.IMP.3MS. (G)	5	54.0	40.0	65.0	10.1
T1Y3	Y-ACC.THOR.SPINE 1 3MS. (G)	5	59.0	43.0	75.0	13.4
T12Y3	Y-ACC.THOR.SPINE 12 3MS (G)	5	60.8	44.0	80.0	12.8
KSY3	Y-ACC HEAD IMP SIDE 3MS (G)	5	53.6	44.0	67.0	9.4
KGY3	Y-ACC HEAD OFFSIDE IMP. 3MS. (G)	5	32.2	27.0	40.0	4.9
	VG=45					
ROSYM	ACC.UPP.RIB IMPACT SID.MAX. (G)	10	119.8	64.0	178.0	31.2
ROSY3	ACC.UPP.RIB IMPACT SID.3MS. (G)	10	75.1	38.0	109.0	20.8
RUSYM	ACC.LOW.RIB IMP.SIDE MAX. (G)	10	189.9	110.0	256.0	42.3
RUSY3	ACC.LOW.RIB IMP.SIDE . 3MS. (G)	10	105.3	80.0	139.0	18.6
ROGYM	ACC.UPP.RIB OFFSID.IMP.MAX. (G)	10	78.5	43.0	109.0	23.3
ROGY3	ACC.UPP.RIB OFFSID.IMP.3MS. (G)	10	58.9	40.0	75.0	11.5
RUGYM	ACC.LOW.RIB OFFSID.IMP.MAX. (G)	10	70.4	45.0	121.0	21.5
RUGY3	ACC.LOW.RIB OFFSID.IMP.3MS. (G)	10	59.7	43.0	79.0	12.2
T1Y3	Y-ACC.THOR.SPINE 1 3MS. (G)	10	63.9	43.0	88.0	14.1
T12Y3	Y-ACC.THOR.SPINE 12 3MS (G)	10	70.0	56.0	90.0	10.7
KSY3	Y-ACC HEAD IMP SIDE 3MS (G)	10	52.2	41.0	62.0	7.4
KGY3	Y-ACC HEAD OFFSIDE IMP. 3MS. (G)	10	35.1	20.0	49.0	9.5
	VG=50					
ROSYM	ACC.UPP.RIB IMPACT SID.MAX. (G)	9	134.7	46.0	244.0	71.2
ROSY3	ACC.UPP.RIB IMPACT SID.3MS. (G)	9	85.3	43.0	140.0	33.8
RUSYM	ACC.LOW.RIB IMP.SIDE MAX. (G)	12	202.2	138.0	271.0	48.9
RUSY3	ACC.LOW.RIB IMP.SIDE 3MS. (G)	12	113.8	92.0	161.0	22.6
ROGYM	ACC.UPP.RIB OFFSID.IMP.MAX. (G)	12	100.5	62.0	164.0	32.9
ROGY3	ACC.UPP.RIB OFFSID.IMP.3MS. (G)	12	77.4	51.0	102.0	16.4
RUGYM	ACC.LOW.RIB OFFSID.IMP.MAX. (G)	12	99.8	51.0	156.0	32.5
RUGY3	ACC.LOW.RIB OFFSID.IMP.3MS. (G)	12	80.8	48.0	111.0	19.5
T1Y3	Y-ACC.THOR.SPINE 1 3MS. (G)	12	71.1	54.0	102.0	14.8
T12Y3	Y-ACC.THOR.SPINE 12 3MS (G)	12	78.8	61.0	97.0	12.9
KSY3	Y-ACC HEAD IMP SIDE 3MS (G)	12	63.3	37.0	87.0	15.1
KGY3	Y-ACC HEAD OFFSIDE IMP. 3MS. (G)	12	43.1	22.0	62.0	13.0

Figure 4. PMHS loadings according to impact speed, right side impact

VARIABLE	LABEL	N	MEAN	MINIMUM VALUE	MAXIMUM VALUE	STANDARD DEVIATION
	VG=40					
ROSYM	ACC.UPP.RIB IMPACT SID.MAX. (G)	4	73.0	62.0	81.0	8.4
ROSY3	ACC.UPP.RIB IMPACT SID.3MS. (G)	4	50.3	39.0	59.0	8.4
RUSYM	ACC.LOW.RIB IMP.SIDE MAX. (G)	3	177.0	122.0	224.0	51.5
RUSY3	ACC.LOW.RIB IMP.SIDE 3MS. (G)	3	77.0	66.0	90.0	12.1
ROGYM	ACC.UPP.RIB OFFSID.IMP.MAX. (G)	4	57.8	48.0	70.0	9.3
ROGY3	ACC.UPP.RIB OFFSID.IMP.3MS. (G)	4	51.5	45.0	60.0	6.4
RUGYM	ACC.LOW.RIB OFFSID.IMP.MAX. (G)	4	59.0	48.0	72.0	9.9
RUGY3	ACC.LOW.RIB OFFSID.IMP.3MS. (G)	4	50.8	42.0	64.0	9.4
T1Y3	Y-ACC.THOR.SPINE 1 3MS. (G)	4	43.0	34.0	52.0	7.3
T12Y3	Y-ACC.THOR.SPINE 12 3MS (G)	4	64.0	51.0	77.0	12.9
KSY3	Y-ACC. HEAD IMP.SIDE 3MS (G)	4	49.3	40.0	66.0	11.6
KGY3	Y-ACC. HEAD OFFSIDE IMP. 3MS. (G)	4	36.5	22.0	66.0	20.1
	VG=45					
ROSYM	ACC.UPP.RIB IMPACT SID.MAX. (G)	10	135.4	67.0	213.0	45.2
ROSY3	ACC.UPP.RIB IMPACT SID.3MS. (G)	10	78.5	44.0	133.0	30.5
RUSYM	ACC.LOW.RIB IMP.SIDE MAX. (G)	10	192.5	89.0	300.0	68.1
RUSY3	ACC.LOW.RIB IMP.SIDE 3MS. (G)	10	90.7	65.0	128.0	17.4
ROGYM	ACC.UPP.RIB OFFSID.IMP.MAX. (G)	9	89.0	47.0	160.0	35.6
ROGY3	ACC.UPP.RIB OFFSID.IMP.3MS. (G)	9	65.3	44.0	85.0	15.8
RUGYM	ACC.LOW.RIB OFFSID.IMP.MAX. (G)	10	90.1	44.0	128.0	27.3
RUGY3	ACC.LOW.RIB OFFSID.IMP.3MS. (G)	10	74.2	42.0	92.0	17.3
T1Y3	Y-ACC.THOR.SPINE 1 3MS. (G)	10	65.2	41.0	102.0	18.7
T12Y3	Y-ACC.THOR.SPINE 12 3MS (G)	10	78.9	57.0	104.0	16.9
KSY3	Y-ACC. HEAD IMP.SIDE 3MS (G)	9	48.4	39.0	70.0	8.9
KGY3	Y-ACC. HEAD OFFSIDE IMP. 3MS. (G)	10	36.7	21.0	51.0	12.0
	VG=50					
ROSYM	ACC.UPP.RIB IMPACT SID.MAX. (G)	12	145.6	89.0	239.0	47.6
ROSY3	ACC.UPP.RIB IMPACT SID.3MS. (G)	12	104.9	58.0	183.0	32.5
RUSYM	ACC.LOW.RIB IMP.SIDE MAX. (G)	12	224.4	121.0	337.0	64.0
RUSY3	ACC.LOW.RIB IMP.SIDE 3MS. (G)	12	126.2	83.0	206.0	35.3
ROGYM	ACC.UPP.RIB OFFSID.IMP.MAX. (G)	11	120.9	86.0	174.0	27.1
ROGY3	ACC.UPP.RIB OFFSID.IMP.3MS. (G)	11	87.6	68.0	132.0	17.3
RUGYM	ACC.LOW.RIB OFFSID.IMP.MAX. (G)	12	121.8	73.0	184.0	31.6
RUGY3	ACC.LOW.RIB OFFSID.IMP.3MS. (G)	12	101.6	60.0	137.0	23.1
T1Y3	Y-ACC.THOR.SPINE 1 3MS. (G)	12	87.3	52.0	173.0	31.0
T12Y3	Y-ACC.THOR.SPINE 12 3MS (G)	12	100.2	71.0	138.0	19.2
KSY3	Y-ACC. HEAD IMP.SIDE 3MS (G)	12	66.8	43.0	88.0	15.3
KGY3	Y-ACC. HEAD OFFSIDE IMP. 3MS. (G)	12	59.0	28.0	86.0	17.7
	VG=60					
ROSYM	ACC.UPP.RIB IMPACT SID.MAX. (G)	4	202.0	170.0	243.0	30.3
ROSY3	ACC.UPP.RIB IMPACT SID.3MS. (G)	4	114.5	68.0	159.0	37.8
RUSYM	ACC.LOW.RIB IMP.SIDE MAX. (G)	5	315.0	214.0	465.0	97.8
RUSY3	ACC.LOW.RIB IMP.SIDE 3MS. (G)	5	143.4	84.0	192.0	43.0
ROGYM	ACC.UPP.RIB OFFSID.IMP.MAX. (G)	2	130.5	104.0	157.0	37.5
ROGY3	ACC.UPP.RIB OFFSID.IMP.3MS. (G)	2	104.0	91.0	117.0	18.4
RUGYM	ACC.LOW.RIB OFFSID.IMP.MAX. (G)	5	130.2	88.0	158.0	27.7
RUGY3	ACC.LOW.RIB OFFSID.IMP.3MS. (G)	5	105.8	73.0	131.0	22.6
T1Y3	Y-ACC.THOR.SPINE 1 3MS. (G)	5	92.8	79.0	108.0	11.7
T12Y3	Y-ACC.THOR.SPINE 12 3MS (G)	5	124.4	108.0	145.0	13.5
KSY3	Y-ACC. HEAD IMP SIDE 3MS (G)	5	74.6	61.0	93.0	12.5
KGY3	Y-ACC. HEAD OFFSIDE IMP. 3MS. (G)	5	50.0	30.0	80.0	20.5

Figure 4a. PMHS loadings according to impact speed, left side impact

Test No.	Impact Side	Speed (km/h)	Age (years)	Sex	Severity of Head Injury (AIS)	Body Height (cm)	HIC 1) Analogue Impact Side	Res. Head Acc., Impact Side (g)	Res. Clivus Acc. (g)	HIC 2) Analogue Clivus	Head Contact
8301	L	50	39	f	3	173	1042	78			3)
8304	L	50	28	m	1	172	1500	112			3)
8317	L	60	50	m	3	173	1115	121			
8319	L	60	40	m	3	175	788	104			
8434	R	45	59	m	6	172	347	65	65	307	
8501	R	45	26	f	2	169	715	120	-	-	
8504	R	45	41	m	3	170	900	95	63	201	
8517	L	60	27	m	2	175	1207	157	133	529	3)

1) Accelerometer at the temporal area
2) Accelerometer at the clivus
3) Head hits the door window sill

The following injuries occurred:

8301: Open fracture of zygomatic bone
8304: Superficial laceration on left upper eyelid
8317: Transverse fracture of clivus, extending into the left petrous bone
8319: Transverse fracture of clivus
8434: Transverse fractures of both occipital condyles, with extension of the fracture on the left to the occipital foramen, partial detachment of the brain stem from the pons (AIS 6).
8501: Transverse fracture of the right occipital condyle with extension to the right margin of the occipital foramen
8504: Transverse fracture of the clivus with extension to the right occipital condyle
8517: Contused-lacerated wound, 4 cm long, at the left eyebrow

Figure 5. Tests with head injuries. Load parameters, anthropometric data and severity of head injuries

skull base or deviations in a lack of uniform bone conditions. The injury mechanism mainly consists of a shearing load on the transitional zone between the cervical vertebral column and the head.

It can be assumed in general that the severity of injury is underestimated, because it is not possible with PMHS to detect smaller brain lesions such as concussions and contusions due to the absence of blood pressure and the impossibility to determine postmortem functional disturbances.

- *Thoracic injuries*

Rib fractures were the most frequent injuries observed independent of the impacted side. They depend mainly on PMHS age.

The direct force on the thorax by the intrusion of the vehicle's side panel is the decisive factor for the occurrence of rib fractures. The deformation of the thorax produces bending stress to the ribs. The rib fractures are mainly at the impacted side of the thorax.

	DIRECTION OF IMPACT						
	Left				Right		
km/h	40	45	50	60	40	45	50
AIS							
0	4	8	9	2	5	5	1
1							
2							
3		1	1				
4		1	2	1		3	2
5				2		2	9

Figure 6. Frequency of abdominal injuries in the individual AIS classes, according to direction of impact and impact speed

- *Abdominal injuries*

The direction of impact and the PMHS age is of decisive importance for the type and severity of abdominal injuries. This result is associated with the sustained test results performed with a flat inner door panel. It relates to the anatomy of the abdomen: The liver, which is particularly injury-prone due to its size, mass and stress sensitivity, is located on the right-hand side and only covered by the body wall. It is therefore most seriously exposed to impacts from the right. In the event of body deformation, it is subjected to pressure to which it cannot yield in the direction of the vertebral column.

In left-side impacts, the spleen is exposed in a similar manner as the liver on the right, but due to its smaller size and mass, it is less vulnerable. It also has the possibility of yielding to the load in the direction of the centre of the body without being compressed between the impacted body wall and the vertebral column.

The distribution of abdominal injuries, arranged according to direction of impact, collision speed and AIS class, is shown in Figure 6.

A qualification must be made with regard to the established postmortem abdominal injuries: moderately severe injuries which would normally be manifested in the form of bleeding in soft tissue and organs in the case of living occupants cannot be reliably determined because of the cessation of circulation. Thus, here again, an underestimation of the extent of injury under these impact conditions is possible.

- *Spinal injuries*

Spinal injuries were identified in 43 test subjects; in 15 cases no injuries were found, despite thorough autopsy (using sectioning techniques) of the isolated frozen vertebral column /7/.

The injuries were predominantly in the region of the cervical spine and the upper thoracic spine. The distribution of the severity of spinal injury according to AIS in the impact groups from the right and left can be derived from Figure 7.

Although in a majority of cases, no side specific injury mechanism can be deduced from the kinematic analysis of high-speed films and the known direction of loading. During the loading phase of the spinal column, there is a tensile strain on the side opposite the impact and a compression on the impacted side. In individual cases this observation was also deducible from the pattern of injury.

- *AIS of body regions*

Figures 8 and 9 list the injuries resulting from all the individual tests, arranged according to the degree of severity of injury to the body regions, the Figures include the maximum AIS (MAIS), the number of rib fractures, some anthropometric data and the test number.

Statistical Evaluation

The purpose of this evaluation was to find the relationship between mechanical and anthropometrical parameters on the one hand and injury severity on the other based upon test results. To determine these relationships the logistic regression analysis method was used. With numerous calculation runs, the best combination of variables was selected in order to predict the AIS probability for a special traumatic or injury index "Z". These prediction functions, derived for different body parts, will be described below:

Prediction of Thorax AIS Probability (TOAIS)

For the prediction of the probability of thorax AIS rankings, numerous calculation runs have shown the following combination of covariables:

- Age (years)
- Maximum acceleration at the 4th rib on the impact side (ROSYM) (g)
- "Damping parameter" on the impact side of the lower thorax half (T12RS2) (gt)

to constitute the best combination of predictors.

(T12RS2) is one of several "damping parameters" tested for the thorax, which is calculated according to the following formula:

$$T12RS2 = RUSYM \times RUSYT - T12RM \times T12RT$$

T12RS2 comprises the products of maximum accelerations and of the duration of the pulse (RUSYT and T12RT) between the onset of vehicle impact and attainment of maxima at measuring points RUSYM (8th rib on impact side) and T12RM (12th thoracic vertebra).

The difference between the values calculated as described above with the dimension of velocity at the 8th rib and the 12th thoracic vertebra corresponds to the loss of velocity as a result of the absorption of energy by the body region concerned, hence "absorption."

Using the selected combination of covariables, the logistic model has estimated the following parameters for the injury index Z:

$$Z_{TO} = 0.15\ AGE + 0.012\ ROSYM - 0.0004\ T12RS2$$

Figure 10 shows the probability curves for each individual thorax AIS (TOAIS) as a function of the injury index Z_{TO}.

The squares in the probability diagram represent the injuries observed. The Z value is calculated according to the above formula and constitutes a measure of the accident severity. The second value represents the injury observed.

Generally, it can be said that the quality of fit is better, the more tests are observed on or near the envelope.

Prediction of Abdomen AIS Probability (ABAIS)

For the prediction of ABAIS one important fact must be considered: For anatomical reasons (e.g. location of liver and spleen) the injury patterns caused by impact from right or left side are different. Therefore satisfactory prediction for ABAIS cannot be achieved without reference to the direction of impact.

The severity of injury of the left side impacted PMHS was also very difficult to predict because only a few cases of abdominal injury were observed in these tests.

AIS	DIRECTION OF IMPACT Left	Right
0	5	10
1	15	10
2	9	6
3	1	1
4	-	-
5	1	-
	31	27

Figure 7. Frequency of the severity of spinal injury, according to direction of impact

OBS	RUN	VEL	SEX	AGE	BMASS	BLEN	NRF	TOAIS	HEAAIS	ABAIS	EXAIS	SPAIS	MAIS
VG = 40													
1	8313	41	W	28	51	159	0	0	0	0	0	1	1
2	8315	41	W	38	60	164	4	2	0	0	0	0	2
3	8318	41	M	28	63	184	0	0	0	0	0	1	1
4	8405	42	M	60	64	170	16	4	0	0	0	1	4
5	8424	40	W	49	70	157	10	3	0	0	0	1	3
VG = 45													
6	8406	45	M	30	60	175	0	0	0	0	0	0	0
7	8410	44	M	53	74	172	17	4	0	5	0	0	5
8	8411	45	M	64	76	172	18	4	0	4	0	2	4
9	8414	45	M	36	67	177	14	4	0	4	2	0	4
10	8416	45	M	45	72	188	14	4	0	0	0	0	4
11	8432	45	M	60	82	174	18	4	0	4	0	1	4
12	8434	44	M	59	87	172	30	4	6	0	2	3	6
13	8501	44	W	26	56	169	10	3	2	5	2	1	5
14	8504	45	M	41	66	175	13	3	3	0	1	1	3
15	8505	44	W	25	65	167	0	0	0	0	0	1	1
VG = 50													
16	8123	50	M	22	71	178	0	0	0	0	0	2	2
17	8203	49	M	21	68	173	0	0	0	5	0	2	5
18	8206	51	M	23	63	175	14	3	0	4	1	1	4
19	8210	51	M	29	70	184	6	2	0	5	0	0	5
20	8213	51	M	47	90	170	27	4	0	5	0	2	5
21	8302	50	M	22	65	175	0	0	0	5	0	0	5
22	8303	51	M	52	68	172	22	4	0	5	0	1	5
23	8306	51	M	59	60	169	20	4	0	4	2	2	4
24	8326	50	M	32	82	171	5	2	0	5	0	0	5
25	8329	51	M	38	77	178	16	4	0	5	0	0	5
26	8401	50	M	36	91	176	6	2	0	5	0	2	5
27	8508	50	M	53	68	163	20	4	0	5	0	0	5

- Impact from right -

Lengend:

OBS:	Serial number	TOAIS:	Thorax AIS
RUN:	Test number	HEAAIS:	Head AIS
VEL:	Test velocity (km/h)	ABAIS:	Abdomen AIS
BMASS:	Body mass (kg)	EXAIS:	Extremity AIS
BLEN:	Body length (cm)	Spais:	Spinal AIS
NRF:	Number of rib fractures	MAIS:	Maximum AIS

Figure 8. Severity of injury in the individual tests

In contrast, it was possible to predict the probability of injury for the right side impacted PMHS examined to a satisfactory degree with the aid of the following covariables:

- Body mass (BMASS) (kg)
- Acceleration (3 ms value) at the 12th thoracic vertebra in the Y direction (T12Y3) (g)
- Acceleration (3 ms value) at the lower end of the sternum in the X direction (BUX3) (g).

For this covariable combination the logistic model has estimated the following parameters for Z:

Z_{AB} = 0.15 BMASS + 0.-65 BUX3 + 0.088 T12Y3

Figure 11 shows the probability curves for the three abdomen AIS grades 0, 4 and 5 over the Z value for impacts from the right. Up to a Z value of almost 20, the probability of an abdomen AIS = 0 is greater than 50%. At a Z value of 20.5 the predicted probability for an ABAIS of 0 and 5 is approximately equal (32%). At Z values above this limit, the probability of an abdomen AIS of 5 increases continuously to 100%. As expected, due to the broad overlapping of the observations as a function of Z it was not possible to predict abdomen AIS 4.

Prediction of Body AIS Probability (TAAIS)

The injury severity of the body is determined by the highest AIS value of the thorax or abdomen. This TAAIS will be discussed here in order to enable a comparison to be drawn with the TTI (Thoracic Trauma Index) /6/ section 5.3. Whereas the covariable combination for the TTI injury index is essentially based on impacts from the left, the logistic model for prediction of the TAAIS demonstrated the best predictive power for impacts from the right. For this reason, the probability curves for TAAIS rankings 0, 4 and 5 are shown in Figure 12 for impacts from the right. The three tests with TAAIS 2 and 3 in the test series were not considered.

The following covariable combination has been found to be the best injury predictor without using the impact velocity and with measurement variables which appear to be interpretable:

- Body mass (BMASS) (kg)
- Acceleration (3 ms value) in X direction at lower sternum (BUX3) (g)
- Acceleration (3 ms value) at the 12th thoracic vertebra in Y direction (T12Y3) (g)

With this covariable combination, the logistic model estimated the following parameters for the injury index Z:

$$Z_{TAR} = 0.15\ \text{BMASS} + 0.08\ \text{T12Y3} + 0.06\ \text{BUX3}$$

Below a Z value of 18.3, the envelope of the AIS probability curves indicates a high degree of probability to be uninjured (the highest degree is below Z = 18). Between Z = 18.3 and Z = 20, a TAAIS of 4 is largely to be expected, and above Z = 20 the probability for a TAAIS 5 of about 45% increases continuously to 100% (at Z = 25).

Prediction of Injuries From Values for Dummies

One of the research goals of this project was to investigate the extent to which dummies are suitable test devices for lateral impacts.

OBS	RUN	VEL	SEX	AGE	BMASS	BLEN	NRF	TOAIS	HEAAIS	ABAIS	EXAIS	SPAIS	MAIS
VG = 40													
1	8404	41	M	51	60	168	17	4	0	0	0	2	4
2	8426	40	M	24	83	176	5	2	0	0	0	0	3
3	8427	40	M	65	91	172	15	4	0	0	0	1	4
4	8431	40	M	26	78	190	1	1	0	0	0	1	1
VG = 45													
5	8327	47	M	53	62	171	4	2	0	0	0	2	2
6	8407	46	W	27	61	160	11	3	0	0	0	0	3
7	8409	44	M	25	95	188	7	3	0	0	0	0	3
8	8412	45	M	54	68	177	12	3	0	0	3	0	3
9	8413	45	W	41	54	169	14	4	0	0	0	1	4
10	8415	45	M	27	58	177	4	1	0	3	0	0	3
11	8433	45	M	47	60	173	11	3	0	0	0	1	3
12	8502	45	M	31	68	174	16	4	0	0	0	1	4
13	8503	46	W	47	61	158	29	4	0	4	2	3	4
14	8506	44	M	50	71	163	21	4	0	0	0	2	4
VG = 50													
15	8204	50	W	40	60	158	18	4	0	0	0	1	4
16	8205	50	M	43	77	178	23	4	0	0	0	1	4
17	8207	51	M	58	68	165	19	4	0	0	0	1	4
18	8211	51	M	29	59	177	14	3	0	0	0	2	3
19	8301	51	W	39	57	173	19	4	3	0	0	1	4
20	8304	50	M	28	61	172	18	4	1	3	0	1	4
21	8305	50	W	42	60	162	26	4	0	4	0	2	4
22	8328	50	M	44	88	186	19	4	0	0	0	1	4
23	8402	53	M	33	73	187	17	4	0	0	0	1	4
24	8430	51	M	63	73	172	20	4	0	0	3	2	4
25	8509	50	M	62	62	168	20	4	0	4	2	2	4
26	8513	50	M	23	62	173	2	1	0	0	0	1	1
VG = 60													
27	8314	61	M	23	80	192	16	4	0	5	1	2	5
28	8317	61	M	50	85	173	22	4	3	4	2	2	4
29	8319	61	M	40	82	175	20	4	3	5	0	5	5
30	8517	60	M	27	74	175	11	4	2	0	0	1	4
31	8518	60	W	19	57	169	0	0	0	0	0	1	1

- Impact from left - (For legend, see Figure 8)

Figure 9. Severity of injury in the individual tests

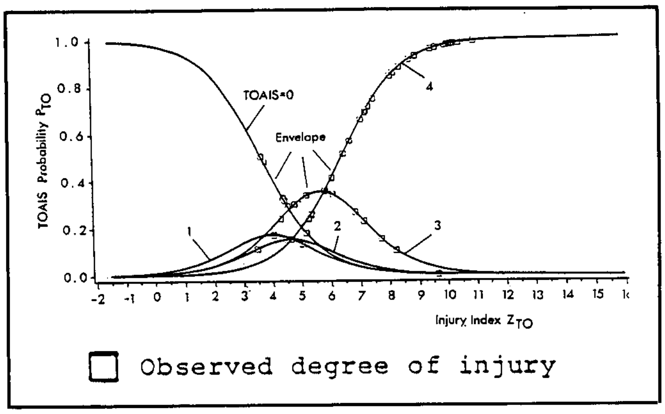


Figure 10. Predicted probabilities of thoracic injury

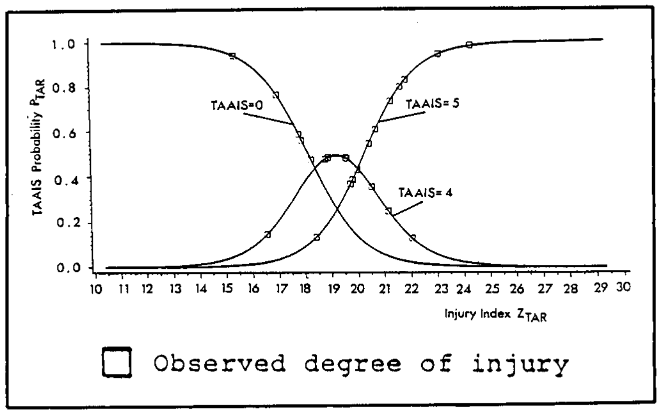


Figure 12. Predicted probability of torso injury (TAAIS) for impacts from the right

Again, the logistic regression method was used to define prediction functions on the basis of dummy loads under the same test conditions. It was based on the prediction functions established for PMHS.

Comparisons made of calculated Z values and predicted AIS probabilities for the three dummies investigated and PMHS follow.

Prediction of Abdomen Injury Probability Using Dummies

Because the abdomen injury index does not include the age as variable (section 4.2) it will be discussed first.

For the prediction for PMHS, the logistic model has estimated the following Z values:

$$Z_{AB} + 0.15\ BMASS + 0.065\ BUX3 + 0.088\ T12Y3$$

Figure 13 shows the Z values determined for PMHS and the specific dummy types investigated. Statistical characteristic data such as mean values, maxima and minima and standard deviations are specified.

In a comparison of the mean injury indices the HYBRID II dummy displays the least deviation from the PMHS and hence also from the predicted probability of ABAIS.

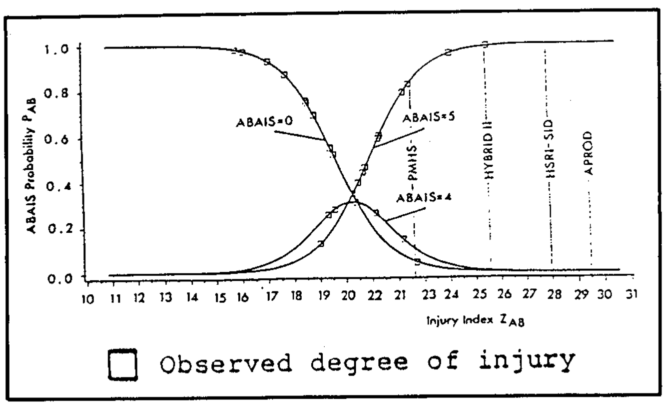


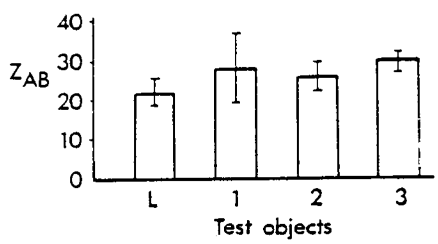
Figure 11. Predicted probability of abdominal injury (ABAIS) for right side PMHS impacts

The variance analysis performed indicates that according to the established prediction function and associated Z values the dummies investigated do not accurately simulate the behavior of the PMHS. If other variables and/or PMHS samples are selected other conclusions may be drawn.

Figure 14 and Figure 11 depict the distribution of AIS probability in relation to the calculated mean Z values.

At a Z value in excess of 21, the logistic model predicts an ABAIS of 5 with the highest probability. All of the dummies, and the PMHs, have a mean Z value above this level at a test velocity of 50 km/h.

		Z-VALUE				
		N	MEAN	MAX	MIN	STD
OBJECT	SPEED GROUP					
L	40	5	17.23	19.53	15.78	1.50
	45	10	20.08	22.68	18.48	1.29
	50	12	21.65	25.52	18.80	1.93
1	50	3	27.91	36.30	19.24	8.54
2	50	3	25.56	29.50	22.05	3.74
3	50	5	29.49	31.97	26.51	2.36



L: PMHS, 1: HSRI-SID,
2: HYBRID II, 3: APROD

Figure 13. Comparison of the injury indices of PMHS and dummies for the abdomen (right side impacts)

Distribution of Injury Severity, Right Side Impacts						
ABAIS	Observed N	Observed (%)	PMHS Calculated (%)	HSRI SID Calculated (%)	Hybrid II Calculated (%)	APROD Calculated (%)
0	1	8,3	11,2	0	0	0
1						
2						
3						
4	2	16,7	19,8	1	0	0
5	9	75,0	69,0	99	100	100
6						
			21,65	27,91	25,56	29,49
			Mean Z_{AB} Values for the Abdomen			

Figure 14. Predicted AIS probability of abdominal injury from tests with PMHS and dummies, mean Z $_{AB}$ related

The logistic model predicts similar AIS probability as observed for PMHS. However, it overestimates the injury severity for each of the dummies.

Changes of PMHS mean Z $_{AB}$ values influence the probability of injury severity much more sensitively than those of the dummies.

		Z-VALUE				
OBJECT	SPEED GROUP	N	MEAN	MAX	MIN	STD
L	40	6	7.05	9.42	3.83	2.44
	45	20	6.89	9.70	4.35	1.80
	50	21	6.69	10.25	3.40	2.23
	60	4	6.40	8.65	4.75	1.89
1	50	3	6.58	6.58	6.57	0.01
2	50	4	6.68	7.04	6.31	0.30
3	50	5	7.13	7.47	6.48	0.40

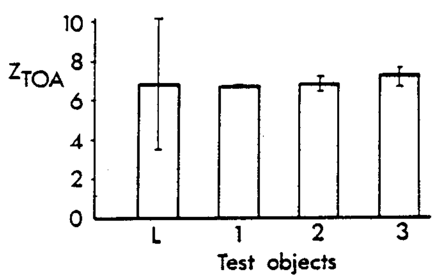


Figure 15. Comparison of thoracic Z $_{TOA}$ values, mean, maximum, minimum data and standard deviation L: PMHS, 1: HSRI-SID, 2: HYBRID II, 3: APROD Selected dummy age: 40 years

Distribution of Injury Severity, All Lateral Impacts						
TOAAIS	Observed N	Observed (%)	PMHS Calculated (%)	HSRI SID Calculated (%)	Hybrid II Calculated (%)	APROD Calculated (%)
0	3	12,5	2,1	2,2	2,1	1,8
1	1	4,2	2,2	2,3	2,2	1,9
2	3	12,5	3,2	3,4	3,2	2,7
3	2	8,3	15,7	18,5	15,7	15,4
4	15	62,5	76,8	73,6	76,8	78,2
5						
6						
			6,69	6,58	6,68	7,13
			Mean Z_{TOA} Values for the Thorax			

Figure 16. Predicted thoracical injury probability for mean Z $_{TOA}$ values

However, at the moment this applies only to the above-described tests at 50 km/h and with the selected vehicles with flat contour of the inner door panel.

Age-Independent Prediction of the Severity of Thoracic Injury From Measured Values for Dummies

To predict thoracic injury probability, age is an essential parameter. The injury probabilities of the body and abdomen are estimated to be age independent.

To compare calculated Z values and to predict the thoracic AIS probability, it is therefore necessary to select a certain age for the dummies or substitute the age by an algorithm where, e.g., the age distribution of the PMHS sample is used as a weighting factor.

Then two possibilities are described below:

a) Assignment of a dummy age:

The 50% male dummies investigated have a defined mass and body size. It therefore appears appropriate to assign a specific human age to the dummy. Analogously to mass and size, it is reasonable to use the mean value of the inner-quantile range of the age of the male adult population, or the mean value of the age of the car-driving male population. This method of age selection is described in /5/ for the estimation of thorax AIS, assuming a mean dummy age of 40 years. This age approximates almost precisely the mean age of the PMHS sample in this project.

A different dummy age can, of course, be substituted to calculate the injury index Z, depending on the target group.

Figure 15 compares the Z $_{TOA}$ values determined (TO*A*: age dependent) including other statistical characteristic data for the prediction function,

$$Z_{TOA} = 0.13 \text{ AGE} + 0.01 \text{ ROSYM}$$

and an age of 40 years.

The twin covariable combination was selected in this case. It can be interpreted more easily than e.g. a triple combination.

For test and development reasons it is questionable whether a maximum acceleration value is suited as predictor variable e.g. in regard to reproducibility and sensitivity to selected filter and measures.

The probability curves for TOAAIS rankings 0 to 4 as function of Z_{TOA} are shown in Figure 17.

The calculated mean Z values are so close together, that the predicted probability of injury severity is similar, for PMHS or the dummies investigated. Figure 16 depicts the observed and calculated AIS probability in connection with the mean Z values.

Figure 16 indicates that the logistic model overestimates the thoracic injury compared to the distribution observed. Independent of the different rib cage designs, each of the dummies investigated has nearly the same response at the test conditions selected for this project.

The variance analysis of the PMHS and dummy calculated Z values shows that the samples of the dummies selected reproduce those of the PMHS.

The curves of AIS probability in general are only valid in the range of observed injuries to estimate the probability to be injured in relation to the calculated Z values.

b) Age substitution:

Instead of assigning a specific age, it is possible to achieve independence from age in the prediction of injury with the aid of dummies by substituting the covariable age in the injury index in accordance with a specified age distribution. Expressed another way, the injuries to PMHS enter the regression to differing degrees, depending on the frequency within the age distribution.

This weighting factor may, for example, be based on the age distribution of motor-vehicle occupants

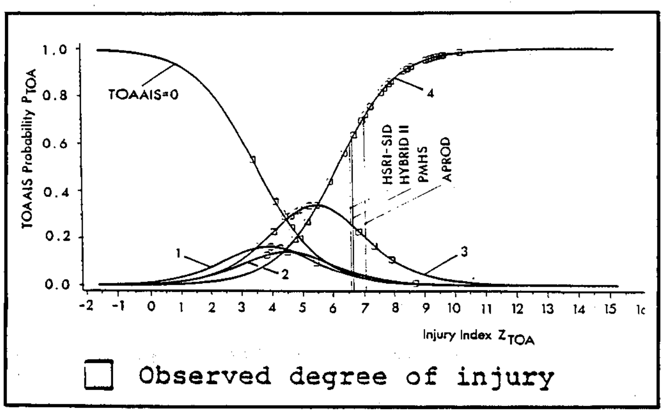


Figure 17. Predicted probabilities of thoracic injury

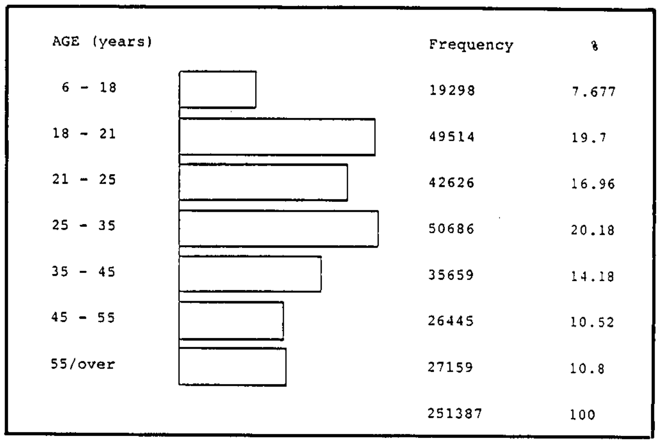


Figure 18. Age distribution of killed and injured motor-vehicle occupants in the Federal Republic of Germany in 1984

who were injured or killed in highway traffic in 1984 (Statistical Yearbook of the Federal Republic of Germany, 1985).

Figure 18 shows a histogram of the age distribution. The age group below 6 years was not taken into account. The mean age of this real accident group is approximately 33 years.

Prediction With the Thoracic Trauma Index (TTI)

On the basis of 49 side load tests Eppinger et al /6/ developed an injury index in 1984 which they describe as the best predictor for side loading. This thoracic trauma index (TTI) takes the following form:

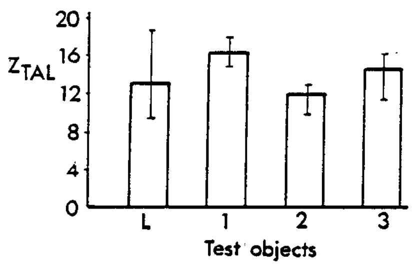
$$\text{TTI} = 1.4 \text{ AGE} + 0.5 \, (\text{T12YM} + \text{ROSYM}) \text{x m}/165$$

where: T12Y is the maximum Y acceleration at the 12th thoracic vertebra (g)
ROSYM is the maximum Y acceleration at the 4th rib on the left (= impact side) (g)
M: Body mass in pounds
1.4 and 0.5: empirically developed coefficients /6/

The TTI was developed empirically on the basis of loading data which originate in 3 cases from vehicle tests, in 4 cases from pendulum tests, and in the remaining tests the left side of the test object collided against an obstacle fitted with various padding materials.

The degrees of injury severity which are predicted with the TTI relate both to injuries to the chest cavity and to the abdominal cavity within the meaning of the AIS body region definition. The injuries thus correspond to the body AIS (TAAIS) in this study. The reasoning behind combining the thorax and abdomen in this way, at variance with the AIS classification, is that the upper abdominal organs—particularly the

		Z-VALUE				
		N	MEAN	MAX	MIN	STD
OBJECT	SPEED GROUP					
L	40	3	10.69	14.13	8.61	3.00
	45	10	10.52	12.00	8.51	1.21
	50	12	12.82	18.36	9.07	2.75
	60	5	15.83	20.13	9.44	4.10
1	50	2	16.10	17.75	14.46	2.33
2	50	4	11.45	12.68	9.76	1.27
3	50	5	14.26	15.97	11.13	1.86



L: PMHS, 1: HSRI-SID, 2: HYBRID II, 3: APROD (the dummies were assigned an age of 40 years)

Figure 19. Comparison of the mean body Z_{TAL} values for prediction of the TALAIS for impacts from the left with PMHS and dummies

liver, spleen and pancreas—are protected in the event of lateral loading by the thoracic cage. It therefore appeared appropriate to unite all organs beneath the thoracic skeleton as belonging to one body region.

The TTI was determined on the basis of left-side impacts. Therefore this group should also be considered from amongst the FAT data to derive the prediction function for the body with logistic regression analysis.

For the FAT impact group investigated, the logistic model provides the best prediction of the body injury with the covariables age, the 3 ms acceleration value at the 8th rib (RUSY3) and the acceleration maximum at the 12th thoracic vertebra. The logistic model estimates the injury index with the modified TTI function by:

$$Z_{TAL} = 0.074\ TTI^*$$

$$\text{with } TTI^* = 1.4\ AGE + 0.5\ (T12YM + RUSY3)\ \frac{BMASS}{75}$$

where AGE (years)

T12YM, RUSY3 (G)
BMASS (kg)
TAL: left side impacted

The TTI* differs from the TTI due to the 3 ms value at the 8th rib. Without the empirical coefficients /6/ the quality of prediction is nearly the same.

Distribution of Injury Severity, All Dummy and Left Side PMHS Impacts						
			PMHS	HSRI SID	Hybrid II	APROD
TALAIS	Observed	(%)	Calculated (%)			
0	1	3,2	0	0	1,1	0
1	2	6,4	0,8	0	2,7	0
2	2	6,4	1,3	0	3,8	0
3	6	19,4	8,4	0	22,7	2,2
4	18	58,2	89,5	91,4	69,7	96,5
5	2	6,4		8,6		1,3
6						
			12,82	16,10	11,45	14,26
			Mean Z_{TAL} Values for the Body			

Figure 20. Predicted probability of body injury (TALAIS) related to mean Z_{TAL} values

Figure 19 shows the mean Z values and other statistical data for PMHS and the dummies investigated (age: 40 years).

A variance analysis indicates that each dummy Z-sample reproduces the Z-sample of the PMHS in the body region for the selected test conditions and an assigned dummy age of 40 years.

Figure 21 depicts the body injury probability curves (TALAIS) for all performed left side PMHS tests of this project. Comparison of the predicted injury probability TALAIS in relation to the mean Z values of the test objects is made in Figure 20.

Analysis of the injury distribution of PMHS observed demonstrates that the logistic model overestimates the injury severity for both PMHS and dummies.

For right-side impacts, in section 4.3 the prediction function

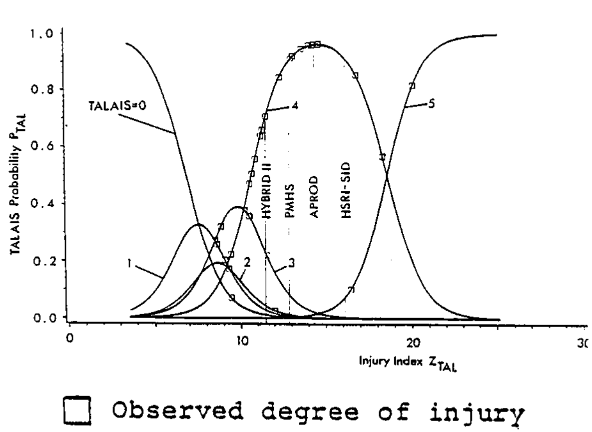


Figure 21. Predicted probability of body injury (TALAIS) for left side impacts

Z_{TAR} = 0.15 BMASS + 0.08 T12Y3 + 0.06 BUX3
TAR: right side impacted

was found to estimate the injury probability TAAIS for the PMHS body. The injury index Z is in this case age independent.

Because the variance analysis of the Z_{TAR}-samples derived from PMHS and dummies shows that these samples are significantly different, this prediction function is not suited to predict body injuries with the dummies investigated.

Summary and Conclusions

An attempt has been made in this investigation to establish a relationship between mechanical inputs measurable on the human being and the injuries occurring in the event of a side impact. The relationships were investigated for the thorax, the abdomen and the body, because these areas where most frequently represented in injuries occurring in the simulation of a serious accident selected.

An appropriate means of achieving this has proved to be logistic regression. This procedure takes the fact into account that there can be no strictly deterministic relationship between mechanical loading on the one hand and human injury on the other. This relationship is expressed by the fact that there is no set AIS value (e.g. AIS = 3.7) corresponding to a particular loading, but rather a distribution of probability for the various discrete AIS values. This then takes the generally experienced observation into account that different people, subjected to approximately comparable mechanical load, may suffer injury of differing severity.

In summary the formulae for injury indices Z, which have to be applied as input variables into the AIS distribution model, are as follows:

Thorax
(for impacts from the left and the right)

Z_{TO} = 0.15 AGE + 0.012 ROSYM − 0.0004 T12RS2
with T12RS2 = RUSYM × RUSYT − T12RM × T12RT

Abdomen:
(for right side impacts only)

Z_{AB} = 0.15 BMASS + 0.065 BUX3 + 0.088 T12Y3

Body
(Thorax and Abdomen)

- *Heidelberg* (for right side impacts only)

  Z_{TAR} = 0.15 BMASS + 0.08 T12Y3 + 0.06 BUX3

- *Eppinger* (for left side impacts only)

  Z_{TAL} = 0.074 TTI*
  with TTI* = 1.4 AGE + 0.5 (T12YM + RUSY3) • $\frac{\text{BMASS}}{75}$

Due to the lack of blood circulation, only three cases with AIS less than 4 could be determined in the sample of abdominal injuries.

It had to be assumed, therefore, that in regression analysis of the abdomen AIS it was only possible to make statements regarding AIS classes 0, 4 and 5.

It was considered to be inappropriate to incorporate highly correlated influencing variables, such as impact velocity, if they obviously only had an indirect influence on the accident mechanism.

Within the scope of this research project it was not possible to include all conceivable influencing variables in the regression analysis. Furthermore, it appeared reasonable to include no more than 3 influencing variables in the analysis at any one time.

Since it can be assumed that many more useful indicators of injury can be extracted from the test data than has so far been possible, an additional evaluation is being carried out by Daimler Benz.

A fundamental comment must be made at this point:

The relationships established during the course of this project are based on a multiplicity of examined, and to a large extent rejected, combinations of influencing variables. It therefore cannot be excluded that the relationships found are fortuitous results of the above test series. For this reason, they cannot be regarded as final results, but rather as well-founded hypotheses. A control investigation is therefore being planned in order to substantiate the results, with which it is intended to examine whether the hypotheses have any permanence—and hence can be considered a genuine result—or whether it will be necessary to formulate new hypotheses.

This reservation also applies to the investigation of the interrelationship of dummy loadings and the probability of injuries to PMHS predicted on the basis of these loadings. It should be examined whether the agreement found with the PMHS does not merely happen to suit the vehicle and test configuration selected here but also maintains its validity for other vehicles and test parameters. It is possible that the dummies show other differences in the test groups at other velocities, shape and stiffness of impacted zones. In the 50 km/h test group, all dummies, irrespective of differing thoracic structure, would predict the same injuries in the thoracic regions with the aid of the Z function. In this case it should be determined to what extent, for example, a thorax must be reproduced with an impactor in order to be able to make similar statements. In respect to the PMHS thoracic and body injury distribution observed in each case the logistic model overestimates the injury severity for both PMHS and dummies.

There is room for doubt in the case of the relationship found between mechanical loading and actual or predicted injury in the abdominal and thoracic regions, since in the additive connection of body mass and acceleration for the abdomen, as well as age, mass and accelerations for the thorax with extreme padding, an injury index Z is established which is dependent virtually solely on body mass or body mass and age and hence results in absurd values.

It has to be kept in mind that the AIS classification has some weak points when PMHS are the basis of analysis.

First, nerve lesions cannot be diagnosed in the case of head and spine AIS classification and second, as there is no blood pressure, it is not possible to stipulate AIS classes 1, 2 and 3 for the abdominal region.

Contributors

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On the Relationship Between Kinematic Variables and Structural Failure in a Viscoelastic Medium Under Impact

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Abstract

Current procedures employed to develop injury indices for use in automotive safety research are limited, by necessity, to obtaining a series of observations on experimental subjects of resulting injuries which are a result of forces or velocities imposed on body segments. What is sometimes desired in this process is knowledge of the relevant stresses and strains within the body segment. Such data would assist in a better understanding of current injury criteria and the mechanisms that lead to injuries. This technical note reports, in a qualitative fashion, on a preliminary attempt to utilize DYNA3D [1], a finite element computer code, to compute the stress and strain fields of a thorax-like structure subject to impact.

Introduction

With the increased availability of finite element codes for the analysis of the dynamic response of three-dimensional solids, an opportunity is available to apply these techniques to the analysis of biomechanical response to impact. The finite element code and supporting programs used in the current study were developed at the Lawrence Livermore Laboratory in Livermore, California. The software consists of three programs: a preprocessor (INGRID), the main program (DYNA3D), and a postprocessor (TAURUS). The preprocessor is used to create the finite element mesh, define material properties, and specify initial and boundary conditions. The postprocessor provides graphic output and time history plots of selected parameters. DYNA3D is an explicit three-dimensional finite element code for analyzing the large deformation dynamic response of inelastic solids and structures. The options utilized to derive the results below included 8-node solid elements and an isotropic viscoelastic material.

The objective of the current research is to model the thorax and compute the stress, strain and displacement fields as a function of several impact conditions and correlate these with laboratory findings from human surrogates and injury criteria. Since the model and results are tentative, this note reports the qualitative results of the initial simulations. Correlation of model results and laboratory findings has not yet been initiated.

Model Description

The thorax-like model used in the initial computer simulations consisted of two parts, a viscoelastic interior with two voids and a ribbed structure surrounding it. The rib-like structure with viscoelastic contents was impacted by rigid cylindrical impactors 4" in diameter. In one case, the impactor was padded and in the other case, it was unpadded. The thoracic structure and the impactors are shown in Figure 1.

The viscoelastic material is characterized in DYNA3D by the bulk modulus K, density r, decay constant b, and the deviatoric shear relaxation modulus [2]

$$G(t) = G_1 + (G_s - G_1)e^{-bt}$$

Where:
G_s = short term shear modulus
G_1 = long term shear modulus

For the simulations below, the rib material was assumed to be elastic (but an inelastic option is available). For reasons of economy, in order to constrain the number of elements to a reasonable value, the number of ribs used was six. The overall dimensions of the structure were 10″ in diameter, and 12″ from top to bottom. The initial conditions were:

*Initial velocity of thorax-like structure = 20 mph

*Velocity of contacting surface = 0 mph

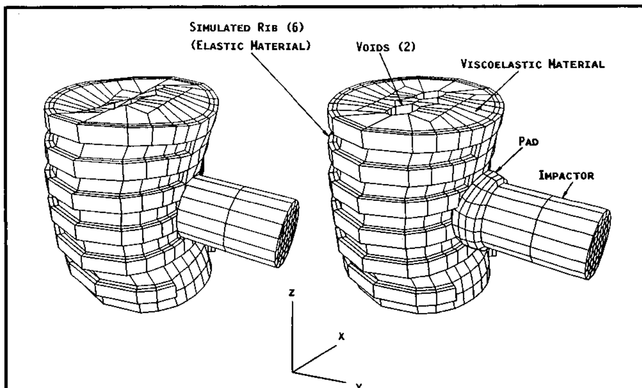


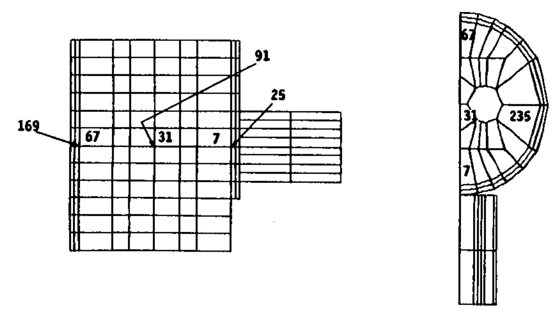
Figure 1. Finite element model of thorax-like system impacting a cylinder (time = 5 msec.)

*Centerline of impactor normal and intersecting centerline of structure

Other data is given in the appendix.

During impact the ribs contacted the fixed cylinder along a center line through the mid sagittal plane and halfway up the vertical dimension. The ribbed structure was free to move in all six degrees of freedom.

A review of the initial data (Case 1) showed that the model required improvements. The stiffness of the simulated rib cage was reduced, the decay constant of the linear viscoelastic interior increased, and a second case was run with an unpadded cylinder.



(a) Mid-Sagittal Plane (b) Mid-Horizontal Plane

Figure 2. Cross sectional illustrations

Results

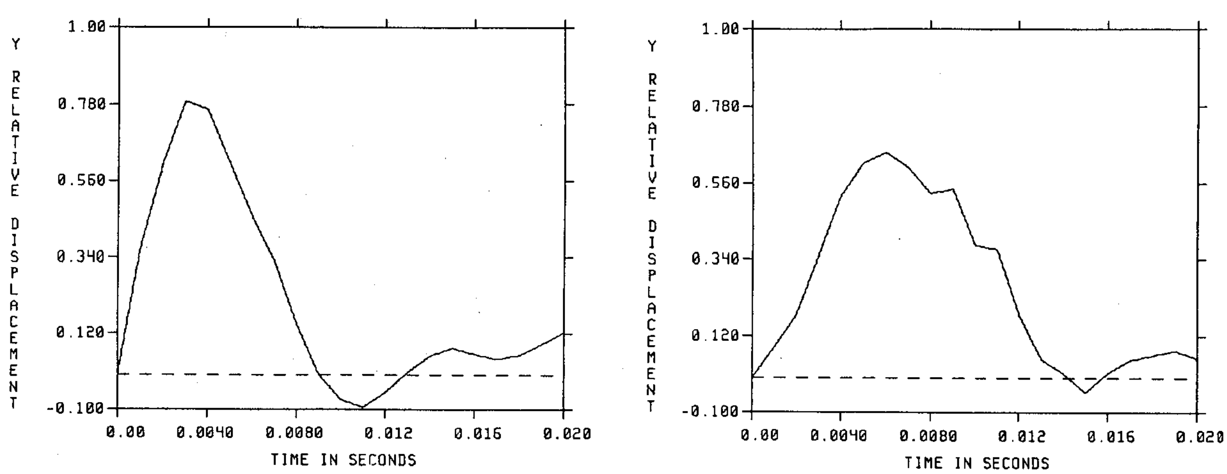
Case 1:

The first simulations performed were for the first 20 milliseconds after initial contact. It is to be noted that on the VAX 11/780, with the parameter values used in Case 1, a single run required 7 hours of central processor time with the unpadded cylinder and 8 1/2 hours of central processor time with the padded cylinder. The simulation yielded stresses, strains, strain rates, displacements, velocities and accelerations at selected points. Figure 2 illustrates mid-sagittal and mid-horizontal cross sections of the model with several elements and nodes, the responses of which were plotted. Figure 3 shows the relative displacement response between the sternum and the spine for both the unpadded and padded examples, and Figure 4 shows the corresponding relative velocity response between these points. Figure 5 shows accelerations of three nodes located in the mid-sagittal plane. Figure 6 is an example of x-y shear strain response for three elements in the horizontal mid-plane (See Figure 2(b)). These were the anterior, center and lateral elements in the horizontal mid-plane. Table 1 gives the six peak strains for these same three elements in the viscoelastic interior.

The results for both unpadded and padded impact are given. As mentioned earlier, these strains are of interest in a qualitative and relative way.

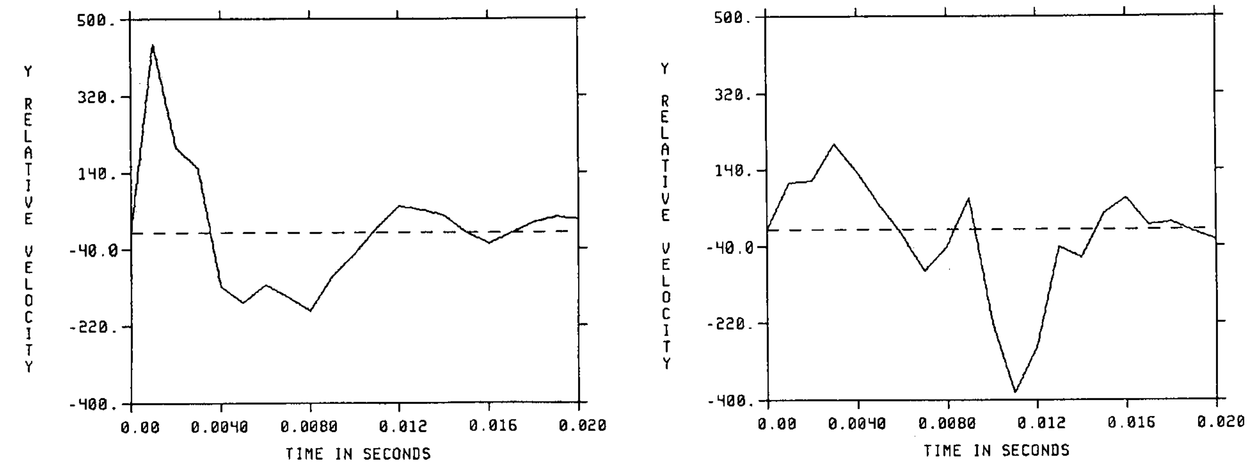
It can be seen in Figures 3 and 4 that the padding reduces the peak magnitudes of displacement and velocity as well as delaying the occurrence of these peaks as expected. For acceleration however, it can be seen in Figure 5 that the acceleration peaks were delayed but their magnitudes were not reduced in the padded case. No explanation for this occurrence is being given at this time.

A comparison of the values of the strains given in Table 1 shows that (1) the normal strains in the padded case are somewhat lower than the normal strains in the unpadded case, (2) the ϵ_{YZ} and ϵ_{ZX} shear strains are higher in the padded case, and (3) the ϵ_{XY} strains are higher in the unpadded case. Also, the shear strains in the center and lateral elements are



(a) Unpadded Cylinder (b) Padded Cylinder

Figure 3. Relative displacement of two points located at the sternum and spine (case 1)



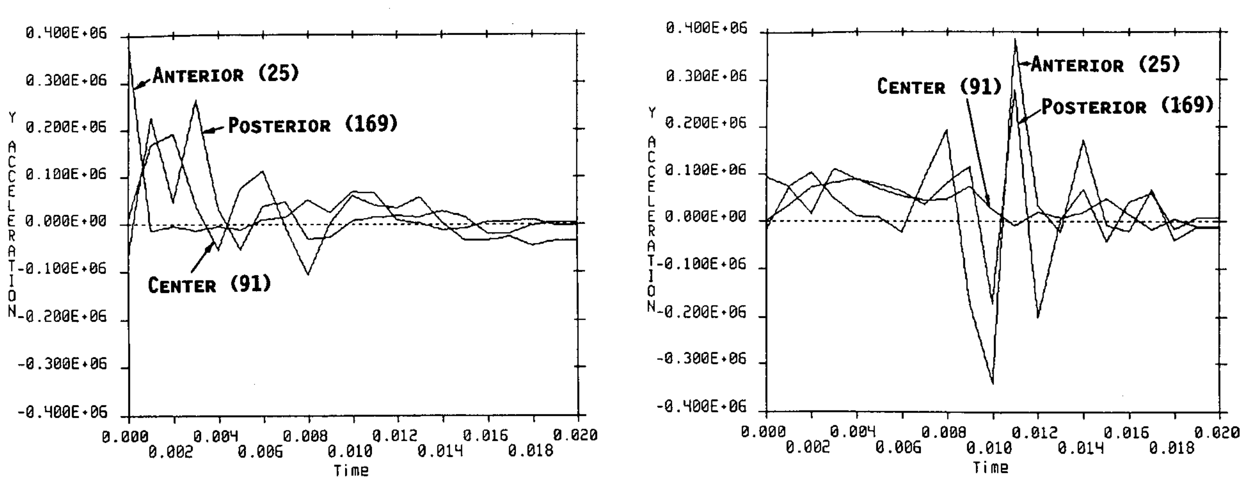
(a) Unpadded Cylinder (b) Padded Cylinder

Figure 4. Relative velocity of two points located at the sternum and spine (case 1)

of the same order and larger than the shear strain of the anterior element. The reason for (2), above, might be due to an observed rotation of the thorax-like model in the padded case during impact which was not observed in the unpadded case. The observation that the shear strains in the lateral location are of the same order as in the central location may possibly be due to (a) the effect of the void in the interior, or (b) shear waves as indicated by Langdon[3].

Case 2:

The model used in the initial runs was judged to be too "stiff". Parameter changes were made to make the model more compliant and the response for the unpadded example only was computed. While the initial runs in Case 1 showed a maximum deflection of 0.8 inches, with an almost complete rebound in 20 milliseconds, the run in Case 2 showed a maximum deflection of almost 4 1/2 inches with little rebound. Figure 7 shows the relative displacement of two points located on the sternum and spine. Figure 8 shows the relative velocities for these two points and Figure 9 shows acceleration at three points in the mid-sagittal plane. An illustration of the model 24 milliseconds after initial contact is shown in Figure 10.



(a) Unpadded Cylinder (b) Padded Cylinder

Figure 5. Y-acceleration for three nodes in the mid-sagittal plane (case 1)

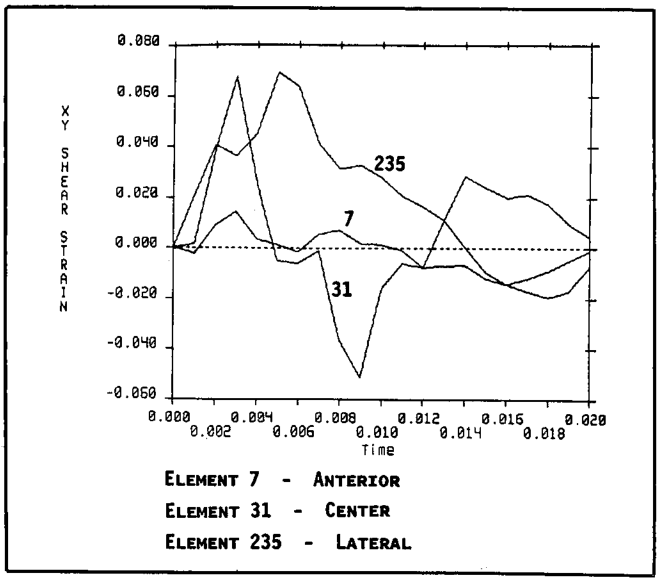


Figure 6. XY shear strain in horizontal mid-plane for elements for 7, 31 and 235 (case 1)

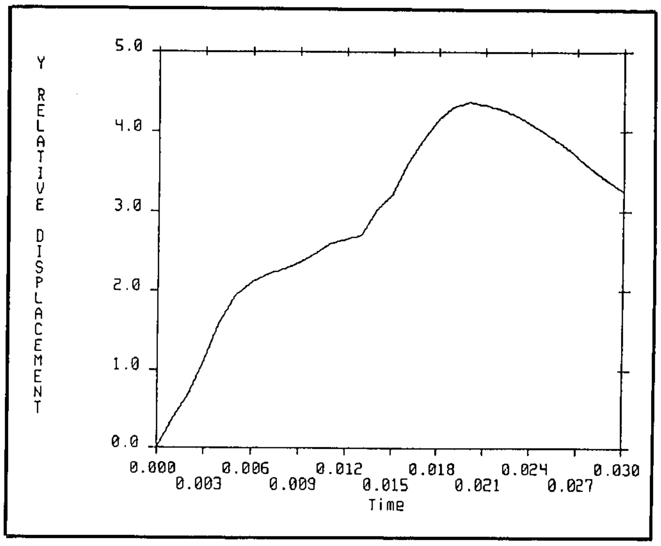


Figure 7. Relative displacement of two points located at the sternum and spine (case 2)

Conclusions

The work described above is still in the preliminary stages. The use of a computer code as described above offers considerable promise. Difficulties encountered to date include lack of availability of reliable material characteristics required for the input and the considerable computer time required for executing a case. As the model is still being refined, it is too early to draw any conclusions from the data. Appropriate material properties will probably lie between the values used in the two cases presented here. Future plans include

Table 1. Comparison of peak strains at selected locations in the viscoelastic interior (case 1).

	Anterior (Unpadded)	Anterior (Padded)	Center (Unpadded)	Center (Padded)	Lateral (Unpadded)	Lateral (Padded)
ε_X	.093	.056	.167	.090	.048	.032
ε_Y	-.120	-.118	-.167	-.130	-.063	-.005
ε_Z	.057	.050	.092	.080	.061	.083
ε_{XY}	-.015	.013	.067	.031	.069	.051
ε_{YZ}	-.011	-.086	-.025	-.096	-.026	-.087
ε_{ZX}	-.007	-.009	.016	.033	-.021	-.030

UNPADDED / PADDED

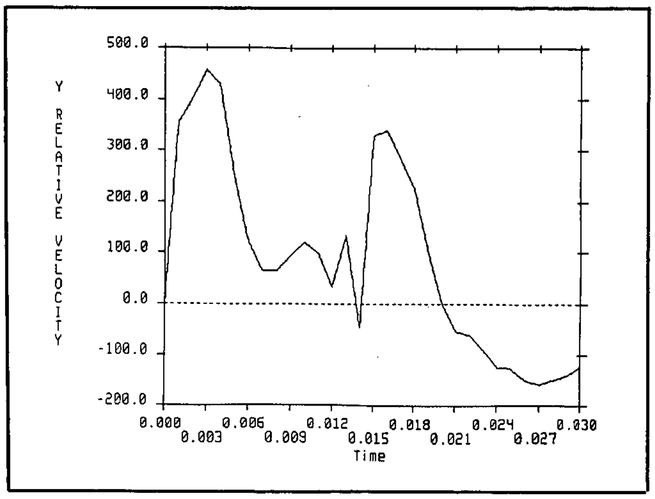


Figure 8. Relative velocity of two points located at the sternum and spine (case 2)

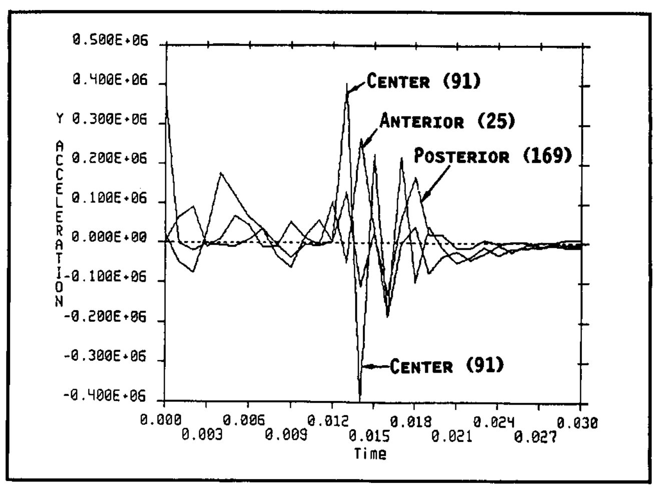


Figure 9. Acceleration of three nodes in the mid-sagittal plane (case 2)

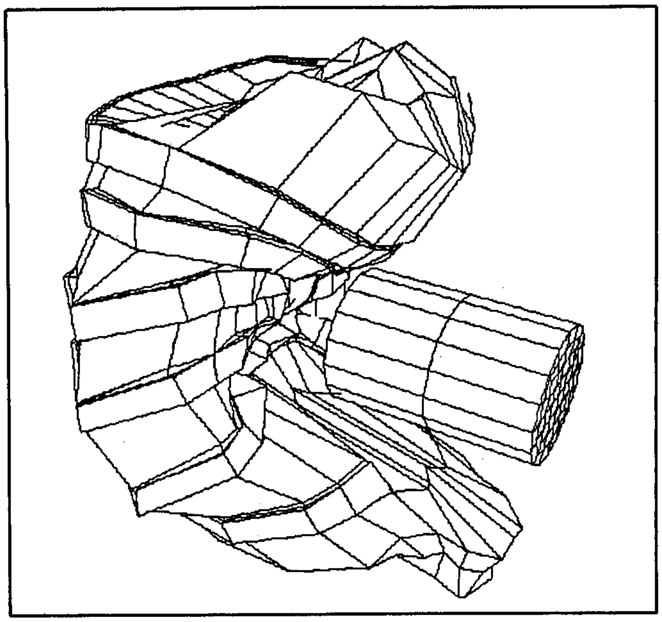


Figure 10. State of model (case 2) 24 milliseconds after initial contact

further adjustments of the geometry of the model, the use of a finer mesh, adjustment of the material properties and the execution of sensitivity studies.

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Appendix

Input Data

Units in all cases are in pounds, inches and seconds.

Case 1		Case 2	
Viscoelastic Interior	Bone	Viscoelastic Interior	Bone
K = 725 [ref #4]	E = $2.6x10^6$ [ref #6]	K = 725	E = $2.6x10^6$
r = $1.0x10^{-4}$ [water]	r = $1.73x10^{-4}$ [ref #7]	r = $1.0x10^{-4}$	r = $1.73x10^{-4}$
b = .4	Poisson's ratio = 0.3 [ref #4]	b = 625	Poisson's ratio = 0.3
G_s = 50 [ref #5]		G_s = 5	
G_l = 10 [ref #5]		G_l = 1	
Rib Thickness = 1/2"		Rib Thickness = 1/4"	
Rib Width = 1.0"		Rib Width = 1.0"	

Contribution and Evaluation of Criteria Proposed for Thorax-Abdomen Protection in Lateral Impact

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Abstract

A new study of the APR data for lateral impact tests using human subjects was undertaken to determine the quality of the possible correlations between the TTI and thorax-abdomen injuries.

The TTI (Thoracic Trauma Index) was proposed by the NHTSA(1) in 1984; it takes into account the age and weight of the subjects tested as well as the maximum of transverse accelerations measured at the 4th and 8th ribs and the acceleration of the dorsal column, to determine injury probability. The TTI values were recalculated using the new formula described by the NHTSA at the last STAPP Conference(2). The conclusion reached was that thorax-abdomen protection cannot be based on this criterion; it does not enable to predict the level of injury resulting from a side impact, nor to correctly classify the results of tests carried out under different conditions.

Accidentological data coming from the LPB survey show that, in many cases severe abdominal injuries were observed without the victim sustaining thoracic injury; these were either abdominal injuries in isolation or abdominal injuries associated with pelvic fractures. It is difficult to see how in such cases a scale of injury severity based solely on thoracic acceleration could account for the severity of abdominal injuries. These arguments also go against the utilization of a criterion such as the TTI and also clearly show the need for separate criteria for abdominal protection and thoracic protection.

Introduction

"An injury criterion can be defined as a biomechanical index of exposure severity which, by its magnitude, indicates the potential for impact induced injury. It is a physical parameter which correlates well with a scale of injury severity of the body region under consideration. This physical parameter can be a function of several variables(3)".

As such, a valid criterion always implies a specific underlying biomechanical response and mechanism of injury".(4)

The search for this type of criterion must lead to an improvement in the understanding of injury mechanisms and serve as a basis for evaluation of the effectiveness of modifications made to vehicles for safety purposes. This paper first sets out the data available from the APR's accidentological survey, focusing in particular on injuries to the thorax and/or the abdomen in side impact. We show in particular that in certain cases severe abdominal injuries are observed when the victim sustained no thoracic injury, or thoracic injuries of very "moderate" severity. This suggests that an injury criterion based solely on thoracic acceleration measurements cannot account for the occurrence and severity of these abdominal injuries, and that it is also necessary to have separate criteria for abdominal protection and thoracic protection.

As far as the thorax is concerned, transverse acceleration alone is not enough to account for the deformation of the impacted wall and the deformation of the thorax. This is a fundamental criticism of any concept based on the measurement of thoracic acceleration in side impact; transverse thoracic acceleration can, at most, give an indication of violence, but it can never be an acceptable indicator of thoracic injury. This analysis shows that the best criterion for thoracic protection is based on deflection measurements.

Thorax-Abdomen Injuries Observed in Real-World Collisions

The sample studied comes from the multidisciplinary survey carried out by the APR Physiology and Biomechanics Laboratory.

Side impact accounts for 16% of total body accidents but its level of severity is so high that it is responsible for 21% of severe injuries and 29% of overall fatalities(5). We are here most concerned with car-to-car collisions involving nearside occupants exposed to intrusion, that is to say, who directly sustained penetration of the wall at pelvis level; this corresponds to the most severe conditions. It must be noted that, in this configuration, the risk of severe or fatal injury reaches 50% above 30 km/h of $\triangle V$ of the struck vehicle. This level of risk is only reached at 40 to 45 km/h of $\triangle V$ for the other occupants, which justifies the priority given to the protection of nearside occupants directly sustaining penetration of the wall. The sample studied corresponds to occupants involved in the following conditions:

- trajectories at 2, 3, 4, 8, 9 and 10 o'clock for occupants involved aboard struck cars,
- unejected occupants of adult size.

It comprises 134 involved, including 20 fatalities (12 unautopsied).

The analysis of the most severe (AIS $\geq$ 3) injuries sustained shows a predominance of injuries to the abdomen (34%) and to the thorax (32%) over the pelvis (16%) and the head (13%).

Table 1 gives a break-down of the nearside injuries observed according to the three most frequently injured body regions in cases where severe abdominal injuries (AIS $\geq$ 3) were present. It concerns 23 cases, on which 16 with at least a liver or spleen or kidney injury (AIS $\geq$ 4) and 7 cases of other abdominal injuries with an AIS $\geq$ 3 value, 5 cases of severe abdominal injuries in isolation, unassociated with injuries to the thorax or pelvis are noted. 8 other cases of severe abdominal injuries were observed for occupants sustaining injuries of "moderate" severity to the rib cage, that is to say, with a number of fractured ribs less than or equal to 5 without flail chest. There are thus 13 cases on 23 corresponding to severe abdominal injuries (AIS $\geq$ 3) associated with either the absence of thoracic injury or with "moderately" severe thoracic cage injuries.

The simultaneous presence of thoracic injury and abdominal injury without pelvic fracture is rare: there are only 4 cases of associated injuries. On the other hand, the association with injuries to the thorax, the abdomen and the pelvis is noted in 40% of cases (9/23).

These data show that an injury criterion based solely on accelerometric measurements *at thorax level* allows us to account for the occurrence of only a minority of severe abdominal injuries (all the more so when the abdominal injuries are isolated).

One may wonder what the cause of these injuries is, especially in the absence of thoracic injury or pelvic injury.

A more in-depth analysis of these 23 cases shows that, on 20 cases where an armrest is present, we can assume a probable role of the penetration of the armrest on the occurrence of abdominal injuries in 17 cases (and in particular in the 5 cases where the abdominal injury was unassociated).

We must however exercise caution with regard to this point, since it is difficult to know precisely what happened in a real-world accident. However, it is justified that an expert can propose his interpretation of a probable or of the most probable hypothesis. We have tried to understand the mechanism responsible for the occurrence of these injuries. Figure 1 shows the wall velocity at pelvis level as a function of the crush at pelvis level for the overall sample (122 cases – the 12 unautopsied subjects were omitted). The cases of AIS $\geq$ 3 abdominal injuries are indicated by the triangles. We have singled out a sub-sample characterized by an impact severity which is equivalent in terms of wall velocity and crush at pelvis level. The population studied corresponds to the points

Table 1. Severe abdominal injuries, with or without thoracic and pelvic injuries

ABDOMINAL INJURIES		NUMBER OF FRACTURED RIBS (IMPACTED SIDE)	CLAVICLE FRACTURE	ABDOMINAL INJURIES		PELVIS FRACTURE (IMPACTED SIDE)	FLAIL CHEST	POSSIBLE ROLE OF ARM REST
				spleen or liver	other AIS $\geqslant$ 3			
spleen or liver injury N = 14 (AIS $\geqslant$ 4)	1	10	●	●	●	● 1 opposite side	●	●
	2	10	●	● □	●			●
	3	10		● □			●	
	4	9		● □	●			●
	5	7		●	●	●	●	●
	6	6		●	●	●		
	7	6		●		●	●	
	8	6		●		●		
	9	4		● □			●	●
	10	3		●	●	●		●
	11	0		● ✱	●			●
	12	0		● ✱	●			●
	13	0		●	●	opposite side ●		●
	14	0		● ✱				●
other severe abdominal injuries (AIS $\geqslant$ 3)	1	9			●	●	●	●
	2	5			●	●		●
	3	4	●		●	●		●
	4	2			●	●		
	5	1			●	side unknown		●
	6	0			●	side unknown		
	7	0		✱	●			●
	8	0			●	●		●
	9	0		✱	●			●

✱: isolated abdominal injuries
□: abdominal injuries associated with thoracic injuries only

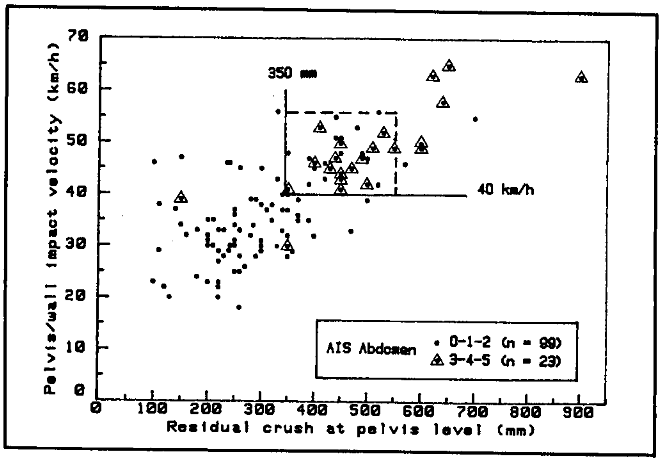

Figure 1. AIS Abdomen according to the pelvis/wall impact velocity and the residual crush at pelvis level

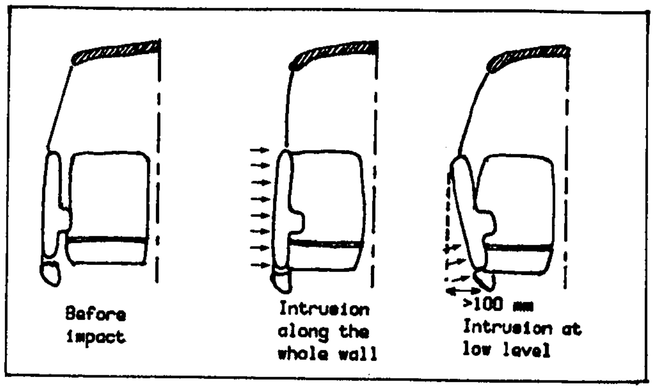

Figure 2. Different kinds of possible deformations of the lateral wall, according to the level of maximum intrusion

located inside the rectangle outlined by dots on figure 1, that is:

- a wall velocity of 40 to 55 km/h,
- a wall crush distance of 350 to 550 mm.

In the area outlined in this way, we notice the cases with or without abdominal injury spread out in the same way, so that no bias intrudes on the comparison.

On the 32 occupants represented in this rectangle, 17 display an AIS abdomen <3 and 15 an AIS abdomen ≥3. The possible influence of an armrest on the occurrence of these injuries cannot be determined. On the 17 cases of AIS < 3, there was a probable influence of the armrest in 11 cases, whilst it was probable in 9 cases on 15 of the severe abdominal injuries (AIS ≥ 3); the difference between these two sub-samples is not significant enough to allow us to draw any conclusions as to the possible role of the armrest in the occurrence of injuries. But one must immediately implement that an other parameter might be sufficient to explain however that the role of the armrest may be different in the two sub-samples: this is the wall deformation described in terms of the deformation gradient at pelvis level on the one hand and at thorax level on the other hand (Figure 2).

The "type" of deformation on the door panel may be different according to whether the whole of the wall along all its height is involved, or whether the penetration occurs solely in the lower part, resulting at the end of impact in a wall with a sloping section (minimum gradient of deformation or intrusion of 100 mm between the top and the bottom of the wall). In the sub-sample of AIS < 3 abdominal injuries, in 12 on 17 cases, the wall was sloping in its final state, whilst this was not the case in 6 on 15 cases for the severe abdominal injuries. We can suppose that this "form" of wall deformation is more favourable in the sense that the armrest is no longer at abdomen level, that it is tipped up, and therefore that, on contact with the armrest the trajectory of the abdomen is deflected against the wall and no longer sustains direct penetration in the direction of the impact.

It should be noted that severe abdominal injuries are associated with pelvic injuries in 9 on 15 cases (3 on 17 cases only for AIS < 3 abdominal injuries) and with very moderate thoracic injuries in 8 on 15 cases (3 on 17 cases only for AIS < 3 injuries). To conclude, the mechanism governing the appearance of severe abdominal injuries is not easy to define on the basis of accidentological data.

This last point suggests that these injuries are associated, in most cases, with injuries to either the pelvis or the thorax, or to both the pelvis and the thorax. We can thus see that a criterion for thorax-abdomen protection based solely on measurements taken at thorax level would be wholly unsuitable to account for the occurrence of the majority of abdominal injuries and their severity. A specific criterion for the protection of these various body regions is needed.

The fact that the armrest is associated with most abdominal injuries, with a possible role in 85% of cases supports the approach of severe injuries linked to two associated parameters: penetration and velocity. Previous experimental work shows that severe abdominal injuries with an AIS ≥ 3 value are observed for an abdominal intrusion of over 39 mm and an associated force of 450 daN(6). The EUROSID dummy was designed and instrumented on this basis. It is suggested that in future any dummy utilized for the evaluation of occupant protection in lateral impact should be instrumented so as to allow this specific abdominal risk detection.

The Criteria Available for Thoracic Protection

The aim of this chapter is to review the various criteria already utilized or proposed for the protection of the thorax in lateral impact and to analyze their strengths and weaknesses so as to arrive at the most pertinent one or ones. It must be noted that the conditions imposed by the various criteria generally result in different requirements, and in the case of lateral impact for example, on the setting up of the walls of the car and also on the construction of the thorax model(7).

The Data Available

Two types of lateral impact tests using human subjects were carried out:

- car-to-car lateral collisions, carried out mainly as part of the KOB programme(8,9,10). The subjects were fitted at thorax level with three-dimensional accelerometers on the dorsal column (on dorsal vertebrae T1, T4, T7 and T12) and with uni-directional accelerometers on the 4th and 8th left- and right-side ribs, as well as on the sternum.
- lateral drop tests of human subjects onto different types of materials (rigid or shock-absorbing). These drops were carried out from a height varying from 0,5 to 2 metres, and the subjects were also fitted with accelerometers.

The thoracic injuries can be of two types, associated or non-associated:

- bone related: fractures of the thoracic cage
- visceral: injuries to the lungs, heart, aorta . . .

Accidentology and biomechanical experiments supply the following data:

- a human being can tolerate a given number of rib fractures without risk of after-effects or an even smaller risk or fatality,
- injuries to internal organs are the most severe, but they are very rarely observed in the absence of rib fractures,
- a large number of rib fractures is necessary—to the order of at least 10—for there to be some risk of internal injury. This level corresponds to the appearance of flail chest. We remember that the risk of internal thoracic injuries associated with less than 4 rib fractures corresponds to 1,3% of occupants in the APR sample(11). On the other hand, in the greater part of cases (around 90%) of lateral impact with flail chest, associated internal injuries are observed.

The experimental data confirm these accidentological observations.

In the tests that we carried out (51 lateral impacts comprising drops and also simulations and reconstructions of car-to-car collisions), no case of internal injury was observed. Some of these impacts were, however, severe, the number of fractures varying between 0 and 25. If we add to this sample the two other series of reconstructions carried out by INRETS as part of the KOB programme(10), the resulting data show:

- in one case, tears to the diaphragm (AIS = 3) associated with 8 rib fractures (AIS = 3),
- in another case, a subject sustained tears to the parenchyma (AIS = 3) with simultaneous bilateral flail chest (AIS = 4)

So on a total of 14 reconstructions of car-to-car collisions using human subjects, only two cases of internal thoracic injury were observed with simultaneous rib fractures.

Since in accidentology no cases of internal injury were observed in equivalent real-world accidents, despite the presence of numerous rib fractures with one case of flail chest, we can assume that the number of rib fractures is a good indicator of thoracic injury severity; what is more, compared to the AIS, it has the advantage of providing a continuous scale of severity. On average, 87% of the fractures observed are located on the side of the impacted semi-thorax. When the AIS level is ≤ 3, all the fractures are located on the impacted side of the thorax; when the level of severity is higher (AIS ≥ 4), flail chest is often observed on the impacted side and several rib fractures can appear on the opposite side to the impact. One could however assume that the number of rib fractures on the struck side of the semi-thorax is sufficient to allow prediction of the impact severity(12); we shall return to this point later on.

The Parameters Available

In all the tests, accelerometric measurements are available, but they have not always been measured at the same locations, depending on the test configurations. When there was the opportunity of a convenient cadaver impact test, the subject instrumentation was increased in order to obtain, at the same time, the maximum number of measurements at thorax level. In order to measure a cadaver's kinematic response to blunt impact, a 12 accelerometer array was developed(13); the resulting data are being used in an attempt to correlate acceleration signatures with the actual injuries occuring in the impact test. These measurements were recorded by APR on a small sample of 10 cases.

In all cases, thoracic deflection measurements were carried out and it was possible to determine the deflection of the semi-thorax (with the exception of the car-to-car collision reconstructions where it was only possible to obtain the deflection for the whole thorax). A precise bone characterization of the subjects' thorax was carried out(14) and the knowledge of the relative resistances of the subjects' skeletons allows the results to be interpreted with greater accuracy.

Transverse thoracic acceleration at T4

Acceleration is the best known and most easily measurable parameter, and measurements of thoracic acceleration were for a long time used to characterize the severity of injuries to the thorax both in frontal impact and lateral impact. Figure 3 gives the total number of rib fractures as a function of the thoracic acceleration ($\gamma 3$ ms) measured at T4 (the subject's 4th thoracic vertebra). There is no obvious correlation between these two parameters which does not mean, a priori, that no conclusions can be drawn on the basis of accelerometric measurements as far as tolerance and protection criteria are concerned. It must be said that the maximum thoracic acceleration is influenced by the subject's anthropometric characteristics, and that no correction has been made here. The tests corresponding to different impact conditions are indicated by different symbols. We can see that, with the exception of one subject, who had weak bone characteristics (BCF = 0,55), no flail chest (AIS = 4) was observed for $\gamma 3$ ms < 50 g. The average of $\gamma 3$ ms accelerations corresponding to AIS = 3 was 49 g (for 9 subjects), with a minimum value of 32 g and a maximum value of 65 g.

These subjects had an average bone condition, midway between the average for the cadavers and that of the individuals who died immediately, whose bone condition is considered to be representative of the population exposed to risk.

Figure 4 shows the $\gamma 3$ ms at T4 in terms of BCF, with the numbers of rib fractures indicated nearside the points. Points corresponding to tests performed in identical conditions are connected by lines. In general, numbers of rib fractures increase when the BCF increases; it is not the same for γT4. In some cases

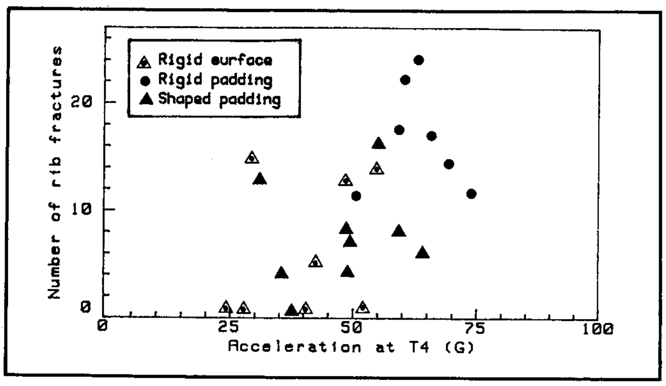

Figure 3. Number of rib fractures versus thoracic acceleration at T4

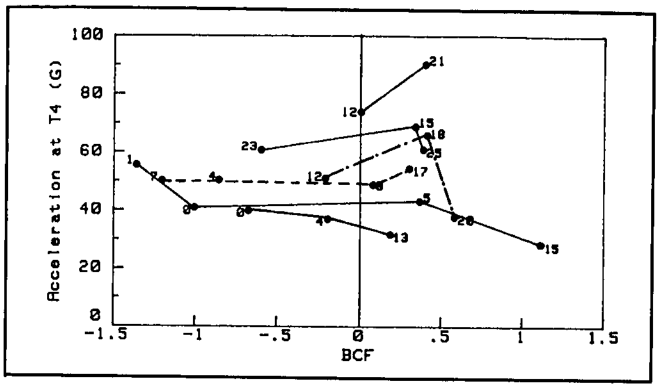

Figure 4. Thoracic acceleration at T4 versus BCF

great numbers of rib fractures, associated with positivalues of BCF correspond to γT4 values lower than for subjects tested in the same conditions but having better bone characteristics and a lower number of rib fractures; this is paradoxical and one can conclude that there is therefore no simple correlation between thoracic acceleration and the number of rib fractures, even if the subjects' bone condition is taken into account.

A protection criterion based on measurement at T4 cannot account for thoracic deformation and, furthermore, the injuries can occur well before the peak in the accelerometric response; this type of criterion would have a bearing solely on an acceleration measurement at one point of the dorsal column; now acceleration of the column can be induced by various mechanisms: forces transmitted through the thoracic cage during impact with the wall, forces transmitted through the abdominal viscerae during abdominal impact . . .

So acceleration of the column is, in general, only an indicator of the overall severity of the impact sustained by the whole of the individual and does not allow us to predict the risk of thoracic injury in isolation. The complexity of the deformations to the rib cage has led to the use of a greater number of acceleration measurements (12 thoracic accelerations array) to describe them. This factor also indicates the necessity for the measurements of deflection at several levels if possible.

BLUR

It has long been an established fact(15) that tolerance to deceleration increases as the time of exposure to deceleration decreases. Tolerance is thus based on velocity variations, since an equivalent variation in velocity can be achieved by reciprocal changes of level and acceleration time. It is this concept which was used in the definition of BLUR. BLUR was presented by D.H. Robbins at the 7th Conference on Experimental Safety Vehicles(13); in the tests performed by the HSRI, it proved to have a high correlation with injury levels, between given threshold values. This parameter is derived from accelerometric measurements performed on the thoraxes of cadavers subjected to side impacts; it is proportional to the logarithm of an integral taken over a specified number of points of an acceleration signal, rather than to such an integral itself:

$$\text{BLUR} = \ln\,[\max\,(|F1|,|F2|)]$$

where F1 = net velocity change achieved over 20 ms of maximum positive acceleration of upper rib, and

F2 = net velocity change achieved over 20 ms of maximum negative acceleration of upper rib.

It was possible to calculate BLUR for a given number of our subjects and the values obtained are shown in Figure 5 as a function of the number of rib fractures. Since the number of data available is small, it is not possible to establish a correlation between these two parameters. In two cases, for example, identical BLUR values (7.52) are associated with numbers of rib fractures recorded of 3 and 15 respectively; so BLUR therefore appears unsuitable to account for injury severity. It is necessary to take the subjects' bone quality into account.

Figure 6 shows BLUR as a function of the BCF bone characterization index and the corresponding numbers of rib fractures are indicated; the points corresponding to subjects exposed to identical impact conditions are joined up. We can see that, for a series of tests, the variations in the BLUR values are very small and are not significant enough to explain the large differences in the number of fractures; the BCF, on abscissa, on its own provides a much better interpretation of injury levels than BLUR does. BLUR here appears as an indicator of violence and does not allow us to account for the differences in injury levels sustained by the subjects.

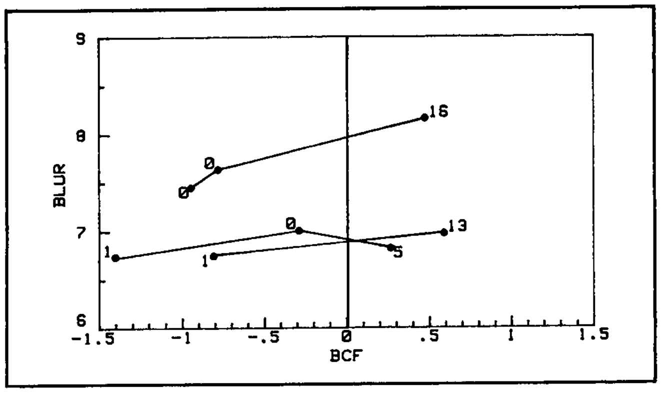

Figure 6. BLUR versus BCF

Average Power

Average Power is another parameter which has been used by certain authors in the prediction of thoracic injuries. The data already published(16) for thoracic AIS as a function of this parameter show a large amount of scatter.

Average Power is calculated in the following manner:

$$\text{A. P} = \frac{1}{T}\int_0^T \gamma(t).\left[\int_0^t \gamma(\zeta)d\zeta\ dt\right]$$

where γ is the transverse acceleration measured at the level of the fourth thoracic vertebra and T is the impact duration. We are here concerned with Average Power per unit of mass. As was done for BLUR, we studied the possible correlation between Average Power and the number of rib fractures. The results show that identical values of this parameter can be associated, as can BLUR, with very different numbers of fractures, and that it is necessary to take the subjects' bone quality into account to interpret the results. Figure 7 shows Average Power as a function of BCF for subjects who have undergone drop tests. The numbers of rib fractures are indicated beside each point and the cases corresponding to subjects tested under identical conditions are joined up.

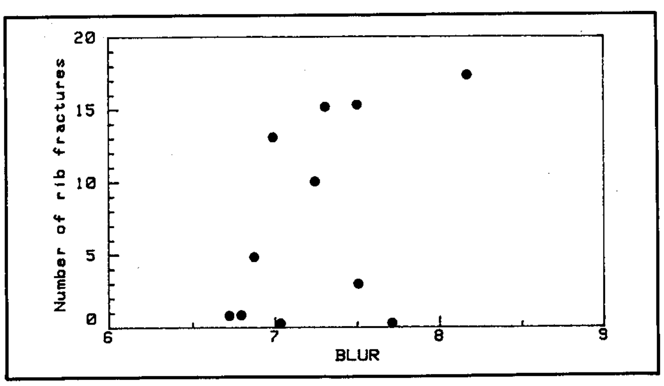

Figure 5. Number of rib fractures versus BLUR

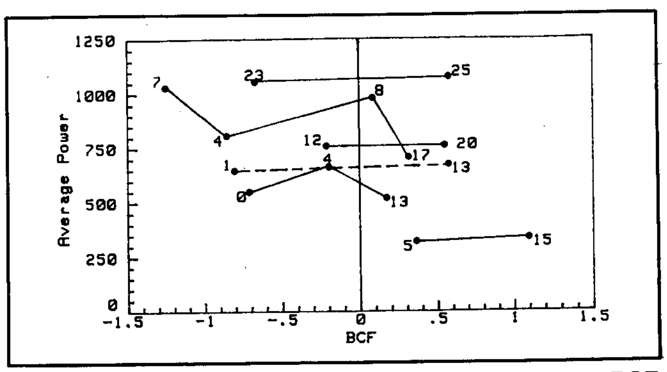

Figure 7. Number of rib fractures according to BCF and Average Power

As before, the differences in the number of fractures for subjects exposed to identical impacts are explained better by the BCF than by Average Power, which also shows itself to be an indicator of impact violence (but to a lesser extent than the BLUR) and not an indicator of injury severity for the subject tested.

Deflection

The thorax deflection caused by impact was measured optically by means of an intra-thoracic shaft equipped with optical test targets which crossed through the thorax. The maximum deflection which we measure is that which corresponds to the maximum crush of the thorax, and the relative deflection is the ratio between the deflection measurement obtained at mesosternal level and the width of the thorax at the same level measured before testing. Two types of deflection will be considered: the deflection of the whole thorax or the deflection of the semi-thorax, since the latter is not measurable in all cases. Figure 8 is a representation of the number of rib fractures as a function of the deflection of the whole thorax for overall lateral impacts, whether they are car-to-car collisions or drop tests onto padding or rigid surfaces. Deflection appears to be a good indicator of the severity of thoracic bone injuries, whatever the conditions of impact. Only one subject is set a little apart on this graph, with a relatively high number of fractures(15) for a small thoracic deflection (18%); this can be explained by the subject's very weak bone characteristics, well below average, with a BCF of 1,135. Figure 9 shows the correlation between the number of rib fractures and the BCF and figure 10 shows the relative deflection of the whole thorax for all the subjects and their bone characterization index, or BCF.

The points corresponding to subjects tested in the same conditions are joined up and the numbers of rib fractures are shown for each point. In one type of tests, the relative deflection increases as the BCF increases; it also increases as a function of the number of rib fractures, as was seen in figure 8. Since the number of rib fractures also increases with the BCF, a simultaneous knowledge of the relative deflection and the BCF index constitutes a good indicator of the risk of injury occurrence. If, for example, we consider only the subjects with a higher than average bone quality (that is to say, who have a BCF of less than 0), we can see that, if the threshold of 7 fractures to maintain the values already used in previous publications(10),(14) is considered as a tolerance threshold not to be exceeded for the thorax, it would be associated with a relative deflection of the thorax to the order of 30%, the tolerance of living persons being certainly slightly higher than this value.

Figure 11 shows the correlation between the total number of rib fractures and the deflection of the impacted semi-thorax, and figure 12 shows the number of rib fractures observed on the impacted semi-thorax as a function of its relative deflection; the correlation is a little better than it is with the total number of fractures.

This highlights the fact that the relative deflection of the semi-thorax gives a good description of the

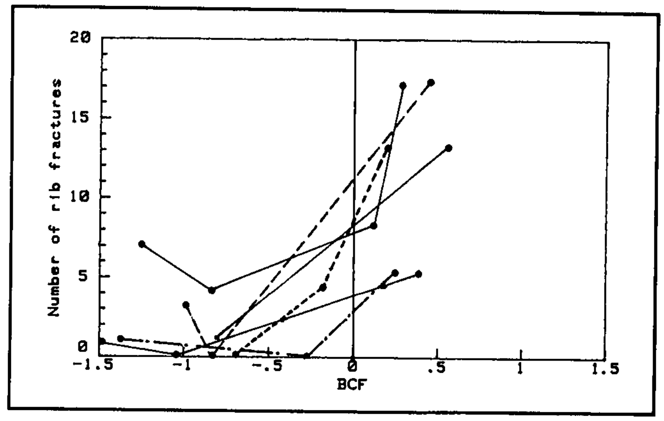

Figure 9. Number of rib fractures versus BCF

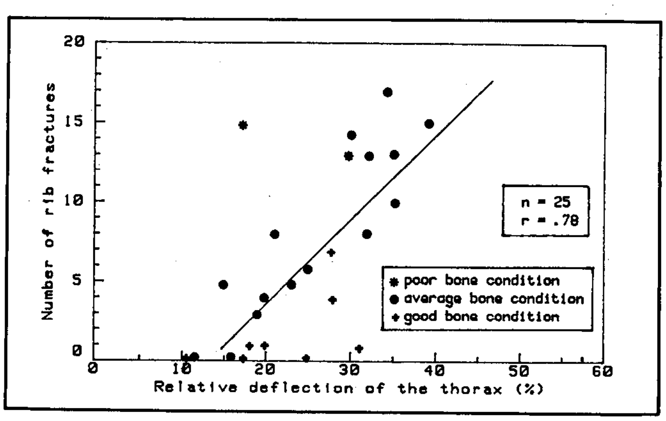

Figure 8. Number of rib fractures versus relative deflection of the whole thorax

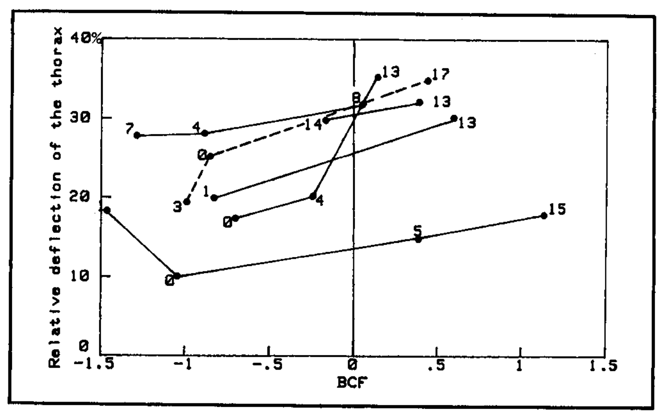

Figure 10. Relative deflection of the whole thorax versus BCF

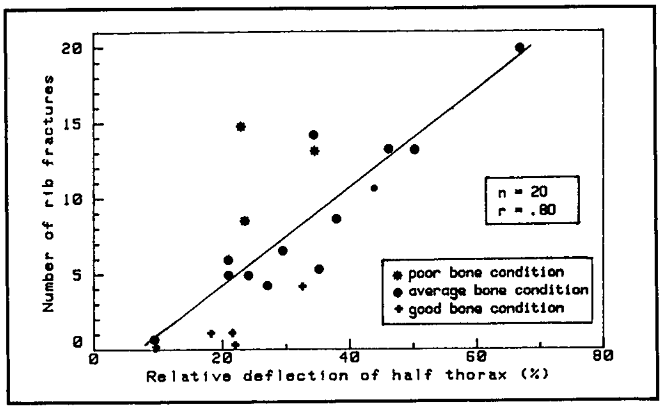

Figure 11. Number of rib fractures versus relative deflection of the half impacted thorax

severity of the impact to which the thorax is exposed. A relative deflection of the semi-thorax of around 35% can be considered as the maximum tolerable if we consider that the number of 7 rib fractures must not be exceeded; this number of rib fractures constitutes a threshold above which there is a risk of the occurrence of flail chest.

It must be noted that the correlation between the two parameters for deflection and number of rib fractures is not perfect, and that scatter appears in the two preceding figures; this can be explained by the differences in the subject' bone characteristics. We can in fact distinguish three groups of different bone conditions in these figures:

- the first group (11 on 20 cases) where the bone condition is close to the average for our population of 150 tests subjects: ($-0.5 <$ BCF < 0.5). These points are on the line of regression, or on one side or other of the line of regression,
- the second group where the bone condition is particularly weak compared to the first group; these points are located above the line of regression (too high a number of rib fractures),
- the third group, with a higher bone condition than the average for the subjects (BCF < -0.5). The corresponding points are located below the line of regression. The thoracic bone resistance in these subjects can be considered to be close to that of the living population exposed to risk. We can thus see that the subjects are spread out differently according to their bone characteristics and that it is necessary to take these characteristics into consideration if we wish to account for the scatter observed in figures 11 and 12.

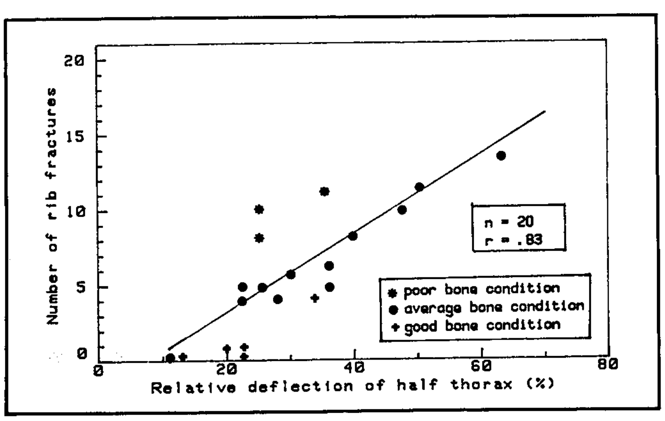

Figure 12. Number of rib fractures on the impacted half thorax versus relative deflection of the half thorax

The need for the BCF to be taken into consideration in order to improve prediction of the risk of thoracic injury is more especially important since the BCF index is strongly correlated with the maximum effort of static rib flexion before rupture; with an identical relative deflection, the number of rib fractures in the impacted semi-thorax will depend solely on its tolerance to flexion before rupture.

Taking these observations into account, a predictive function of thoracic injuries in side impact was established(12). This predictive function is called TIP (Thoracic Injury Prediction), and can be expressed as follows:

$$TIP = 0.2275\,\delta + 2.4824\,BCF - 1.098$$

where:

- TIP is the number of rib fractures observed to the impacted semi-thorax,
- δ is the relative deflection (0-100%) of the impacted semi-thorax,
- BCF is the subject's Bone Condition Factor.

The numbers of rib fractures observed on cadavers and those predicted by the above equation are illustrated in figure 13. This function's degree of significance was determined to be 1/1,000, which is highly significant.

An extrapolation of these predictive functions to living persons was made, based on a sample of 44 individuals who died from sudden death (traffic accident victims, suicides, etc) for which a BCF value was calculated as for the test subjects. The average age of these individuals was 42 years and their average BCF was -1.2. The latter value will thus be taken to establish the maximum admissible value for the protection criterion compared to the average population at risk. The TIP for a living person with average bone resistance (i.e. with BCF $= -1.2$), exposed to the same impact as test subjects, can be predicted by the following equation:

$$TIP = 0.2275\delta - 4.07688$$

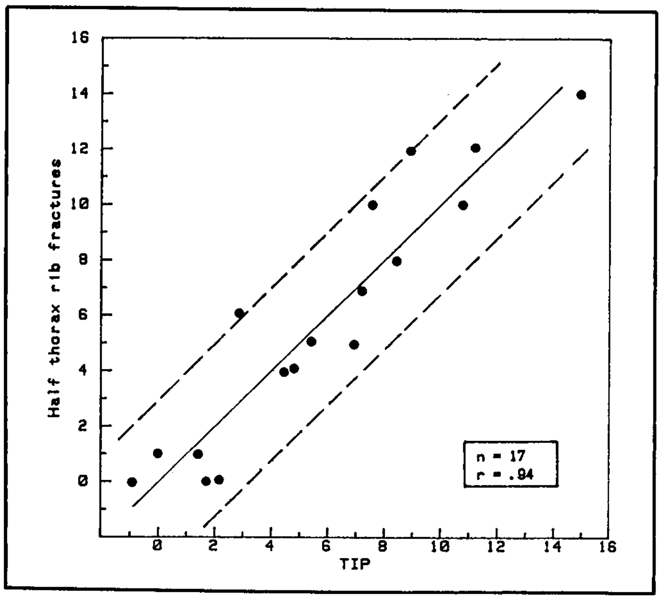

Figure 13. Number of rib fractures on the half impacted thorax versus TIP (Thoracic Injury Prediction)

Assuming that the dummy's thorax has a "human-like" response and considering, for example, that 5 rib fractures is a very acceptable limit (indeed, a very conservative limit, excluding all risk of flail chest) for thoracic injury severity, then the maximum acceptable relative deflection value for the dummy's semi-thorax will be 40%, which corresponds to a penetration distance of 60 mm for the 50th-percentile dummy. This value of 60 mm, the total deflection of the semi-thorax, must be differentiated from the value stipulated for a dummy, which will always be lower since it does not take the deflection of the soft tissue inside the impacted thoracic wall into account. (For example, a total deflection of 60 mm would correspond to 48 mm in terms of "internal" measurement of the deflection for dummies such as the APROD or EUROSID).

TTI

TTI was presented by the NHTSA in 1984(1) and the analytical process used by the authors in its development was a tentative interpretation of the existing data, guided by several underlying assumptions. During the last Stapp Car Crash Conference, a new definition was given for TTI(2); TTI is a predictive function based on the age of the subject, his weight, the maximum transverse accelerations measured on either the fourth or the eighth rib on the impacted side, and the maximum acceleration measured on the 12th dorsal vertebra. It has the following formula:

$$TTI = 1.4\ AGE + 0.5\ (RIBY + T12Y)\ \frac{Mass}{Mstd}$$

RIBY	=	maximum absolute value of acceleration of rib on struck side in a lateral direction after the acceleration signal has been filtered according to specifications
T12Y	=	maximum absolute value of the twelfth thoracic vertebra in a lateral direction, after the acceleration signal has been filtered according to specifications.
Mass	=	subject mass
Mstd	=	standard mass (165 lb = 75 kg)

In the NHTSA analysis, the concept of the Hard Thorax is used when considering injury severity level. The hard thorax includes not only the traditional thorax contents such as the heart, lungs and ribs, but also the thoracic spinal column and the abdominal organs contained inside the rib cage such as the liver, kidneys and spleen. Since the TTI was proposed, several teams have used it to specify the quality of the correlation between TTI and thoracic and/or abdominal injuries. We can point in particular to the important work carried out in 1985 by Klaus et al(17) which highlighted the lack of correlation and the great variance between data measured with the HSRI dummy and human subjects and thus the lack of suitability of this algorithm in the calculation of the theoretical TTI from data measured with this dummy.

Since the recent modification of the definition for TTI, we recalculated the values for this parameter for subjects for which we had all the available data. Figure 14 shows the number of rib fractures as a function of the TTI for Heidelberg data (shown by the circles) and our data (shown by the triangles). The relation observed is not good enough between these two parameters to consider the TTI as an acceptable indicator of the severity of the thoracic injuries, in terms of rib fractures. Only one indication is given: the number of rib fractures is less or more than 18 following that TTI is less or more than 190. As the BCF was not available for the Heidelberg tests, we investigated possible relation between TTI, BCF and

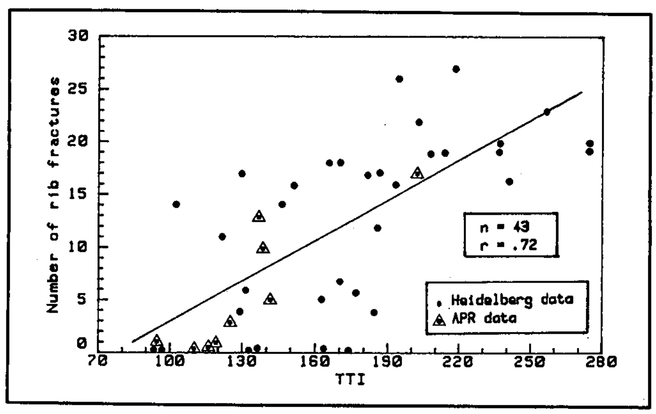

Figure 14. Number of rib fractures versus TTI

numbers of rib fractures only for our tests. It is a small sample and the results are indicated on figure 15. TTI increases when BCF increases. So TTI, associated with an index of bone characterization of the subjects, could probably improve the prediction of number of rib fractures. Given the size of the available sample, it was not possible to undertake a more in depth statistical analysis.

Figure 16 shows the AIS for the "hard thorax" as a function of the TTI for the overall Heidelberg subjects and our subjects. The points corresponding to our data integrate well with the Heidelberg data but, instead of reducing the scatter, they tend to accentuate it.

The overlaps of the different AIS values for the same TTI value are very important. We do not have an index of bone characterization for the subjects to interpret these results and although age comes into the calculation of TTI, it is not a sufficiently precise indicator of bone resistance to allow us to explain the scatter observed. Several other criticisms can be made concerning the TTI and its unsuitability for the EUROSID dummy. Detailed analysis of a NHTSA's team last publication on this subject(2) shows that:

- the γ lower rib/γ upper rib ratio varies with the location of the injuries, according to whether they are thoracic injuries (1.52) or abdominal injuries (1.95),
- the average value for this ratio is 1.73 for cadavers in the tests carried out at 31 mph whilst it is only 1.05 for the 4 tests with the SID dummy, tested under the same conditions. The authors recognize the problem and propose a correction in order to adjust the SID upper rib response; applying their scaling formula, they obtain new rib responses, which lead to an average value of 1.32 for this ratio. This value is still a long way from the values obtained with human subjects under the same conditions and this puts into

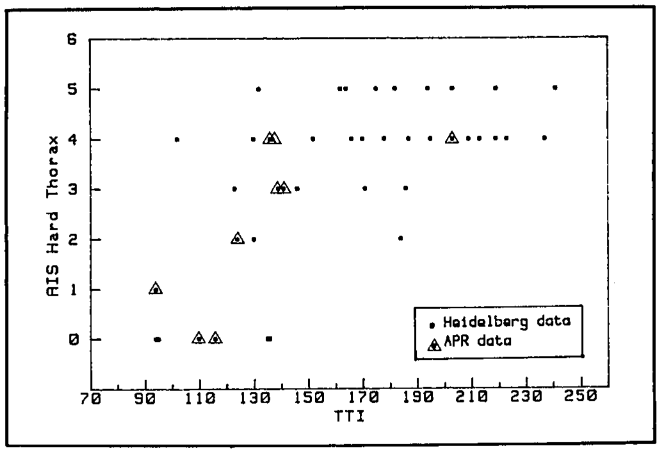

Figure 16. AIS for the "Hard Thorax" versus TTI

questions either the SID's biofidelity if one were to consider TTI as a good criterion, or the validity of TTI itself as a predictor of the thoracic injury level resulting from a lateral impact.

TTI as a predictor of "hard thorax" injuries including some abdominal injuries (liver, spleen and kidneys).

As it is shown elsewhere(18), the lowest part of the rib cage is considered by anatomists as softer than the part corresponding to the upper part, from the first to the seventh rib. It is difficult to consider that some abdominal organs could be protected by the so-called hard thorax. In any case, the figure 16 includes liver, spleen, and kidney injuries in the AIS of hard thorax and the correlation with TTI is poor. If TTI was also a good predictor of these abdominal injuries, it would be possible to get a good correlation between TTI and AIS for abdomen itself, when restricted to liver, spleen and kidney injuries. It is not the case as can be seen on figure 17; this specific analysis is worse than for hard thorax as a whole.

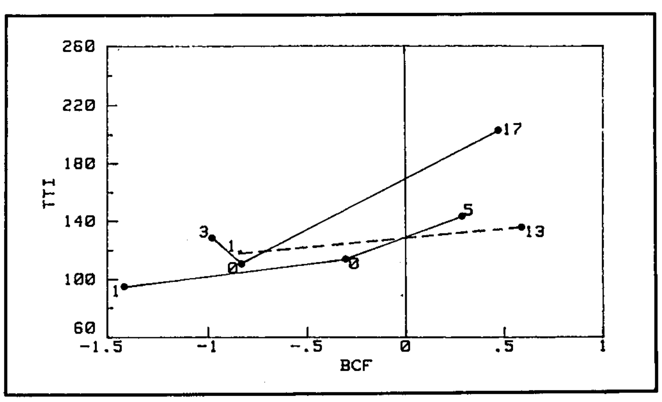

Figure 15. Number of rib fractures according to BCF and TTI

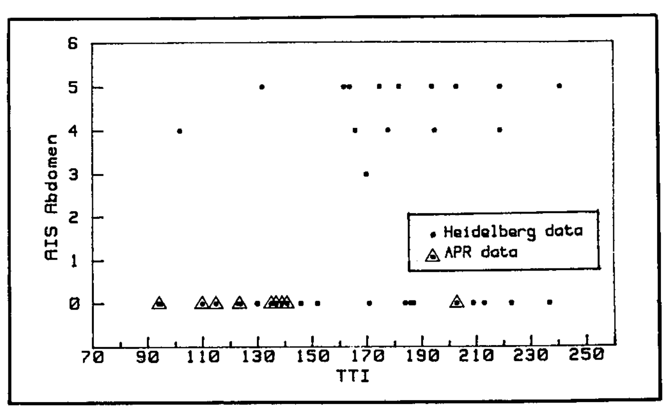

Figure 17. AIS Abdomen Versus TTI

Discussion

The review which has been made of the different protection criteria which have been, and still are, used by various authors, shows that the relative deflection of the thorax is the best criterion if we consider the number of rib fractures as an indicator of thoracic injuries. Our goal was to succeed in determining the best thoracic protection criterion and, at the same time, in determining whether a more continuous injury severity indicator than the AIS could be related to parameters resulting directly from measurements. The criticism that can be made of any criterion based on accelerometric measurements at thorax level in lateral impact is the following: thoracic acceleration alone does not allow us to account for both deformation of the side-wall and deformation of the thorax. The same thoracic acceleration value can be obtained with a too rigid padding and a severely injured human thorax, or with a normal padding and a very undeformed because too stiff dummy thorax (figure 18).

This is a fundamental criticism of a lateral protection concept based on thoracic acceleration. This confusion between the deformation of the thorax and that of the impacted wall cannot occur with the measurement of deflection of the thorax itself. The age of the subject as an indicator of his skeletal resistance is taken into account in the formula for TTI; but this is not enough to account for the scatter between the subjects, as the correlation between age and the BCF bone resistance index is poor, only 0.60(14).

An exhaustive analysis requires a knowledge of the BCF and we thus obtain in an unfortunately small sample, a best correlation between TTI and the number of rib fractures when we take the BCF into account.

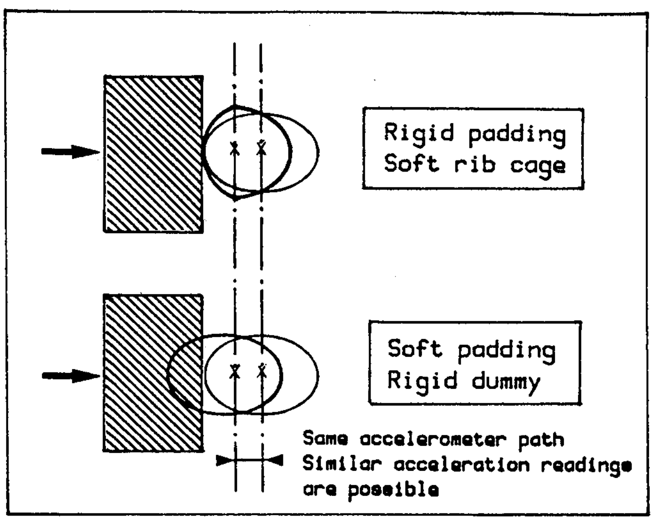

Figure 18. Two different situations where similar acceleration readings are possible

A further difficulty connected to acceleration measurements lies in the fact that, in the case of rib fractures, the accelerations measured decrease abruptly when all the energy is not yet dissipated in the rib cage. This is not the case with thoracic deflection which continues to increase following the first rib fractures up until the total dissipation of the energy due to the impact.

In an earlier publication(7), linear regressions were used to correlate the number of rib fractures to several parameters expressing impact severity; these relationships took into account the subject's bone characteristics.

The global sample analyzed corresponded to 38 lateral impacts, but certain equations were established for a limited number of tests because of the unavailability of all the data at once.

The main relationships used to predict the number of rib fractures using the BCF were the following:

$$NRF = -2.25 + 2.36BCF + 0.29\delta, \; r = 0.80, \; n = 15$$

$$NRF = 3.62 + 0.5BCF + 0.03R4, \; r = 0.72, \; n = 9$$

$$NRF = 4.69 + 5.12BCF + 0.072T4, \; r = 0.58, \; n = 38$$

Despite the relatively high correlation coefficients of these relationships overall, some of them are not significant enough, particularly when the size of the sample on which they were established and the Fischer test are taken into account.

This analysis showed however that the best correlation with the number of rib fractures was obtained with deflection, which has since been confirmed by the TIP (Thorax Injuries Predictor), then with acceleration at R4, although the latter was not sufficiently significant. The worst result was obtained with T4, with a correlation of only 0.58 for a sample of 38 subjects.

Figure 19 shows the AIS thorax as a function of wall velocity at thorax level for 80 occupants, involved in real-world lateral collisions, who directly sustained intrusion. There is no correlation between these parameters and the overlap between the different AIS values is very large: for the same wall $\triangle V$ value, we can find AIS values of between 0 to 5. Such a parameter cannot be used to predict injury levels. The same conclusion was reached for the TTI, as we saw in figure 16. In fact, as several previous analyses suggested, measurements based on accelerations can be, at best, only an indication of violence but in no case an acceptable indicator of thoracic injury.

In this context, the TTI concept and the SID, whose principal element it is, cannot be considered to be acceptable in any future regulation. No specific

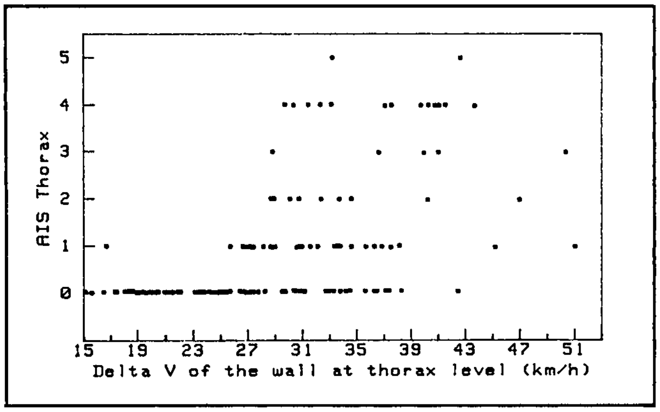

Figure 19. AIS thorax versus △V of the wall at thorax level

criterion exists for the abdomen despite the priority which should be given to it, and the absence of an abdominal segment whose biofidelity could be recognized is an important handicap for this dummy. The SID and TTI concepts are presented as if the accelerations measured at thorax and pelvis levels were sufficient to account for thoracic, pelvic and abdominal injuries.

Conclusions

- Accidentological analysis of car-to-car lateral collisions as well as experimental simulations with cadavers show that severe abdominal injuries are more frequent than severe thoracic injuries.
- The presence of an armrest was associated in the great majority of cases with severe abdominal injuries, with a possible role in 85% of cases.
- Severe abdominal injuries associated with thoracic injuries without pelvis fracture are the least frequent in the APR sample (17%). The majority of severe abdominal injuries are either isolated (22%) either associated simultaneously to pelvic fractures and rib cage fracture (39%). This implies that a protection criterion based solely on thoracic acceleration measurements cannot account for the occurrence and severity of the majority of the *abdominal* injuries.
- TTI appears to be a weak predictor of thorax injuries and injuries of the upper part of abdomen (liver, spleen, kidneys), whether these injuries are associated in the AIS of the "hard thorax" or considered separately: rib cage on one side, in terms of number of rib fractures and AIS of the upper part of the abdomen on the other side. The correlation between TTI and number of rib fractures could probably be improved by taking into account the Rib Bone Condition Factor (BCF); this could not be checked because of lack of available data. In these conditions, the TTI cannot be considered as an acceptable protection criterion. It appears that the criterion of thoracic deflection, associated to a specific abdominal protection criterion, constitutes the best approach of prediction of THORAX-ABDOMEN injuries in side impact.

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Results of Full-Scale Tests with EUROSID Under Different Test Conditions

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Abstract

The European Side Impact Dummy "EUROSID" has been extensively tested in European Laboratories over recent years. Together with and under the leadership of TRRL in Great Britain, INRETS and APR in France and TNO in the Netherlands, the Federal Highway Research Institute (BASt) has participated in a research project called "EUROSID Evaluation Program" which was partly sponsored by the EEC in 1986.

The objective of the BASt's investigation was to examine the behavior of the EUROSID—here especially the repeatability, the sensitivity, the durability and the dummy handling—in full-scale tests under different test conditions.

Seven side impact tests were conducted with a Volkswagen Golf I (Rabbit) as test car and a movable deformable barrier with EEVC IV element. The test conditions selected were the European proposal with the 90 degree impact angle and the US proposal with the crabbed mode of deformable barrier, both equipped with an EEC barrier face.

The results show that the dummy is able to distinguish between different impact conditions, that the repeatability of the test results is satisfactory and no problems with the durability (and handling) occurred in tests up to 55 km/h. The EUROSID therefore appears to be suitable for use in procedures for testing the lateral protection of vehicles.

The European Side Impact Dummy "EUROSID"

The EUROSID dummy was developed jointly by TRRL (with Ogle), INRETS, TNO and APR between 1984 and 1985 after the end of the EEC Biomechanics Program. A full description of the dummy may be found in[1] and[2] and is also given at this conference. The presentation of the EUROSID took place in December 1986 in Brussels. The papers presented then are now in the process of being printed by the EEC. A detailed description will therefore not be given in this paper. Figure 1 shows the EUROSID dummy.

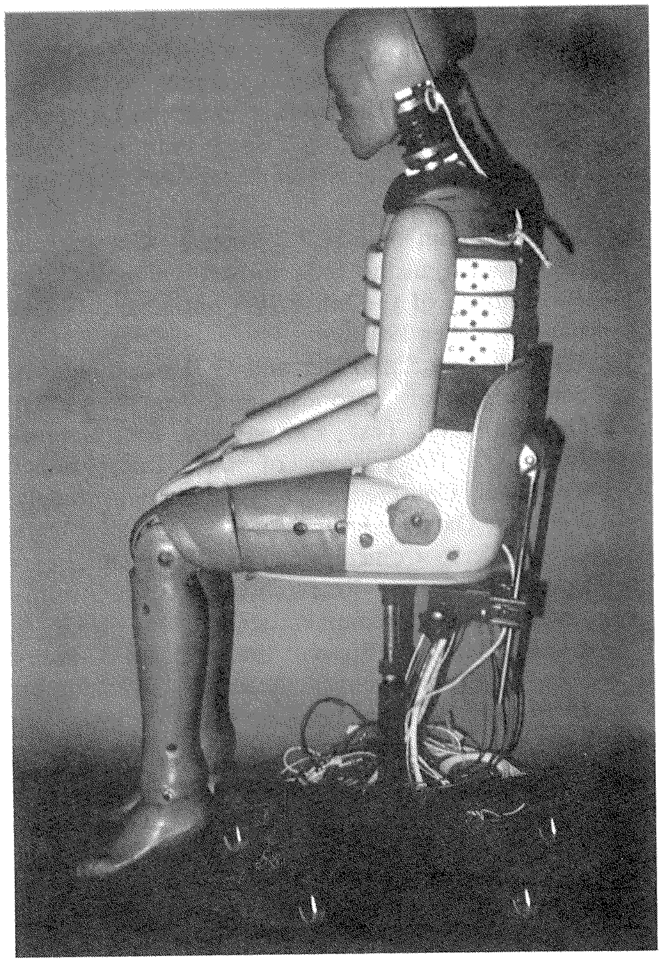

Figure 1. "EUROSID", the European side impact dummy

The following variable can be measured on the EUROSID:

- Head acceleration, triaxial
- Chest acceleration, triaxial
- Chest deflection (for each of the three ribs)
- Rib acceleration (uniaxial, for each of the three ribs)
- Abdomen load (on/off contact switch)
- Pelvis acceleration, triaxial
- Forces on each ilium (strain gauges)
- Forces on the pubic symphysis (force transducer)

The individual measurement sensors should be calibrated before commencement of the test series. The measuring channels of the EUROSID dummy should be filtered as follows /SAE J211b/:

Instrumentation CFC (Channel Frequency Filter Class)

• Head: triaxial accel. c. of g.	1000
• Chest: triaxial accel. c. of g.	800
rib accel. 3 x	180
deflection transducer 3 x	180
• Abdomen: on/off switch 3 x	1000
• Pelvis: triaxial accel. c. of g.	180
force transducer	600
strain gauge (2 x)	600

In March 1987, the EEVC ad hoc group on Side Impact Dummy Development provisionally suggested the following EUROSID measuring values as tolerance limits corresponding to AIS 3 for accident victims:

HEAD:	HIC [1000]*
CHEST:	DEFL [max 40-45 mm]*
ABDOMEN:	CONTACT Y/N [4500 N]*
PELVIS:	ILIUM FORCE [10 kN]*
	PUBIS SYMPH. FORCE [10 kN]*

*Values in square brackets are suggestions of the EEVC ad hoc group. They will not be finally defined before the end of 1987.

Test Method

The 7 full-scale vehicle tests were conducted with a Volkswagen Golf Type I (Rabbit, two-door) as the test vehicle. The colliding car was simulated by a movable deformable barrier with an EEVC-IV element on the front. Two different test constellations were used. A full description of these may be found in /3/and /4/. The most important parameters are shown in Table 1.

Table 1. Test parameters.

Test constellation / Barrier	90° (EEVC) mode (5 tests)	Crabbed (NHTSA) mode (2 tests)
Impact angle	90°	63°
Velocity	50 (45, 55) km/h	54 (lateral portion = 50) km/h
Mass	950 kg	1100 kg (as per compromise proposal of the CCMC) /4/
Ground clearance	250 mm*	250 mm
Face (deformable element)	EEVC IV	EEVC IV
Impact point	R point	37" in front of centre of wheel base

* The ground clearance requirement was increased to 300 mm by EEVC decision in November 1986 after these tests had been carried out.

The test vehicles were second-hand Type I VW Golfs (built in 1978 and 1979). The test weight (including measuring equipment but without dummies) was 800 kg. Different velocities were used in the tests. The test series was as follows:

Table 2. Test sequence.

Test No.	ESID 1	ESID 2	ESID 3	ESID 4	ESID 5	ESID 6	ESID 7
Velocity km/h	45	50	50	50	55	54	54 (corresponding to 50 km/h lateral velocity)
Impact angle	90°					63°	

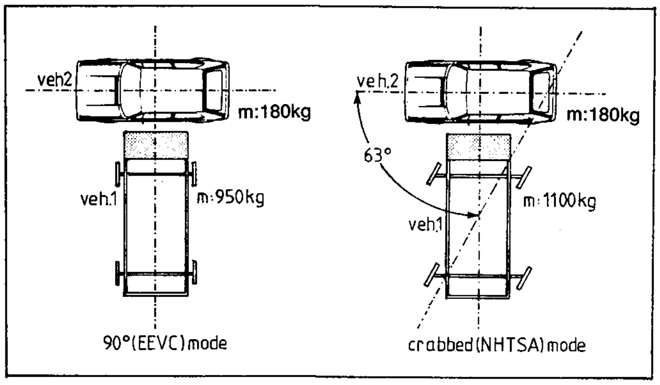

Figure 2. Test constellations

The following dummy criteria can be examined by comparing the following tests:

- Repeatability: ESID 2, ESID 3, ESID 4 and ESID 6, ESID 7
- Sensitivity: ESID 1, ESID 2,3,4; ESID 5
- Comparison of test constellations: ESID 2,3,4; ESID 6,7
- Handling and durability: all tests
- Biofidelity: none of these tests

The test vehicles were equipped with standard measuring sensors. However, since the aim of this investigation was to test the EUROSID dummy, these car measurement results will not be analysed any further in this report.

The surface of the side of the vehicle was marked with targets points. The position of the measurement points on the vehicle were measured before and after the crash. The difference gave the permanent vehicle deformation.

The driver's seat was adjusted so that the H point lay 50 mm in front of the R point. The angle of the seat back rest was 25 degrees. The EUROSID dummy was placed in the driver's seat, pressed firmly into the seat and its hands attached to the steering wheel with adhesive tape.

If different test vehicles are used, the angles of the upper arm relative to the upper torso centre line vary in accordance with the position of the steering wheel and length of the seat guide rails. As a result, the arm of the impacted side may either come to rest in a protective position in front of the ribs, or the arm may be outstretched, leaving the ribs, which are unprotected, to be hit directly by the pentrating door. It goes without saying that the dummy measurement values for these different cases will vary. In future tests or regulations, it is therefore advisable to fix the angle of the upper and lower arm and leave the hands in an open position on the steering wheel. A proposal for the EUROSID seating procedure was made in the meantime, but it is still under discussion.

Test Results

Car Deformations

The test velocity was maintained to an accuracy of 1% in all tests. The vehicle deformations for tests with the same velocity (ESID 2,3,4) are very similar.

Higher test velocities naturally led to higher vehicle deformations. Consequently, the door deformation—measured in the middle of the door—was about 10 cm higher for the 55 km/h test than for the 45 km/h tests.

The car deformation patterns are similar in all respects, apart from the fact that the "crabbed test constellation" initially also involves a deformation of the A pillar of this special test car.

After measuring the vehicle deformation it becomes clear that the dents caused to the door—particularly in the area where the dummy is struck—are considerably less deep for the crabbed constellation than for the 90 degrees constellation.

The effect of this behavior on the dummy load will be described later. The film recordings show that for both test configurations the door is deformed almost parallel to the central longitudinal plane of the vehicle at the moment the dummy is struck.

Dummy Kinematics

The film recordings show that, because of the dummy seating position, the arm is struck first by the penetrating door. The lower arm is pushed downwards in front of the abdomen by the lower side window frame. This also places a downward load on the shoulder. The upper part of the body bends in the direction of impact, the head moves towards the side window and then leans out of the window with the neck bent almost at right angles to the body. The following neck angles relative to the upper part of the body were determined by film analysis:

Table 3. Neck bending angle of the EUROSID.

		Constellation I				Constellation II	
	ESID 1	ESID 2	ESID 3	ESID 4	ESID 5	ESID 6	ESID 7
	45 km/h	50 km/h			55 km/h	54 km/h	
Angle of neck bending	90°	95°	92°	91°	111°	?*	92°

* Could not be evaluated

It can be seen that the neck angle is governed by the velocity. It was possible to make an accurate study of the behavior of the shoulder because in two tests (one with test constellation I and one with test constellation II) the dummy was tested without the thorax jacket. The shoulder only moves forwards at a very late stage. In fact, no movement of the shoulder is required because of the low position of the lower window frame of this car make. Thus, the dummy kinematics of the "crabbed tests" were very similar to those of the "EEVC tests". The two lower ribs were deflected by the door while the upper rib was subject to loading by the arm. In test 2, the upper rib was also subject to loading by the door because the seat was too soft due to the age of the car and the seating position of the dummy was therefore too low. The head did not strike any part of the vehicle because the B pillar of the vehicle is positioned too far towards the rear (2-door car). The abdomen was not subject to loading of the arm rest because it is attached well towards the front of the door.

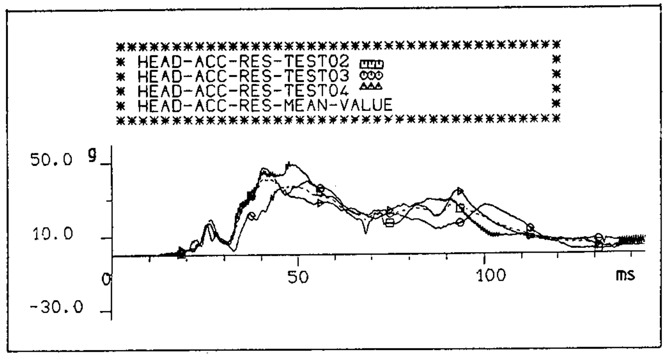

Figure 3. Resultant head accelerations

Dummy Measurement Values for Repeatability Tests

As in the preceding section, this section also uses the various test series as a basis for discussing the different aims of the tests.

The repeatability tests are the tests 2, 3 and 4. They were carried out at *50 km/h* and with the *EEVC Constellation*. Table 5 shows the results of the EUROSID measuring values. Since the maximum and 3-ms values referred to (slightly) different times, the mean values for tests 2, 3 and 4 were ascertained for each moment of time and the 3-ms value ascertained from this curve (shown as a broken line in the following diagrams) for comparison purposes. Column 4 does therefore not refer to the mean value of the 3-ms values for the different tests.

Table 4. Dummy measurement values for repeatability tests (50 km/h, constellation I).

			ESID 2 50 km/h 90° max.	ESID 2 3 ms	ESID 3 50 km/h 90° max.	ESID 3 3 ms	ESID 4 50 km/h 90° max.	ESID 4 3 ms	MEAN VALUES ESID 2, 3, 4 3 ms (time)
HEAD	ACCEL. /g/	Y	34.0	32.1	22.1	21.1	28.0	27.5	23.0 (40)
		Z	39.6	38.2	33.1	31.9	34.0	30.1	25.1 (40)
		RES	50.3	47.1	39.8	38.0	47.0	43.3	39.9 (40)
		HIC	300		217		277		265
CHEST	ACCEL. /g/	Y	84.3	74.1	59.5	57.4	75.7	71.8	67.9 (33)
		RES	84.9	74.3	64.5	59.4	74.3	72.3	69.3 (33)
		SI	485		341		415		413
		RIB U	125.5	98.9	107.7	85.8	128.9	103.2	94.2 (27)
		RIB M	284.1	146.7	136.4	108.6	265.3	131.7	139.9 (26)
		RIB L	259.8	135.5	115.3	97.2	254.7	99.1	108.7 (24)
	DEFL. /mm/	RIB U	45.5		29.9		28.9		32.5 (38)
		RIB M	42.0		31.9		36.5		36.5 (38)
		RIB L	32.5		38.6		41.6		38.5 (33)
PELVIS	ACCEL. /g/	Y	89.8	83.3	82.7	74.1	83.0	74.4	77.6 (26)
		RES	102.5	86.3	90.3	78.1	85.2	79.2	79.9 (26)
	FORCE /kN/	PUB.S.	5.3	5.0	5.6	5.1	4.5	4.2	4.7 (27)
		ILIUM L	1.3	1.2	1.2	1.0	0.9	0.9	1.0 (29)
		ILIUM R	1.5	1.5	0.6	0.6	0.5	0.4	0.7 (79)

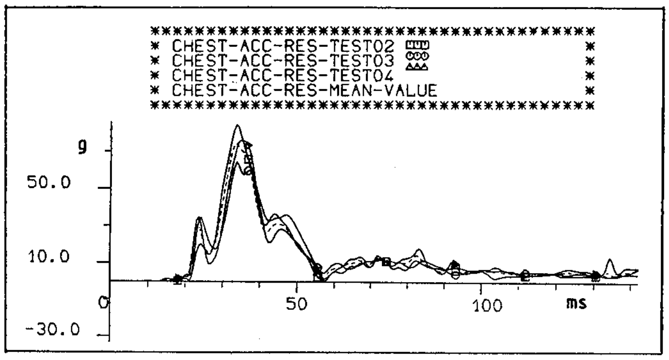

Figure 4. Resultant chest accelerations

Normal deviations of the measurement values—i.e., within the range familiar from frontal collision tests—were recorded for the head, chest and pelvic c.g. accelerations and for the pelvic forces. The right ilium was not subject to load. The touch-sensitive switches on the abdomen were not actuated—i.e., the force remained below 4500 N. Noticeable differences were recorded for the rib loads.

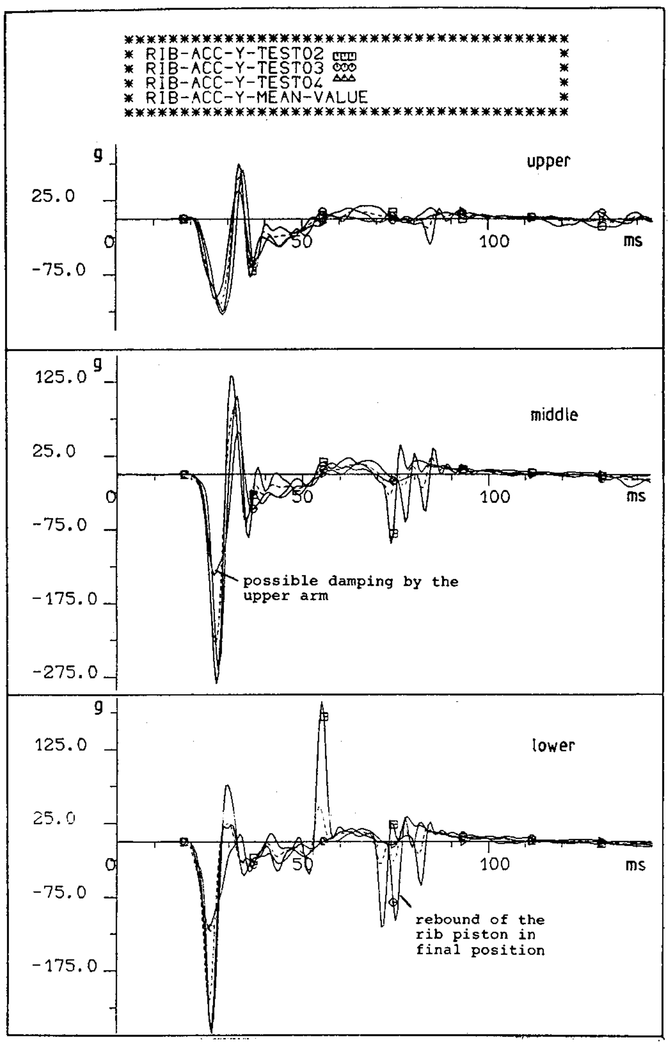

Figure 5. Rib accelerations

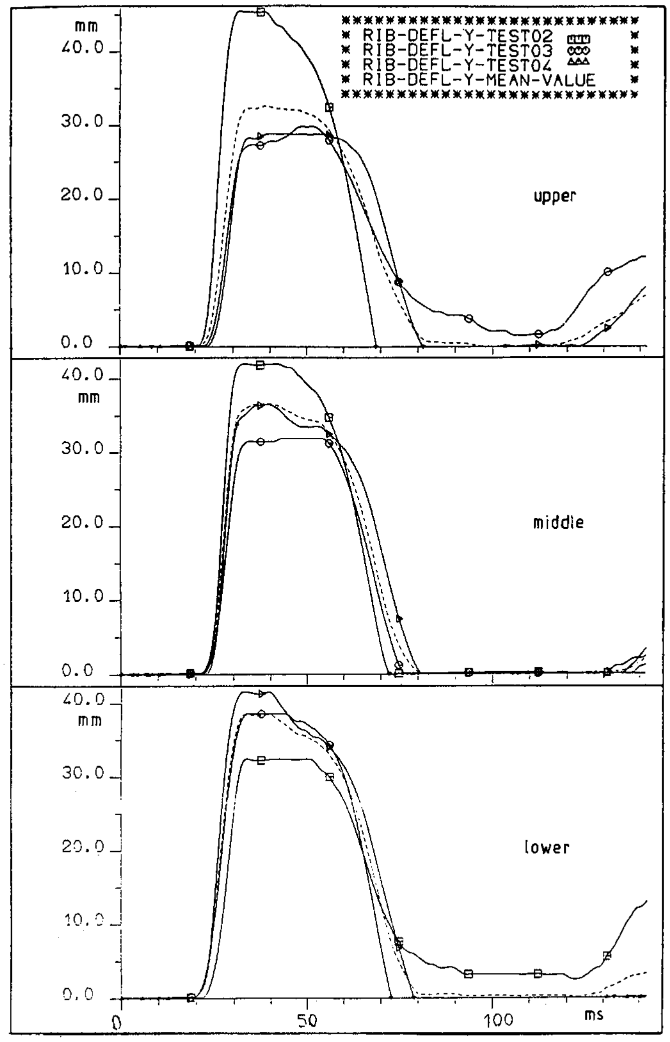

Figure 6. Rib deflections

As already described earlier, the sitting position of the EUROSID dummy was lower in test 2—i.e., the upper rib (RIBU) was subject to loading by the door. This is also reflected in the rib deflection (DEFL.), although not necessarily in the rib acceleration. Tests 3 and 4 show very similar results for rib deflection. The largest differences between tests were recorded for the rib acceleration. The highest acceleration

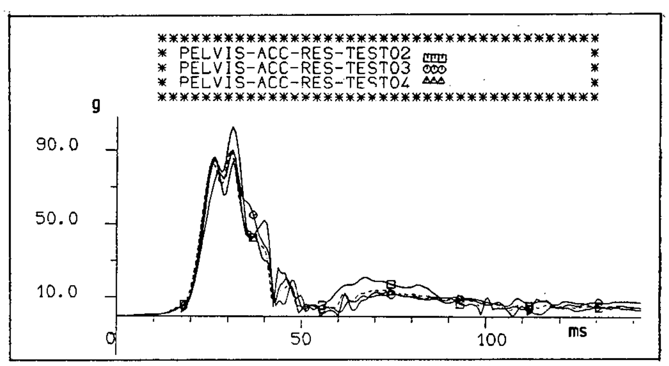

Figure 7. Pelvis accelerations

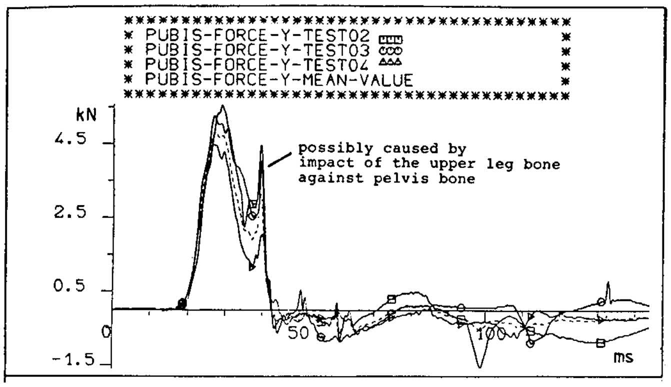

Figure 8. Pubis symphysis forces

recorded was on the middle rib in contrast to the rib deflection where the highest values were recorded for the lower rib. It is therefore difficult to establish a limit value for rib acceleration.

The following graphs of dummy measurement values show the individual signals and the high degree of congruency of the curves.

The measurement channels in which no measurement values of any consequence were recorded have been excluded in order to provide greater clarity.

The two tests with a *velocity of 54 km/h* using the "crabbed *Constellation II*" (tests ESID 6 and ESID 7) can also be compared with one another to examine the reproducibility of the test results. Table 6 shows the measurement results.

All measurement values show a high degree of congruency, the only exception being once again the values for the rib accelerations. Here, too, the highest load was also recorded for the middle rib, while the highest load suffered in the case of deflection (as in tests 2, 3, 4) was recorded for the lower rib. The

Table 5. Dummy measurement values for repeatability tests (54 km/h, constellation II).

			ESID 6* 54 km/h crabbed max.	3 ms	ESID 7 54 km/h crabbed max.	3 ms	MEAN VALUES ESID 6, 7 3 ms (time)
HEAD	ACCEL. /g/	Y	23.5	22.5	18.0	17.1	20.0 (37)
		Z	31.7	31.7	33.6	32.6	31.8 (51)
		RES	47.2	39.0	36.4**	35.8	34.7 (50)
		HIC	179		141		160
CHEST	ACCEL. /g/	Y	56.8	52.4	55.4	50.1	50.3 (33)
		RES	57.0	52.8	55.6	51.1	51.2 (33)
		SI	265		268		266.5
		RIB U	118.7	83.0	90.5	65.9	70.9 (24)
		RIB M	161.5	120.6	131.0	102.0	107.3 (24)
		RIB L	158.9	103.9	115.9	91.2	97.1 (22)
	DEFL. /mm/	RIB U	27.6		30.6		29.1 (46)
		RIB M	29.1		33.7		31.3 (47)
		RIB L	34.6		40.2		37.4 (37)
PELVIS	ACCEL. /g/	Y	70.2	62.0	80.6	70.4	67.0 (28)
		RES	75.1	63.1	85.0	73.4	70.3 (28)
	FORCE /kN/	PUB.S.	3.7	3.3	4.8	4.0	3.6 (26)
		IL.L.	0.7	0.7	1.0	0.9	0.7 (30)
		IL.R.	0.4	0.3	0.5	0.4	0.2 (98)

* signals only up to 65 ms available (defect)
** without x-axis (defect)

Table 6. Results of the sensitivity tests: (constellation I, 45,50,55 km/h).

			ESID 1 45 km/h 90° 3 ms (time) values	MEAN VALUES OF ESID 2, 3, 4 50 km/h 90° 3 ms (time) values	ESID 5 55 km/h 90° 3 ms (time) values
HEAD	ACCEL. /g/	Y	23.6 (44)	23.0 (40)	26.9 (38)
		Z	30.7 (55)	25.1 (40)	defect
		RES	38.9 (52)	39.9 (40)	defect
		HIC	167	265	defect
CHEST	ACCEL. /g/	Y	44.4 (41)	67.9 (33)	62.6 (29)
		RES	45.5 (41)	69.3 (33)	64.8 (29)
		SI	219	413	390
		RIB U	48.5 (27)	94.2 (27)	113.1 (24)
		RIB M	66.3 (29)	139.9 (26)	135.7 (24)
		RIB L	110.0 (28)	108.7 (24)	128.8 (24)
	DEFL. /mm/	RIB U	14.4 (36)	32.5 (38)	32.4 (30)
		RIB M	19.4 (36)	36.5 (38)	39.2 (29)
		RIB L	30.2 (46)	38.5 (33)	44.7 (28)
PELVIS	ACCEL. /g/	Y	73.4 (35)	77.6 (26)	90.8 (23)
		RES	76.9 (35)	79.9 (26)	95.2 (23)
	FORCE /kN/	PUB.S.	4.4 (32)	4.7 (27)	5.6 (24)
		IL.L.	0.9 (32)	1.0 (29)	1.2 (23)
		IL.R.	0.2	0.7 (79)	0.5 (87)

contact switches on the abdomen were not actuated in these tests either.

Dummy Measurement Values for Sensitivity Tests

The measurement values (3 ms) for the tests ESID 1 (45 km/h), the mean values from the tests ESID 2, 3, 4 (50 km/h) and the measurement values for test ESID 5 (55 km/h) are compared in Table 7.

The measurement values tend to increase as the test velocity increases. It should also be noted that there is a reduction in the time taken until the maximum values are reached or until the beginning of the 3-ms values. However, the chest accelerations are almost

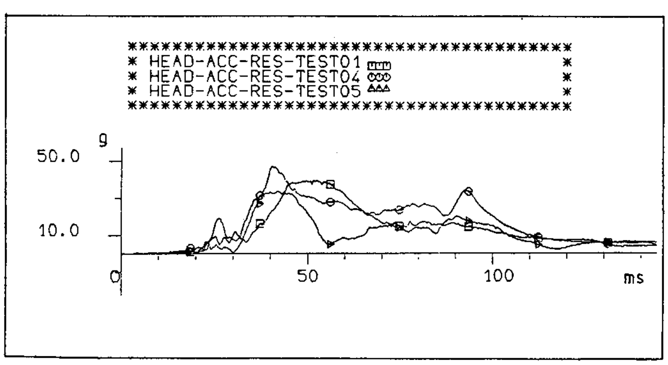

Figure 9. Resultant head accelerations

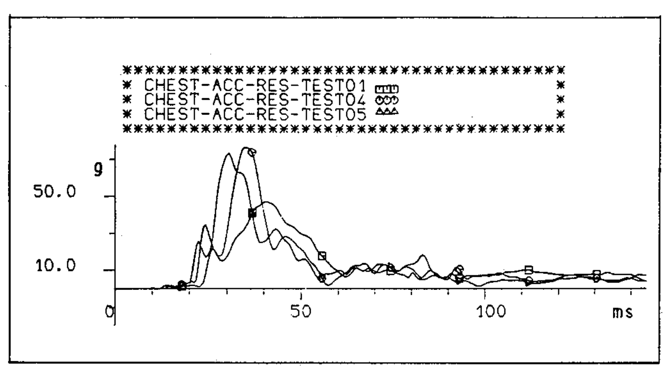

Figure 10. Resultant chest accelerations

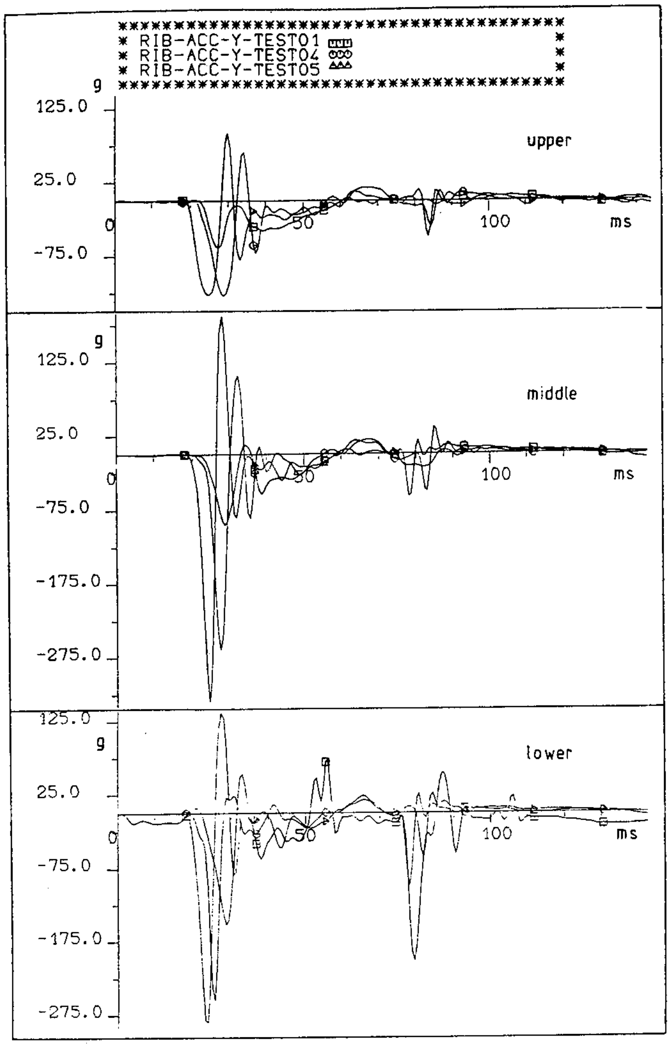

Figure 11. Rib accelerations

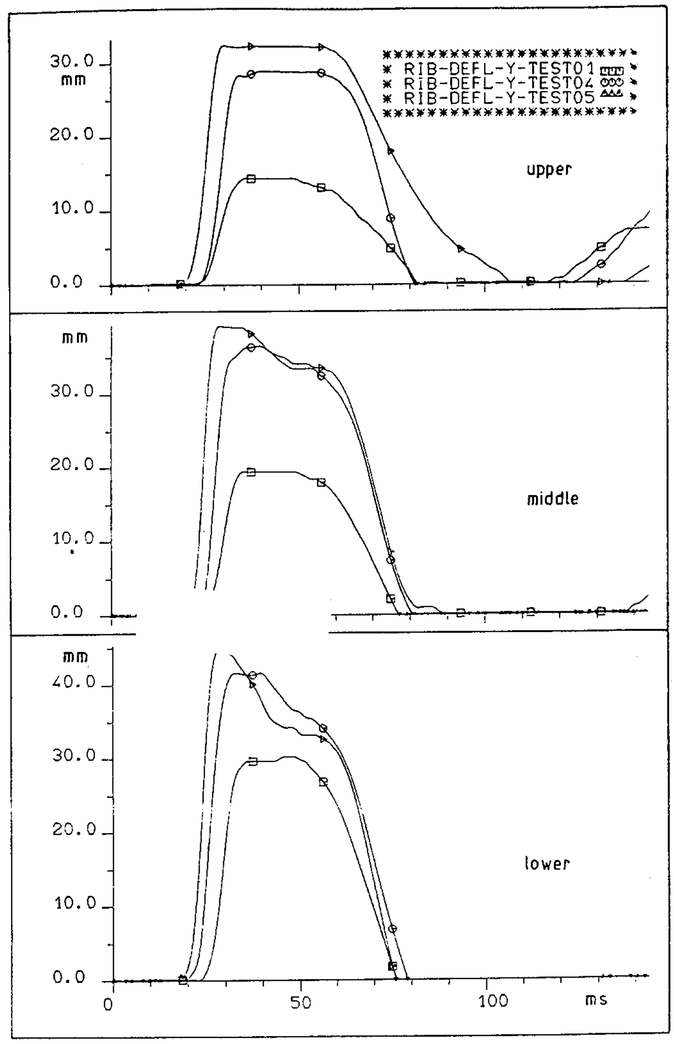

Figure 12. Rib deformations

identical for the tests conducted at 50 and 55 km/h (considerably lower for the 45 km/h tests), while the pelvis load for the test at 45 km/h is almost the same as that at 50 km/h (considerably higher for the 55 km/h test). The vehicle deformations show in the "1. level" almost identical values for 45 and 50 km/h.

The table simply serves to clarify the statements made above. The following graphs of the measurement values for the EUROSID dummy show the progressions of the curves, which begin earlier as velocity increases. For reasons connected with the computer, it was not possible to plot the mean value of the 50 km/h tests (2,3,4). One test (ESID 4) was therefore selected to represent the 50 km/h tests.

A fewer further remarks should be made here about the sensitivity of the thorax, which is of special importance. Figure 15 shows the rib deflection for the various velocities excluding test 2. Considering that in this test a freak value for the rib deflection was produced.

An impact velocity-related progression emerges after the mean value has been determined. The values with the 54 km/h "crabbed constellation" are in each case lower than those with the 50 km/h and 90 degree constellations.

The rib deflection values for the crabbed tests and the rib deflection values for the 90° tests up to 50

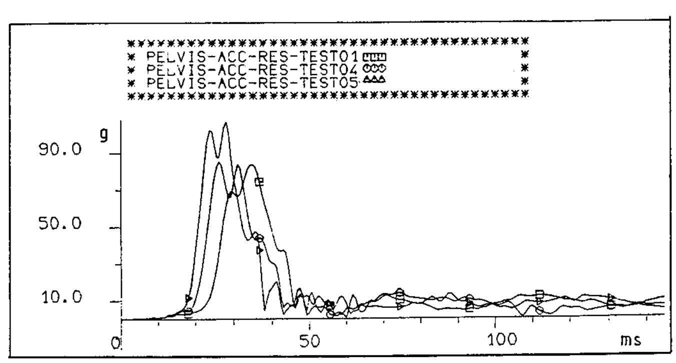

Figure 13. Pelvis accelerations

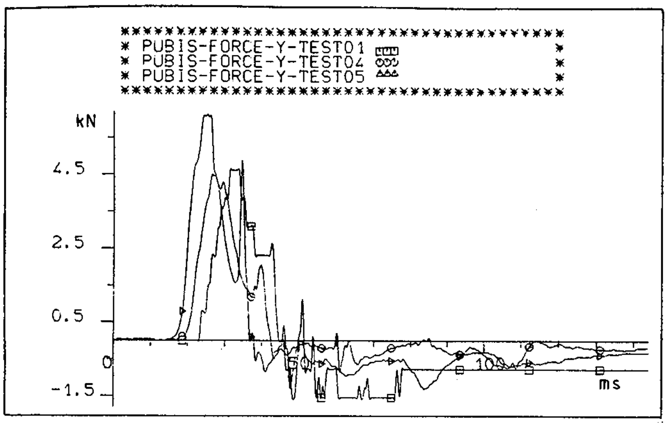

Figure 14. Pubis symphysis forces

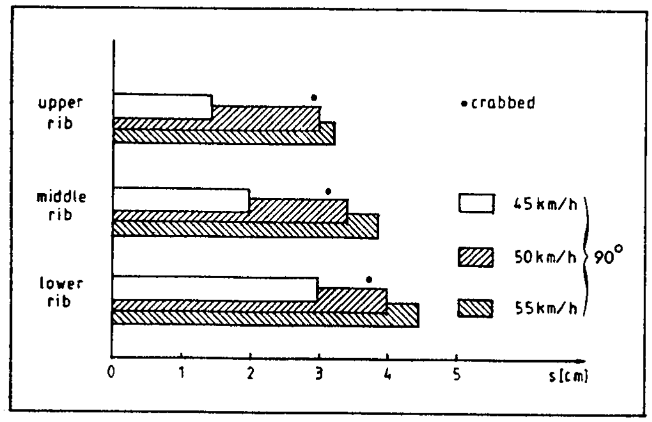

Figure 15. Maximum values for the individual rib deflection (without test 2)

km/h remain (in part considerably) lower than the proposed tolerance value of /40-45 mm/. Additionally, in all tests of this test series the pelvis loads (ilium and pubis symphysis) were much lower than the proposed /10 kN/.

Comparison of the Dummy Results for Different Test Constellations

The previous section has already dealt with rib deflection, which is approximately 5-15% lower for

Table 7. Comparison of the EUROSID measurement values for different test constellations.

			I 3 ms MEAN VALUES ESID 6, 7 crabbed, 54 km/h	II 3 ms MEAN VALUES ESID 2, 3, 4 90°, 50 km/h	I as % of II $D = \frac{100}{II} \cdot I$
HEAD	ACCEL. /g/	Y	20.0	23.0	87%
		Z	31.8	25.1	127%
		RES	34.7	39.9	87%
		HIC	160	265	60%
CHEST	ACCEL. /g/	Y	50.3	67.9	74%
		RES	51.2	69.3	74%
		SI	266.5	413	64%
		RIB U	70.9	94.2	75%
		RIB M	107.3	139.9	77%
		RIB L	97.1	108.7	89%
	DEFL. /mm/	RIB U	29.1	32.5	90%
		RIB M	31.3	36.5	86%
		RIB L	37.4	38.5	97%
PELVIS	ACCEL. /g/	Y	67.0	77.6	86%
		RES	70.3	79.9	88%
	FORCE /kN/	PUB.S.	3.6	4.7	77%
		IL.L.	0.7	1.0	70%
		IL.R.	0.2	0.7	29%

the "crabbed constellation" than for the 90° constellation. The best overall view of the results is obtained by listing the 3 ms mean values from the tests simultaneously, as shown in Table 8. This table also gives the percentage of the "crabbed value" over the "90° values" (column 3). With the exception of the head acceleration in the z direction (which is in any case unimportant as no head impact occurs), the loads for an occupant are approximately 10-30% lower in all regions of the body for the "crabbed constellation". One reason for this appears to be the point of impact of the barrier on the vehicle (i.e. whether or not the A pillar is deformed). The fact must be considered in the efforts to harmonize test constellations. Detailed discussion on the matter of different barriers, barrier faces and test constellation can be found in /3/.

Dummy Durability and Handling

Since there are no "crumple zones" on the side of the vehicle, a side impact dummy must be able to withstand strong impacts. If possible, no dummy components should be damaged during a crash. Although test methods provide for a velocity of approximately 50 km/h for the striking vehicle (barrier), higher velocities are also used for research purposes. In the tests conducted at the BASt, no serious damage occurred at velocities up to 50 km/h.

Deformation to the ribs was satisfactory and, in most cases, the shoulder was—as planned—twisted towards the front after the tests; however, on one occasion, the arm was torn out of the shoulder thread (see section on dummy kinematics). In some tests the spring guiding mechanism of one or two ribs locked, but could easily be loosened by a screwdriver. The flesh of the pelvis tore during the calibration procedure.

Easy handling, i.e. the simple, uncomplicated use of the dummy, is also important for the dummy to be acceptable in the test procedure. No difficulties were encountered here, either. However, a few slight improvements with regard to dummy handling and durability were proposed and incorporated in the EUROSIDs of the next batch.

Summary of the Test Results

The suitability of the European side impact dummy EUROSID was tested in 7 full-scale vehicle tests with different test constellations and velocities. The test vehicle was a VW Golf I (Rabbit), the colliding body a movable deformable barrier (EEVC IV face), which struck the test vehicle both at right angles (EEVC proposal, but with former ground clearance of 250 mm) and in the crabbed direction of travel (NHTSA proposal, but with the CCMC proposed barrier mass of 1100 kg).

The following can be concluded from the results:

1. The repeatability of the results is satisfactory.
2. Tests with velocities of up to 55 km/h may be conducted without causing any damage to the dummy.
3. The dummy is able to make a sufficient distinction between lateral collisions of varying degrees of severity.
4. Handling of EUROSID is no more difficult than that for other dummies.
5. In the tests with the crabbed test condition, all dummy-related measuring values were 10-30% lower than for the 90°, 50 km/h condition.
6. In both test conditions, all dummy-related measuring values were lower than the proposed preliminary EUROSID tolerance levels.

The last statement must be discussed further in the EEVC ad hoc Group on "Side Impact Dummy Development".

For us, the European Side Impact Dummy EUROSID appears suitable for use in procedures for testing the lateral protection of vehicles. The European Commission is in favor of introducing this dummy as standard dummy in the legislation for a European test procedure for side impact protection.

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Cadaver Response to Axial Impacts of the Femur

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Abstract

Axial impacts of the right and left femurs were performed on nine unembalmed human cadavers. A load cell was implanted in the right femur to measure the internal axial force. The impacts were performed at the same velocity within subjects and at various velocities between subjects. The forces and injuries produced in the right patella-femur-pelvis complex were similar to the injuries incurred in the left leg. The applied force of the impactor was also compared to the axial force measured in the right femur load cell. Results indicate that the ratio between applied and femoral force remained constant for the cadavers regardless of impact conditions.

Introduction

It has been recognized for many years that injuries to the patella-femur-pelvis complex of occupants involved in an automobile crash are numerous and serious. The results of a 1980 study of the National Crash Severity Study (NCSS) indicated that approximately 70-80% of the injuries to the lower extremities occur in frontal accidents. Furthermore, the study pointed out that the lower extremity injuries are frequently equal to or are the most severe injury for AIS 2-4 trauma when a lower extremity injury occurs (Reference 1).

Over the past twenty years, there has been considerable interest and research in the area of femur response, injury, and injury tolerance to axial impacts. A recent paper by Nyquist(Reference 2) presented a very thorough summary of the existing literature involved with femur injury as well as with leg and ankle injuries.

In 1965 Patrick, Kroell, and Mertz performed a series of ten impact tests with cadavers where the knees were impacted into a padded rigid surface(Reference 3). In this limited sample, it was not possible to determine whether the patella, femur, or pelvis was the weakest structure. Nonetheless, a judgment was made that 1400 lbs. was a reasonably conservative value for a maximum allowable axial load of the patella-femur-pelvis complex. This value became the accepted standard for the automotive industry and part of the FMVSS 208 Occupant Protection injury criteria.

In a follow-up study performed several years later the same investigators performed an additional four tests and found no fractures in the patella, femur, or pelvis with applied femur loads as high as 1950 lbs.(Reference 4). This resulted in the femur injury criteria of FMVSS 208 being raised to a level of 1700 lbs.

Further studies indicated that the patella-femur-pelvis complex can withstand larger loads for short durations(References 5,6,7,8). This finding led to an increase of the femur injury criterion to 2250 lbs., a level that coincided with the thinking of European investigators. This 2250 lbs. of axial femur load applied through the patella to a flexed leg remains part of FMVSS 208—Occupant Protection Standard today.

More recently the entire question of a valid injury criterion for fracture of the patella-femur-pelvis complex was reviewed by Viano and a new standard proposed(Reference 1). This criterion is based on analysis of all of the cadaver testing to date and upon finite element modeling of the femur(Reference 9).

The proposed guideline is:

$$F = 5200 - 160T \quad \text{for } T \leq 20 \text{ msec.}$$
$$F = 2000 \quad \text{for } T \geq 20 \text{ msec.}$$

where F = allowable femur load in lbs.
T = pulse duration

This equation recognizes that the femur responds in a static manner to loads longer than 20 msec in duration. For loading of less than this duration strain rate effects and structural dynamic response causes the allowable loads to increase dramatically. The same author in a review of all lower limb data(Reference 10) concluded that the maximum allowable load for impact durations longer than 20 msec was 2000 ± 450 lbs. For shorter periods of impact the allowable load could be much larger.

The following discussion presents femur loads and injuries from axial cadaver knee impacts with an implanted femur load cell in one limb and compares these to femur loads and injuries for axial knee impacts in the other limb.

Cadaver Testing

Axial impacts to the femurs of 15 unembalmed cadaver subjects were performed with a linear impactor at Calspan. Nine of these subjects had a load cell implanted in the right femur and are the topic of this paper. Of the remaining six subjects, one had instrumentation difficulties while the other five could not accept the load cell implant due to the small diameter of their femurs.

The impacts were performed on the right and left femurs with the subjects in a seated position with no back support. A summary of the subject and impactor characteristics is presented in Table 1. The weight of the impactor varied slightly in these tests but the difference is considered negligible. The velocity of the impactor ranged from 12 ft/sec to 39 ft/sec with similar impact conditions for the right and left femurs of each subject.

The selection criteria for the subjects used in this study included:

Age
Cause of Death
Bone Quality
Height
Weight
Injuries or Illness

A radiologist was asked to give a qualitative opinion of the skeleton using pre-test radiographs, thereby screening for osteoporotic subjects and identifying any existing conditions that might compromise the data obtained.

Right Knee/Femur Impact

All nine impacts were performed with the femur axially loaded at the distal end (patella) with the femur at 90° to the body and the leg drawn slightly back to avoid tibial contact during impact. Figure 1 shows a schematic of the test set-up with the impactor aligned axially to the subjects' femur using the patella and greater trochanter as landmarks. A triaxial accelerometer package was secured to the distal end of the femur prior to impact. The right femur was used for insertion of the load cell. The load cell ends have a diameter of 0.75 inches, therefore, the diameter of the femurs implanted were at least 1.00 inch to accommodate the load cell without fracturing.

Load Cell Implant

A 4-inch incision was made in the skin roughly 5 inches above the knee joint. The skin was reflected and the depression between the rectus femoris and vastus lateralis was exposed. The rectus femoris was reflected medially and the vasti muscles were separated between the vastus intermedius and vastus lateralis. The femur was subsequently exposed. From a point approximately 5 inches proximal to the femoral condyles a 2-3/4 inch portion of the shaft of the femur was cut and removed. The proximal and distal portions of the femur were then drilled, if required, to enlarge the diameter of the marrow cavity to 3/4 inch. Both proximal and distal cavities were packed with an acrylic resin, polymethyl-methacralate, and the load cell inserted. Hoseclamps were secured

Table 1. Impact summary.

Subject Characteristics					Impactor Characteristics		
Calman	Sex	Age	Ht. (in.)	Wt. (lbs.)	Impactor Weight (lbs.)	Right Femur Velocity (ft/sec)	Left Femur Velocity (ft/sec)
25	Male	55	72.0	188	55.8	12.8	13.2
29	Male	71	67.0	151	55.8	12.3	11.6
30	Male	66	69.0	185	55.8	21.5	21.4
31	Male	57	67.6	161	55.8	22.2	21.3
34	Male	55	71.0	141	51.5	20.7	20.7
35	Male	69	69.0	150	51.5	28.0	27.0
36	Male	61	69.0	160	53.7	26.7	31.2
41	Male	62	72.0	180	55.8	37.9	35.8
42	Male	60	68.0	190	55.8	38.6	39.3

around the ends of the femur at the load cell insertion location to preclude fractures at these points. The muscles and skin were then sutured. Care was taken in this procedure to avoid damaging the femoral vessels and their branches. Figure 2 presents a radiograph showing a typical implanted load cell.

Accelerometer Implant

A 2 inch length of skin overlying the distal end of the femur was incised, the underlying tissue reflected and a triaxial accelerometer package was fastened to the femur 4-1/4 inches from the most distal point of the patella (measured with the femur and lower leg placed at a 90° angle). The package was fastened with two 6 x 1/2″ mounting screws to each leg immediately prior to the knee impact. The muscle and skin were then sutured. The triaxial accelerometer package was secured at the time of the load cell insertion in the right femur.

Left Knee/Femur Impact

The left knee/femur impact was performed in the same manner as the right extremity. A load cell was not implanted in this femur so that a direct comparison between legs could be performed to identify the effect of the load cell on injuries.

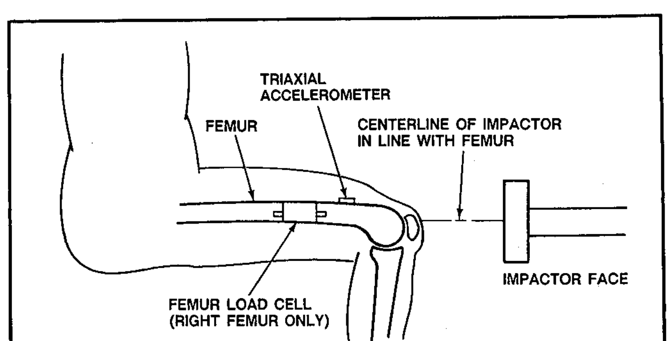

Figure 1. Schematic of implanted load cell

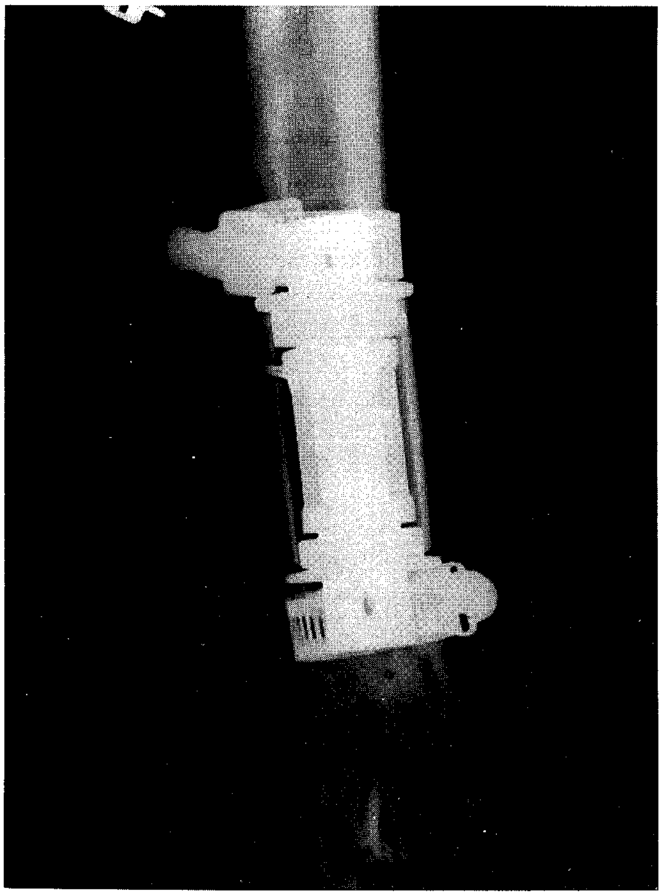

Figure 2. Implanted load cell in right femur

Cadaver Test Results

Table 2 presents a summary of the test conditions and injuries sustained in the right and left femur impacts, respectively. It is evident from this table that the fracture injuries in the right leg with the load cell implanted are similar to the fracture injuries in the left leg (with no load cell implanted) for each subject with only two exceptions. Calman 25 showed linear fractures at the site of the load cell insertions into the femur of the right leg and a fracture of the acetabulum in the left hip; Calman 34 displayed a fracture of the patella in the right leg and a intercondylar Y fracture in the left leg.

A post test examination of subject 25 indicated that prostate cancer had spread to the pelvis and most likely weakened the bone. This was the only hip fracture among these eighteen impact tests of nine subjects. The linear fractures at the load cell insertion site of subject 25 were judged to have been caused by the load cell and not by the impact alone. Calman 34 had comparable injuries in the right and left legs although they were not identical.

The fact that the injuries sustained in the right and left legs were essentially similar indicates that the insertion of the load cell did not have any substantial effect on the response of the right leg to impact.

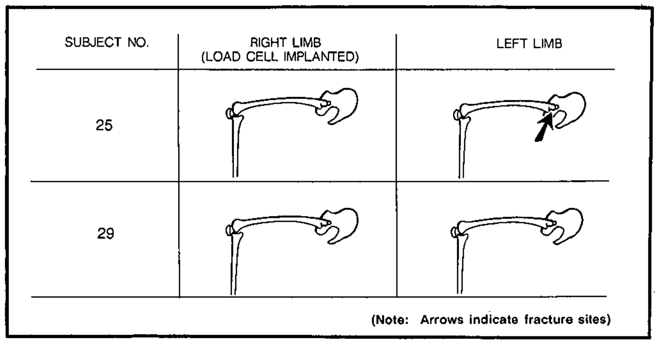

Figure 3. Comparison of right and left lower limb fractures—mild impacts

Mild Exposures

The nine subjects were impacted at velocities ranging from 12 ft/sec to 39 ft/sec in order to examine non-injurious, marginal, and highly injurious exposures. These have been labeled mild exposure, moderate exposure and severe exposure, respectively. Figures 3, 4, and 5 schematically summarize the fracture injuries for both lower limbs of each subject. Soft tissue injuries often accompanied these fractures but are not shown. In general, the 12 ft/sec mild exposure impacts (Figure 3), resulted in no skeletal injury to the patella-femur-pelvis complex.

Moderate Exposure

Figure 4 summarizes the injuries sustained in a series of five impacts performed at velocities from

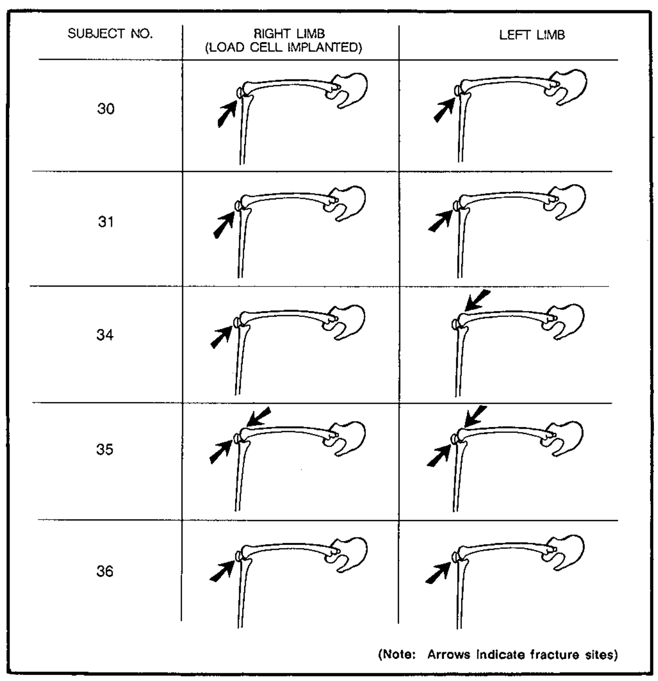

Figure 4. Comparison of right and left lower limb fractures—moderate impacts

Table 2. Femur impact results.

	Series 1 - Right Knee-Femur Impact				Series 2 - Left Knee-Femur Impact		
Calman	Velocity	Femur Load Cell	Injury	AIS	Velocity	Injury	AIS
25	12.8	1050	1) Linear fracture distal and proximal to load cell (b)	0	13.2	1) Fracture on lateral part of acetabulum (a)	0
29	12.3	1125	—	0	11.6	1) Anterior cruciate ligament torn on medial side 2) Medial meniscus displayed bucket handle tear (c)	3 —
30	21.5	2250	1) Patella fractures (6 pcs)	2	21.4	1) Patella fracture (5 pcs)	2
31	22.2	2189	1) Patella fractures (multiple)	2	21.3	1) Patella fracture (Y)	2
34	20.7	1400	1) Patella fracture (undisplaced) 2) 3 linear fracture of femur cartilage (c) 3) Linear undisplaced cartilage fracture of superiolateral surface of acetabulum	2 - 2	20.7	1) Intercondylar Y fracture (medial and lateral condyles displaced from shaft of femur) 2) Displaced cartilage fracture of superiolateral part of acetabulum	3 3
35	28.0	2100	1) Patella fracture (multiple) 2) Medial femoral condyle chipped off (c) 3) Horizontal fracture of tibial plateau between plateau and tibia 4) Cartilage fractures on superior part of acetabulum	2 — 2	27.0	1) Patella fractures (multiple) 2) Lateral femoral condyle cartilage fracture (c)	2 —
36	26.7	2070	1) Patella fractures (multiple) 2) Cartilage abrasions on lateral femoral condyle (c) 3) Femur fracture proximal to load cell (b)	2 - -	31.2	1) Patella fractures (multiple) 2) Femoral condyle cartilage abrasion (c) 3) Medial tibial plateau cartilage chip (c)	2 - -
41	37.9	2450	1) Patella fracture 2) Intercondylar fracture 3) Femur fracture distal to load cell.	2 3 3	35.8	1) Patella fracture (stellate) 2) Joint capsule of fibula had undisplaced fracture 3) Vastus lateralis and iliotibial band lacerated 4) Linear fracture of tibial plateau cartilage 5) Undisplaced intercondylar fracture 6) Spiral fracture at distal femur	2 2 1 2 3 3
42	38.6	2900	1) Patella fracture 2) Lateral condyle fracture 3) Complete fracture femur superior to epicondyles 4) Dislocated fibula 5) Slight laceration of biceps femoris and semimembranosus	2 3 3 2 1	39.3	1) Patella fracture (stellate) 2) Semimembranosus and biceps femoris lacerated 3) Complete fracture of distal femur 4) Complete intercondylar fracture 5) Lateral part of head and tibial plateau fractured	2 1 3 3 2

(a)Injury was due to weakening of pelvis through prostate cancer.
(b)Injury was due to installation of load cell.
(c)Injury was coded with other injuries in this joint.

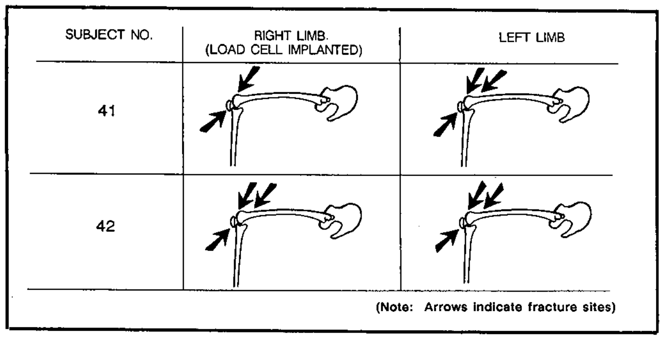

Figure 5. Comparison of right and left lower limb fractures—severe impacts

20.7 fps to 31.2 fps. In nine out of ten exposures the patella was fractured. Subject 34 sustained a patellar fracture of the right limb and an intercondylar Y fracture of the femur of the left limb. It is not clear why the femur fracture occurred rather than a patellar fracture for this case. Subject 34 also sustained cartilage fractures of the surface of the acetabulum on both sides. Subjects 35 and 36 were impacted at slightly higher velocities and sustained various cartilage fractures in both limbs in addition to patellar fractures.

For this range of impact velocities, a patellar fracture generally occurred. At the higher velocities, approximately 25 fps, these patellar fractures were frequently accompanied by a cartilage fracture. In all of these subjects the injuries to the right and left limbs are similar despite the load cell implanted in the right femur.

Severe Exposure

Figure 5 summarizes the injuries for the two subjects impacted at approximately 36 fps. Severe injuries were sustained including patellar and femoral fractures. Three out of four exposures resulted in intercondylar Y fractures of the femoral condyles while the fourth sustained a lateral condylar fracture. In three out of four cases the femoral shaft was also fractured immediately above the condyles.

As with the previous subjects the right and left sides sustained similar injuries.

Table 3. Cadaver femur impact summary.

Calman	Load Cell Force (lbs)[1]	Load Cell Duration (ms)[2]	Right Femur Applied Force (lbs)[3]	Right Femur Duration (ms)[2]	Left Femur Applied Force (lbs)[3]	Left Femur Duration (ms)[2]
25	1070	12.0	2280	12.0	2060	16.5
29	1120	18.4	2110	8.4	1630	12.0
30	2250	7.0	4300	7.5	4100	10.0
31	2200	12.0	3120	10.4	2900	10.0
34	1410	8.4	2450	9.6	2320	11.4
35	2110	10.0	2600	8.8	3450	10.0
36	2090	7.5	3870	7.5	3920	11.0
41	2480	7.5	4580	7.0	6550	8.8
42	2930	7.5	5360	6.0	5470	7.5

(1) Filtered at 600 Hz (SAE J211)

(2) Measured at 20% of maximum force

(3) Calculated from accelerometer filtered at 180 Hz (SAE J211)

Comparison of Results with Viano Tolerance Curve

The peak force measured for these nine subjects was obtained from the product of the peak acceleration filtered at SAE J211 class 180 Hz and the impactor mass. The duration of each force pulse was measured at 20% of the peak force. The peak force and pulse duration are listed in Table 3.

Figure 6 shows the peak applied force for these nine subjects plotted against the duration of the force pulse. The fracture tolerance curve proposed by Viano and based on all previous cadaver femur impacts is also shown. The Viano tolerance curve recognizes that for short-duration impacts the patella-femur-pelvis complex is able to withstand much higher force than for long-duration impacts. At durations of 20 msec or longer the femur is felt to respond statically with a failure load of approximately 2000 lbs. For durations less than 20 msec the failure load increases rapidly.

It can be seen that the mild exposure impacts lie well below the tolerance curve (subjects 25 and 29). The moderate exposures are clustered around the tolerance line with approximately one-half above the line and one-half below the line. The severe exposures all plot well above the line. The fracture versus force-duration results from these nine subjects do not differ significantly from results reported previously

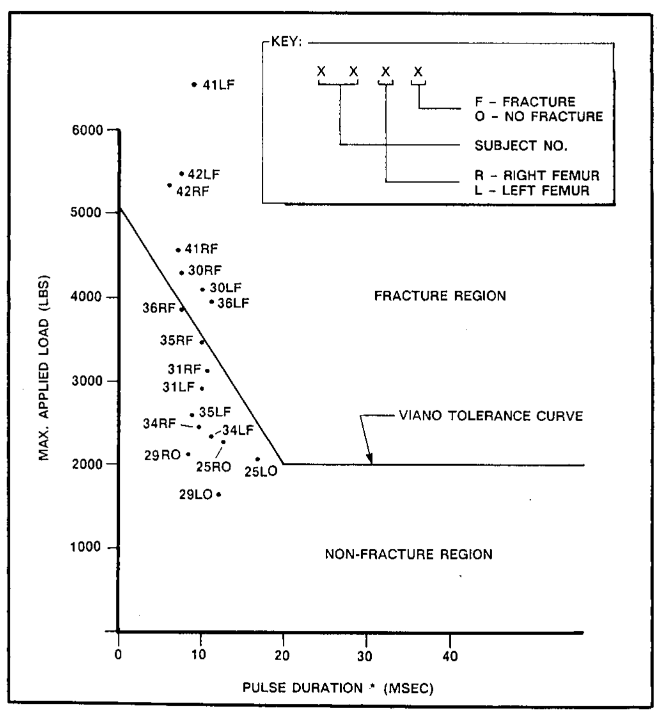

Figure 6

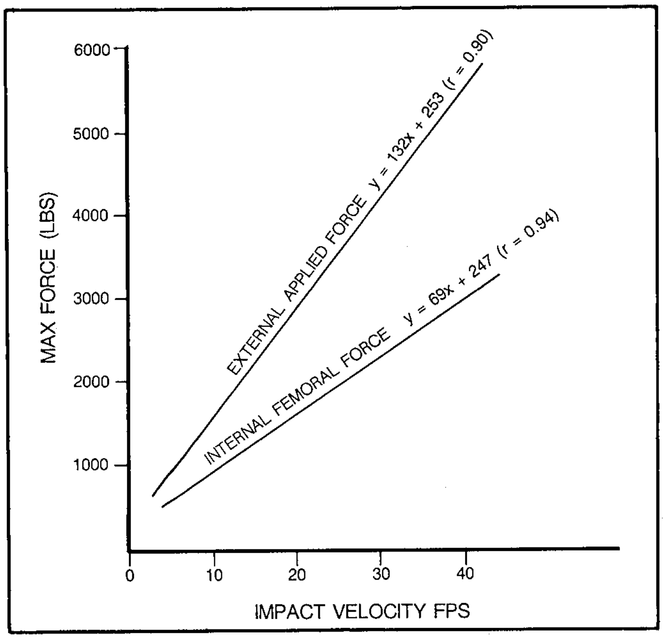

Figure 7. Comparison of cadaver applied and femonal force

and summarized by Viano. All of the non-fracture cases lie below the tolerance line and most of the fracture cases lie on or above the tolerance line.

Comparison of Cadaver Applied to Femoral Force

Figure 7 shows the external applied force and the internal femoral force plotted against the impact velocity. Linear regression lines were fit to each set of data; the equation and the correlation coefficients are shown in the figure. Clearly, the applied force is much larger than the internally measured femoral force. This is to be expected because a considerable mass is located between the internal load cell and the impact point (see Appendix A). The ratio of the slopes of the internal force curve to the external force curve is 0.53. This indicates that 53% of the effective struck mass is located proximal to the load cell and 47% of the effective mass is distal to the load cell. It is unlikely that almost half of the mass of the thigh was distal to the load cell in these tests. It would thus appear that a portion of the leg is included in the effective struck mass of the thigh and contributes to the reduction of the internally measured femoral force.

The tolerance curve proposed by Viano relates applied force to fracture injury. If the internal femoral forced were to be used in a similar type of tolerance curve the fracture injury force levels would have to be reduced by a factor of approximately 0.53.

Conclusions

The following conclusions can be made from this study of cadaver knee impact.

- A load cell can be implanted in a cadaver femur to measure femoral axial load.
- The implanted femoral load cell does not affect the injury pattern or severity sustained in a knee impact.
- The internally measured femoral force is always less than the external applied force.
- The ratio of the internal femoral force to the external applied force is approximately 0.53.

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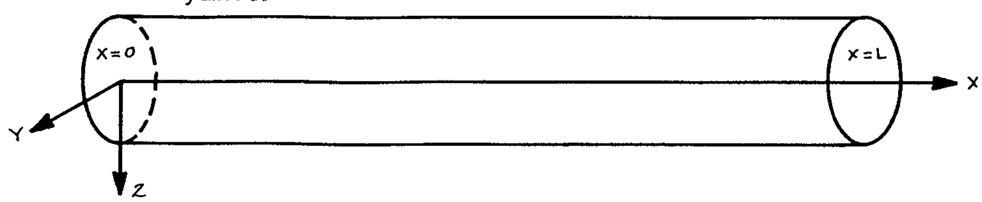
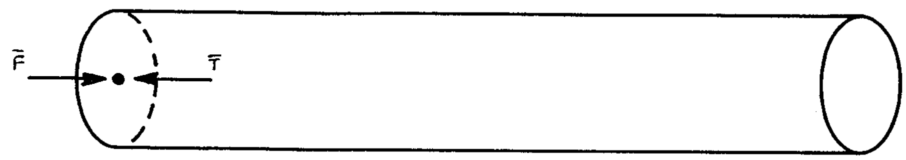
Appendix A
Variation of Stress Along the Length of a Cylinder

Consider a cylinder



The cylinder is impacted at the left end, X = O, and accelerates in the positive x-direction. All forces and motions are colinear. A free body diagram of the cylinder is,



where $\bar{F}$ is the resultant applied force vector
$\bar{T}$ is the internal stress field vector

From equilibrium

$$\bar{F} = \int_A \bar{T} dA = \int_V \frac{\partial(\rho \bar{v})}{\partial t} dV$$

where A is the area of the cylinder
ρ is the density function
$\bar{v}$ is the velocity at each point of the cylinder

Since force is in the x-direction only, Cauchy's theorem gives

$$\int_A \bar{T} dA = \int_A \tau_x \bar{n}_x \, dA = \int_V \frac{\partial(\rho v)}{\partial t} dV$$

where τ_x is the normal stress in the x-direction
$\bar{n}_x$ is the outward normal in the x-direction on the surface at $x = 0$

Applying Gauss' theorem,

$$\int_V \frac{\partial \bar{\tau}_x}{\partial x} dV = \int_V \frac{\partial(\rho \bar{v})}{\partial t} dV$$

For these integrals to be equal the integrands must be equal. For a rigid homogeneous material the right hand integrand is a constant, therefore the stress varies linearly in the x-direction.

Integrating over the volume of a cylinder of constant cross section, applying the boundary condition that $\tau_x = 0$ at $x = L$ and writing the scalar equation gives

$$\tau_x A = \rho A \frac{dv_x}{dt} (x - L)$$

The equation holds at any point in the body. It can be seen that the stress at any point in the cylinder, and therefore the force if the body were cut at that point, is inversly proportional to the distance from the impacted end.

at $x = 0$,

$F_x = \tau_x A = -\rho A L \frac{dv_x}{dt}$ (negative sign indicates compression)

at $x = L$,

$F_x = \tau_x A = 0$

Chest Compression Response of Hybrid III With Combined Restraint Systems

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Abstract

Studies with Hybrid II dummies have demonstrated that the chest acceleration is a poor indicator of chest injury potential and that a realistic evaluation of the effectiveness of restraint systems should be based on measurement of the chest compression by means of strain gauges on a dummy's ribs. The projected introduction of the Hybrid III dummy into crash testing along with the requirement to measure sternum deflection for injury assessment has brought about the need to evaluate deflection measuring technology used on the Hybrid III. Tests revealed that different load distributions, i.e. due to a diagonal shoulder belt or an airbag, did not produce the expected different chest deflection patterns. That is, the thorax experienced the same deflection pattern with both restraint systems. This appears to be the result of an excessively stiff sternum assembly. To be sure that the highly protective effect of airbags, which is evident from real world accident analysis, be reflected in laboratory tests with dummies, this study suggests that the thorax of the Hybrid III dummy must be improved with respect to its sensitivity to injury producing local forces.

Introduction

Recent tests with cadavers and animals confirmed that the maximum chest acceleration alone was a poor indicator of chest injury potential and that the chest compression and other parameters derived from compression are superior predictors(1-7). Furthermore, several studies with Hybrid II dummies have demonstrated that a realistic evaluation of the effectiveness of restraint systems such as airbags or combined systems should be based on individual measurements of each rib deflection as accomplished by a system of strain gauges(8,9).

The projected introduction of the Hybrid III dummy into crash testing along with the requirement to measure sternum deflection for injury assessment (FMVSS 208) has brought about the need to understand how well this dummy provides equivalent responses. Particularly, it is essential to know how well the Hybrid III dummy's centrally located interior deflection gauge senses chest compression under various loading conditions.

Static Tests

The primary interest of this portion of the study was to determine if the local forces of a shoulder belt as well as the widely distributed load of an airbag are realistically reflected in the deflections of the dummy's ribcage. Therefore, static tests were conducted to compare the deflection characteristics of the ribcage structures of both, Hybrid II and Hybrid III dummy.

Test Method

Each ribcage was mounted in an universal test machine as shown in Fig. 1 and loaded with either a shoulder belt or driver airbag (filled with foam to maintain inflated condition). The belt was located in a geometric position similar to its position on a passenger car occupant. The compression was produced by pressing the ribcage into the restrain system until a force level of 2 kN was reached at the dummy's spine. The tests were conducted without the soft tissue coverings, both with and without the clavicle. Deflection measurements were taken between the spine and each single rib (70 mm to the left (d_1) and right (d_2) side of the center line). In addition to the 12 direct deflection measurements, each rib deflection (center of sternum relative to spine) was calculated from the strains (S_1, S_2) measured on gauges located on the ribs on both sides close to the spine.

Results and Data Analysis

The results of the tests are summarized in Fig. 2-4, indicating significant differences between Hybrid II and Hybrid III:

- The ribcage of the Hybrid III is much less stiff under frontal loads—its compression is more than three times higher.
- In comparison to the Hybrid II, where the path of the shoulder belt is distinctively reflected in the deflection pattern of the ribs, the ribcage of the Hybrid III again has greater overall deflection but the deflections are nearly as uniform as with the airbag.
- Different load distributions did not produce the expected different chest deflection patterns, that is, the Hybrid III's thorax experienced nearly the same deflection pattern with both restraint systems. It appears, because of the extremely stiff design of the sternum, that local forces are not sufficiently reflected in the rib deflections.

- Lateral deformations of the Hybrid III's chest due to the nonsymmetric load of a shoulder belt, although similar in magnitude to those of the Hybrid II, are small in comparison to the sagittal components.
- It appears, because of the dominating sagittal deflections, that lateral loadings are not sufficiently reflected in the resultant deflections of the Hybrid III, resulting from the sagittal and lateral components.
- As a consequence, the average value of all the six resultant rib ring deflections of the Hybrid III's ribcage is only about 30% smaller with airbag than with shoulder belt—in comparison to a reduction of about 75% with the Hybrid II.
- Particularly interesting is the distinctive belt load concentration on the Hybrid III dummy's clavicle, resulting in a significant reduction of the chest compression (Fig. 3). In contrast to this, the deflection of the Hybrid's II's ribcage was the same with or without clavicle.

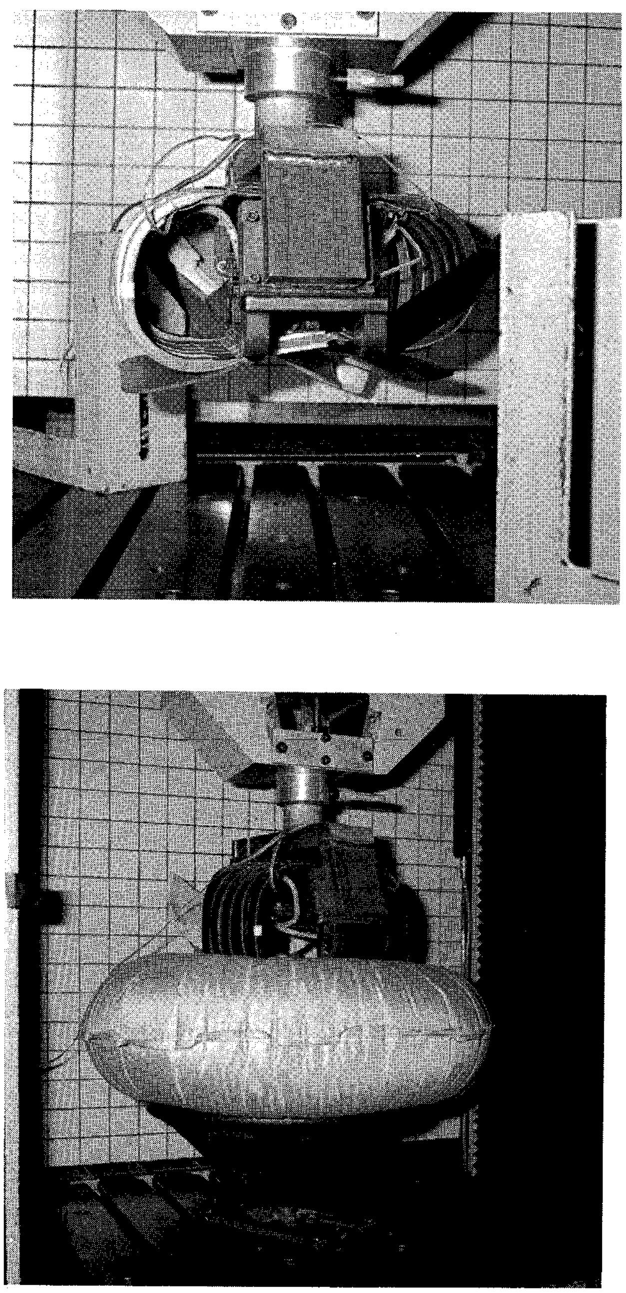
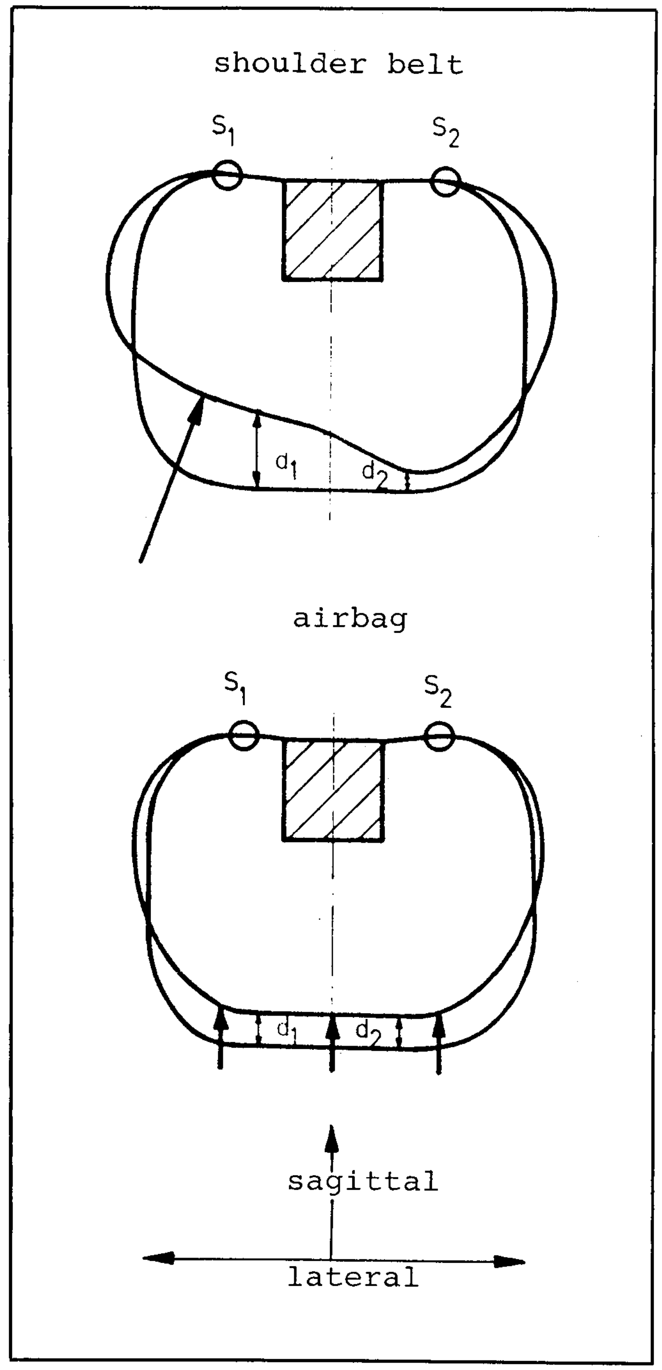


Figure 1—Ribcage of the dummy in the universal test machine



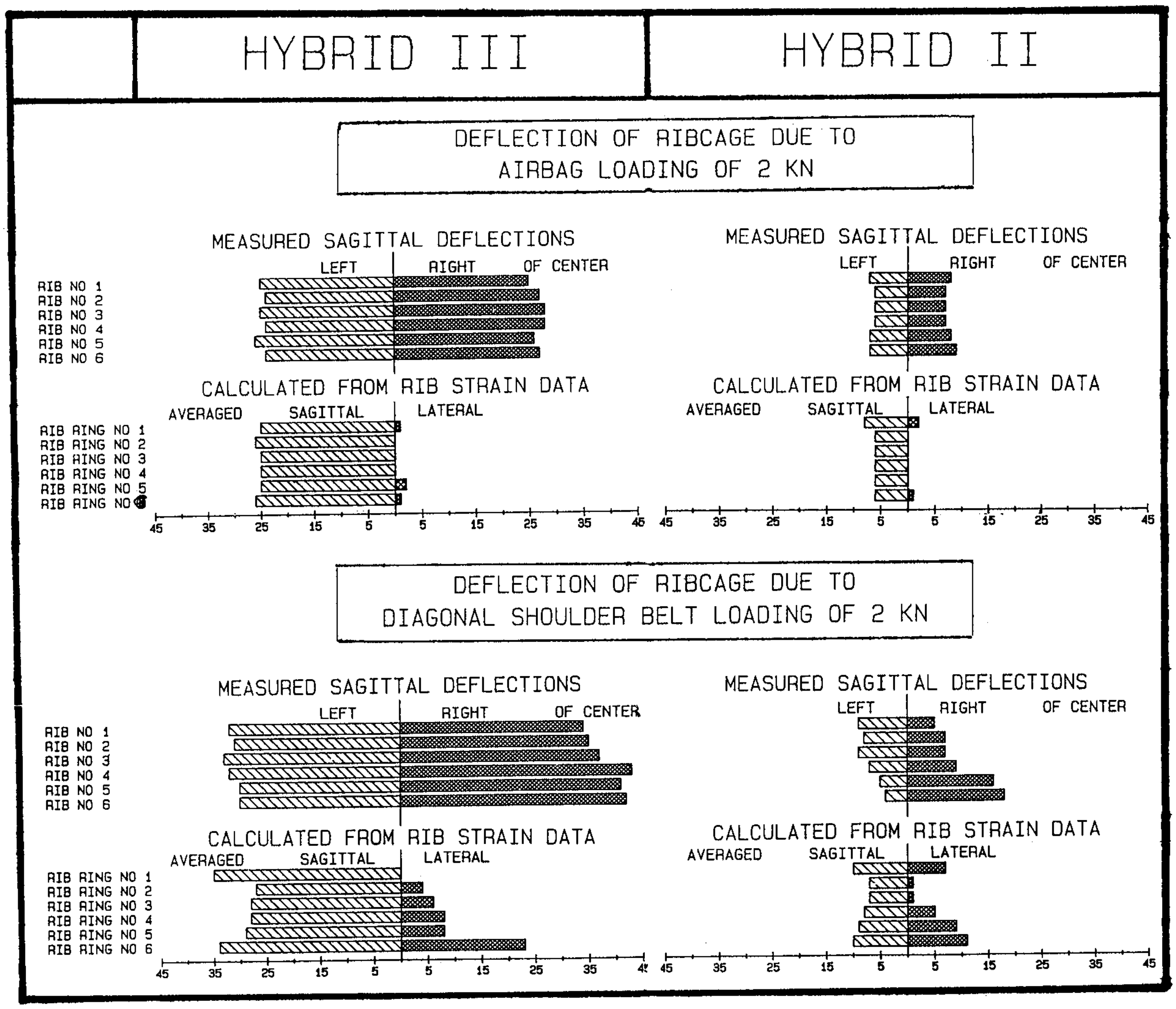


Figure 2—Deflection of ribcage due to different static loadings

Sled Tests

In order to establish how well the Hybrid III dummy's ribcage senses chest loadings under various dynamical loading conditions, 30 mph sled tests with different types of restraint systems were conducted, namely

- three point safety belt,
- driver airbag with knee bolster,
- airbag combined with three point belt.

The test device was a Hybrid III dummy, which was, in addition to the sternum displacement sensor, equipped with 12 strain gauges symmetrically attached on the six ribs on both sides close to the spine(7). From this rib strain data, the sagittal, lateral and resultant deflections of each single rib ring as well as the resulting average values were calculated.

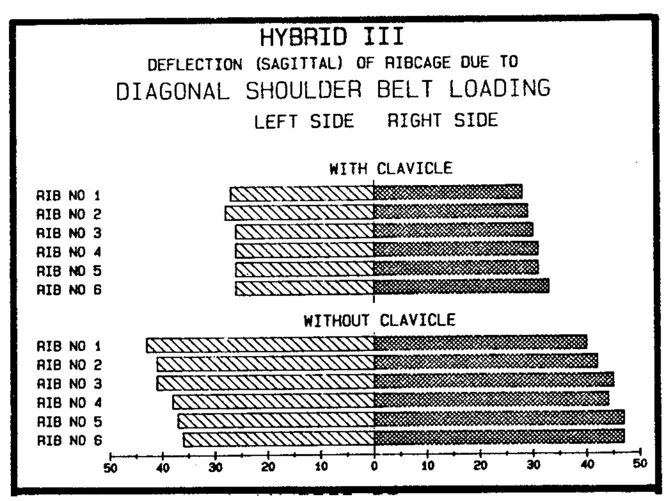


Figure 3—Deflection of the Hybrid III's ribcage due to diagonal shoulder bet loading—either with or without clavicle

Results and Data Analysis

A comparison of the chest deflection patterns between those observed on Hybrid III and Hybrid II, is given in Fig. 4 and 5. The data confirms the results of the static tests, namely:

- Sagittal deflections with the Hybrid III dummy are more than three times larger than with the Hybrid II; they are uniformly distributed over all the ribs with either of the selected restraint systems (higher magnitudes for the uppermost ribs);
- Lateral deflections are small with airbag and large with seat belts (with increasing tendency towards the lower part of the ribcage) with either of the dummies—however relatively small in magnitude in comparison to the Hybrid III's sagittal deflections.
- In contrast to the Hybrid II, the resultant deflections of the Hybrid III, because dominated by the sagittal components, do not reflect differences in either load distribution or direction. As a result, the Hybrid III's chest compression response is nearly the same with either of the restraint systems.
- Sternum displacement values, measured with the centrally mounted internal deflection gauge, are nearly the same as the averaged resultant rib deflections (Fig. 5).

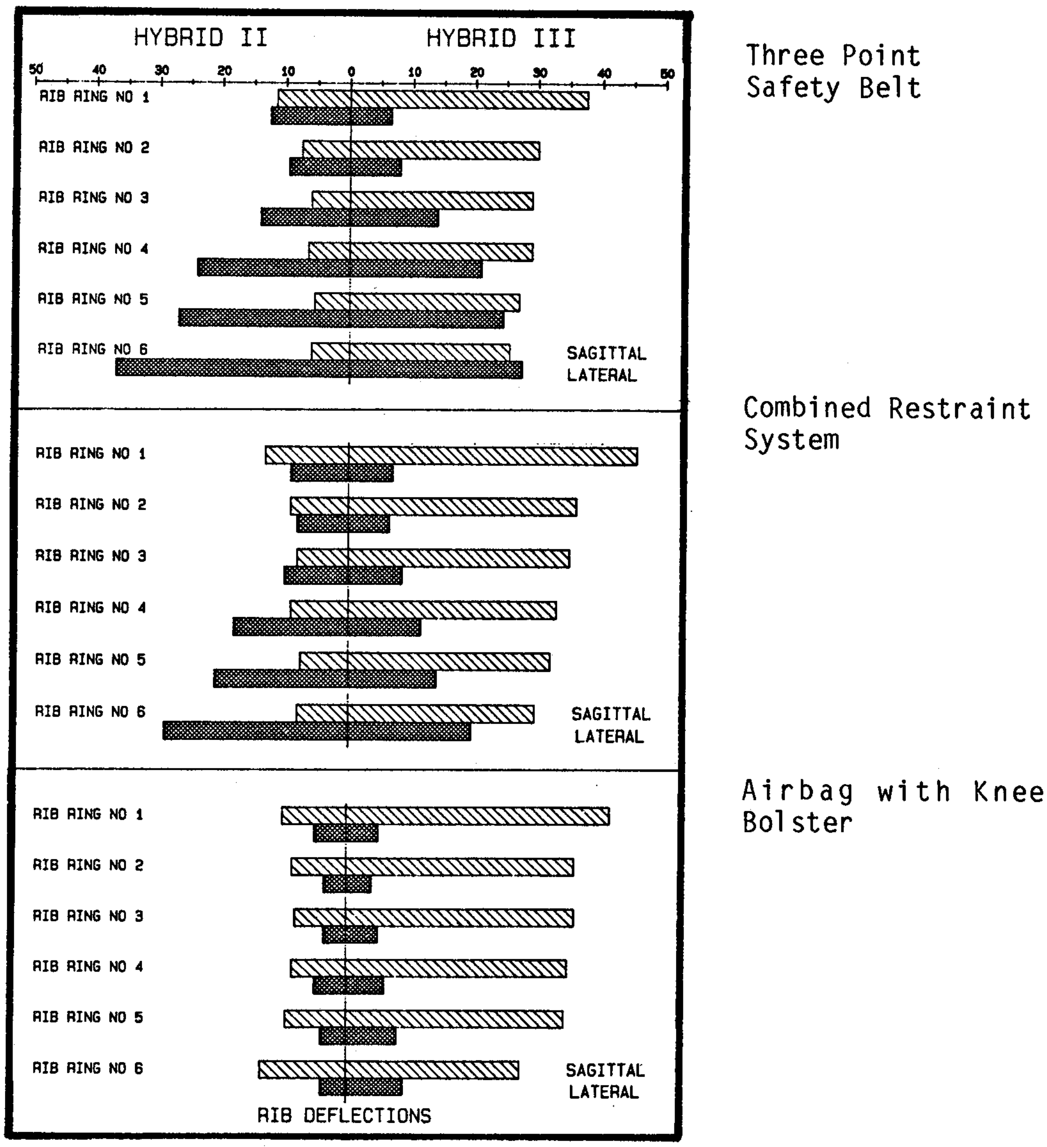


Figure 4—Sagittal and lateral deflections due to different restraint systems (30 mph sled tests—driver position)

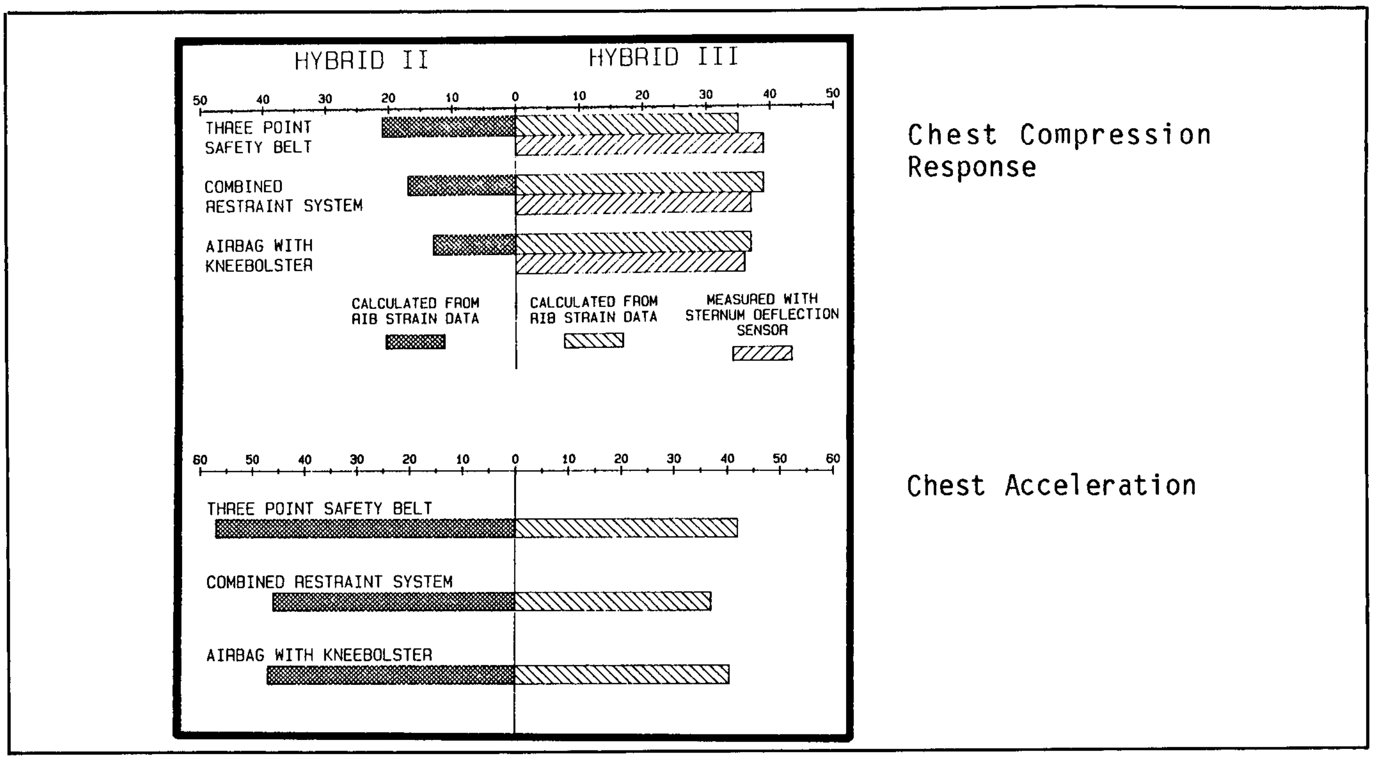


Figure 5—Chest loading due to different restraint systems

Summary and Conclusions

Obviously, there are significant differences between the Hybrid III and Hybrid II with respect to its chest compression responses:

- Evaluations of the chest injury risk on the base of rib strain data of Hybrid II are plausible and completely in line with our experience of real world accidents.
- In contrast to this, the Hybrid III's ribcage, although superior in biofidelity, does not sufficiently discriminate between the widely distributed load of an airbag and the highly concentrated load of a diagonal shoulder belt.
- Different load distributions, i.e. due to a diagonal shoulder belt or an airbag, did not produce the expected deflection patterns on the Hybrid III's ribcage. Particularly, it appears, because of the extremely stiff design of the sternum, that local deflection of rib is not sufficiently realized.
- In our opinion, the Hybrid III's chest compression response appears to be a poor indicator of chest injury risk, just as unsuitable as chest acceleration.
- It is open to discussion whether lateral deflections of the Hybrid II's ribcage, measured with the strain gauges, are unrealistically large. However, this study suggests that the Hybrid III's ribcage underestimates lateral loadings considerably. As a matter of fact, high local forces, i.e. due to shoulder belt loading, do cause multiple rib fractures (Fig. 6) as well as injuries to internal organs. This is particularly true for the lowermost ribs as well as for injuries of liver and spleen. It appears, because the lowermost ribs of the human ribcage are not linked to the sternum, that the biofidelity of a dummy's ribcage is not suitable with respect to this.

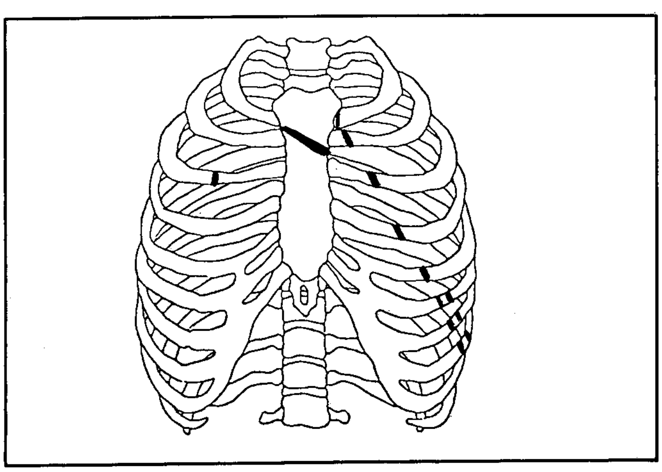


Figure 6—Rib fractures due to diagonal shoulder belt loading

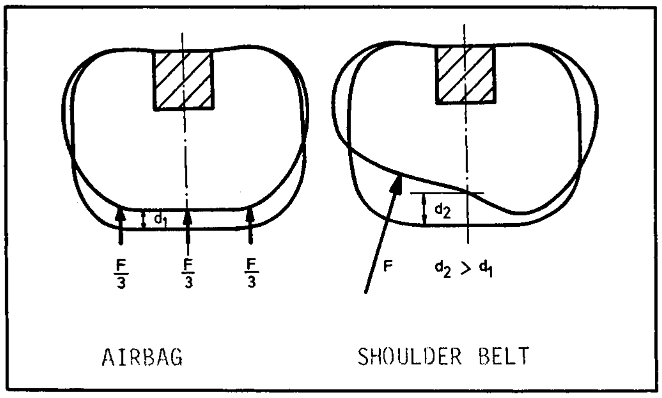


Figure 7—Expected deflection due to different load distributions

As a consequence, to be sure that the highly protective effect of airbags, which is evident from real world accidents, be reflected in the chest compression response, this study suggests that the thorax of the Hybrid III dummy must be improved with respect to its sensitivity to injury producing local forces (Fig. 7).

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Development of a Two-Dimensional Sensor Determining Abdominal Loading on TNO-Dummies

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Abstract

A two-dimensional fluid-filled sensor is described in determining abdominal loading separately in horizontal and vertical direction on TNO-child-dummies.

It is shown that an intrusion-capable surface-pressure onto the abdomen will result in a corresponding load within the abdomen. This can be measured by a hydraulic tube package positioned on the lumbar spine (x-direction) resp. underneath the lower thorax (z-direction) and equipped with piezo-electrical pressure sensors. A series of tests were executed to prove that the proposed device clearly discriminates the degree of abdominal loading among various child safety systems and maintains the reproducibility required with no influence on other test criteria.

State of Art

The European Child Safety Standard ECE-Regulation No. 44[1] specifies a rectangular block of modelling clay placed upon the lumbar spine of the TNO-dummies to detect possible abdominal penetration. However, after applying this method over a number of years experience shows that calibration and positioning of the dummy on the test-seat can deform the clay without occuring of abdominal penetration. As a consequence it is difficult to assess whether penetration has happened during the dynamic test. Therefore, Roberts/Lowne proposed an alternative device to unequivocally detect abdominal penetration on TNO-child-dummies[2]. They designed a mechanical event detector based on bubble film packaging material, which is clamped onto a flexible backing plate supported by the lumbar spine. But again it was found that the bubble film detector sometimes also failed to detect penetration in comparison to the high-speed crash-film. Due to possible malfunction it can be stated that both the clay- and bubble film-method additionally require the analysis of the crash-film with the latter overruling both special methods in case of doubt.

Leung et al. developed a detecting system by using a transducer bolted to the dummy's pelvic crest at a position where abdominal injuries are likely to happen upon submarining[3]. It is not known whether this method has been successfully applied to TNO-child-dummies.

Within a proposal for a booster cushion standard Waters used the hip movement as a parameter for evaluation submarining[4]. The criteria was: 150 mm maximum forward movement of the hip together with a minimum chest movement in case of excess maximum hip movement.

Up to now abdominal penetration is no test criteria within the American Child Safety Standard FMVSS 213. Melvin/Weber considered it to be useful for the child restraint designer to have a device within the dummy abdomen that could clearly indicate the degree of penetration, dynamic pressure level and rate of penetration[5]. In 1986 the same authors proposed an abdominal intrusion sensor designed for the Part 572 3-year-old dummy[6].

This new concept using a fluid-filled tube-package within the dummy abdomen was capable to continuously measure intrusion initiated by a restraint system. Upon testing the device could discriminate the degree of abdominal loading among a variety of different restraint systems. The extremely promising results lead the Physicians of Automotive Safety requesting NHTSA to amend the American Child Restraint Standard FMVSS 213 to include maximum criteria for abdominal pressure[7].

Design Target

For this work the target was set to design a device, comparable to the system described in [6], for the TNO dummy family. The principal differences between the American and European dummies require a new design mainly due to the restricted space within the abdomen of the TNO dummies and the off-center position of the spine, which does not allow wounding loops of rubber tubing around the lumbar spine as done on the Part 572 dummy. As a further consequence the transducer-type cylinder as described in [6] could not be used, regardless that it was desired to avoid possible influences by friction and mass effects. It is reported that the spine in the Part 572 3-year-old dummy is less flexible and less realistic than the comparable P3 TNO dummy[8]. Furthermore the deformation of the spine is much greater for the P3 than for the Part 572. The sensor device to be proposed should not influence these positive elements of the TNO dummies.

Though it is not proposed to directly compare unknown biomechanical limits with measured sensor data nevertheless these data might establish valuable design limits. They could be used as an auxiliary base in a child safety standard such as ECE 44 comparable to chest deceleration measurements and—if approved—replacing the present method of abdominal penetration assessment.

The proposed system should be able to abandon the evaluation of crash films which is presently required in order to judge excessive abdominal penetration.

Requirements

At the initial design stage the following basic requirements were established for the given reasons.

Continuous Measurements

As described the state of art includes a number of systems detecting abdominal penetration on a Yes-No basis, such as modelling clay, bubble film, electrical switches or the evaluation of crash films. Different to one-dimensional functions the degree of abdominal penetration is a highly complex process which really cannot be reduced to a Yes-No assessment. Comparable to head and/or chest deceleration the penetration-time history is as important to be known than the peak loads. Continuously measured data can provide the basis for failure research and design optimization. Designers and quality control will welcome the possibility to know whether a restraining system stays within the safe tolerances of given limits or not.

Reproducibility

Good reproducibility is always an important precondition for a functional test system, even more if the test criteria is used within a standard for the purpose of approving or rejecting safety products. This also applies for in-house conformity-of-production test work.

Measurement in horizontal and vertical direction

The known abdominal penetration sensors for TNO dummies are designed for detecting horizontal loads (x-direction). However, abdominal penetration due to strong submarining may mainly result in vertical loads (z-direction). Therefore both x- and z-direction loads should be detected. It seems to be wise to separate both measurements since the absolute data measured may have different biomechanical relevance, analog the data for chest deceleration in ECE 44. If possible interference between both sensors should be avoided. And finally, the sensor for horizontal loads should be able also to pick up excessive lateral loads.

Good Discrimination

The sensor should prove the linear interdependence between actual load and measured data. Extreme data such as no-load or high-load should be clearly detected as no-reading resp. high-reading, both in x- and z-direction.

No Time lagging

For direct comparison with other test data, with the crash film and for failure research there should be no time lagging for continuously measuring abdominal penetration. Therefore the sensor must react instantaneously.

Simple Calibration

To reduce the expense on time and costs the calibration of the sensor must be simple. Thereafter the device should maintain a long data consistency which is important for a good data reproducibility. If possible, there should be linear calibration curves.

No Influence on other Test Criteria

The function of the sensor should not interfere with the dummy motion or with other test criteria such as head excursion and deceleration of chest and head.

No Substantial Changes to Dummy

The implantation of the sensor must be simple with no substantial changes to the dummy itself. The structure and durability of the dummy should not be deteriorated.

Basic Consideration and Tests

It seems to be obvious to measure abdominal penetration as a function of intrusion depth in mm. This, however, requires that the dummy abdomen is sufficiently comparable to the child abdomen regarding structure, deformation and other properties. The dummy abdomen consisting of a homogeneous foam insert cannot fulfill this basic requirement. Therefore a load measurement might be advantageous. Basically the surface pressure onto the abdomen should be measured but only where intrusion can occur, i.e. not at the iliac crest or the lower thorax, however within the area in between.

Such a surface pressure will result in loads within the abdomen, therefore it is proposed to use a sensor measuring the loads within the abdomen.

The correlation between surface pressure and abdominal loading is shown in fig. 1. A static load of 500N was applied onto the dummy abdomen and the reaction load within the abdomen was measured at the height of the lumbar spine:

- If the load was applied to the lower thorax (D), there was no abdominal reading,
- loads onto the iliac crest (C) resulted in noticeable readings only at the lower lumbar vertebrae(4,5),
- loads between iliac crest and lower thorax (A,B) resulted in a parabolic load distribution with peaks directly underneath the load application points.

Upon design of the sensor it was preferred to directly measure the abdominal loading thus avoiding an indirect measurement by means of a transducer cylinder. For this purpose preliminary tests were done with a hydraulic sensor to which a piezo-resistive pressure transducer was directly connected. The curves achieved were comparable to the corresponding results described in [6].

As a result it was noted that a hydraulic sensor positioned on the lumbar spine within the abdomen can prove intrusion-capable abdominal loading, if the sensor extends approximately between the first and fourth lumbar vertebra. (see fig. 2).

Non-intrusion-capable loads are not recorded. It can be concluded that the pressure distribution—mainly pressure peaks—in this area represent a measure for intrusion-capable abdominal loading.

In order to measure this loading a number of alternatives exists:

- a hydraulic tube package equipped with a piezo-electrical pressure sensor (fig. 2)
- direct pressure measurement via a piezo-sensitive wire (fig. 3)
- a piezo-sensitive foil (fig. 4)

Basic tests performed on the calibration unit as described later could prove that all 3 alternatives should have the same reaction upon applying a dynamic force and therefore represent equivalent solutions.

The hydraulic sensor similar to the system described in [6] has a simple and advantageous design. In order to prove that pressure peaks are correctly recorded a test as per fig. 5 was performed. The influence of load distribution onto the actual reading was studied. For this purpose the hydraulic sensor was subject to an equal load successively applied to 1 loop, 2 loops, etc. Ideally the recorded reading should be indirect proportional to the number of loops if non-expandable tubes are used (see fig. 5, curve A). With fully expandable tubes a peak load will be suppressed as shown in curve C.

The hydraulic sensor tubes finally used somewhat compromise both extreme versions, i.e. the pressure peak will be slightly mitigated (see curve B).

This possible disadvantage of a hydraulic sensor is reduced by the balancing effect of the abdominal foam insert, which will transfer a pressure peak at the

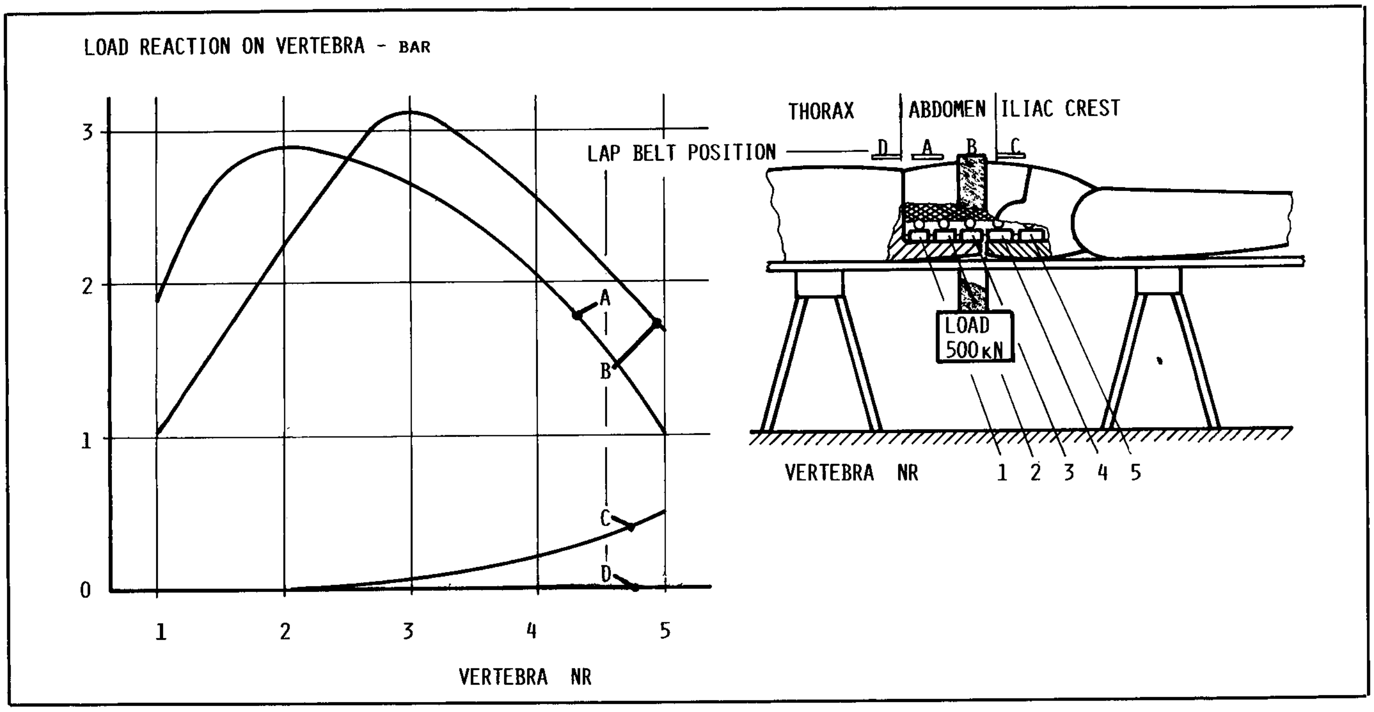

Figure 1. Correlation between surface load and load reaction on vertebrae

surface into a smoother pressure distribution at the lumbar spine.

Both the piezo-sensitive wire and foil will require comparatively little space. Upon paralleling the individual circuits (i.e. 1 circuit per vertebra) the sensor will perform like a fully-expandable tube package, which is a disadvantage compared with the actual hydraulic sensor. The actual performance and possible improvements by means of different wiring was not studied in depth. Therefore the further development work was concentrated on the hydraulic sensor.

Description

The total measuring device is shown in fig. 6. It consists of two separate packages of non-expandable reinforced tubes. The tube package for measuring in

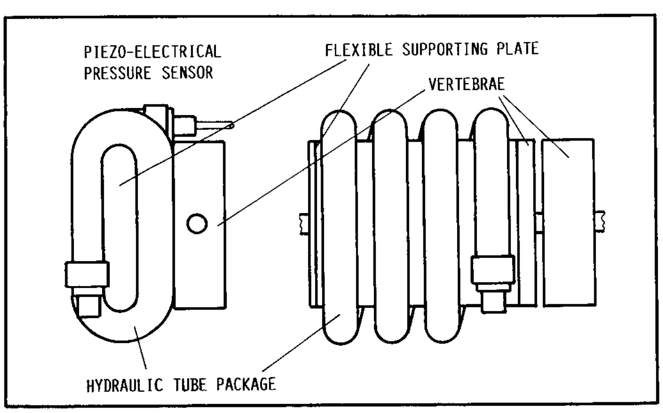

Figure 2

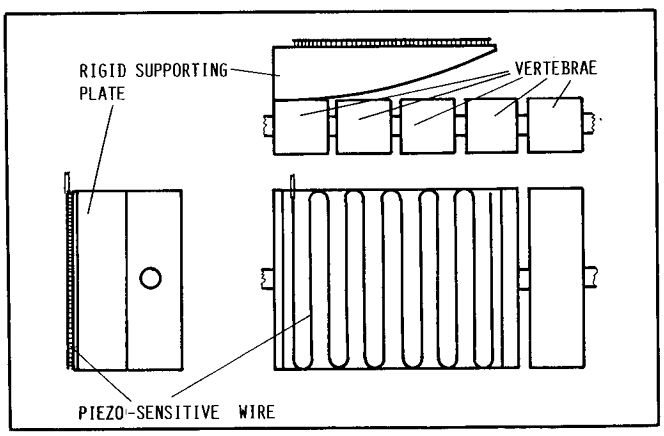

Figure 3. Piezo-wire sensor

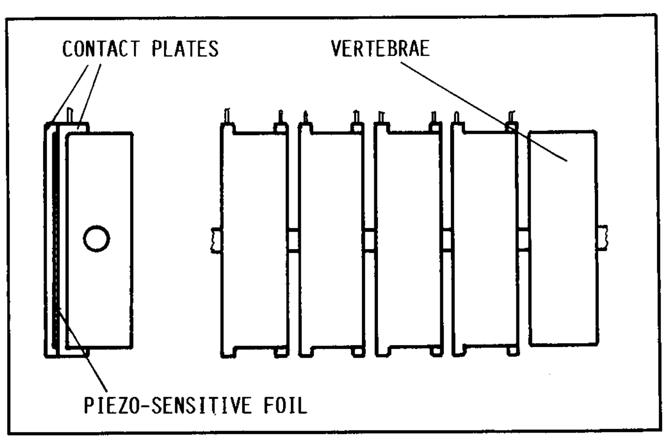

Figure 4. Piezo-foil sensor

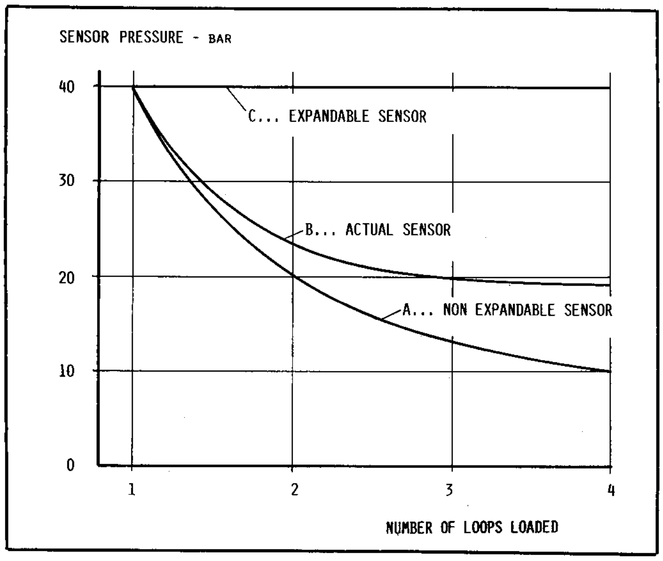

Figure 5. Influence of Load Distribution onto Sensor Reading

x-direction is helically wrapped around a flexible supporting plate with little space between the loops, thus avoiding interaction between the loops upon bending. It is placed against the lumbar spine. The tube package measuring the z-direction consists of a half loop which is sewn on a rigid plate, which again is screwed onto the base of the inner thorax (see fig. 7).

Both tube packages are equipped with an inlet valve at one end of the tube, the other end is equipped with a piezo-electrical-pressure sensor. The tubes are filled with hydraulic oil, pre-pressurized at approx. 4 bar.

Upon crash, parts intruding into the abdomen initially load the foam abdomen generating a proportional pressure increase within the tube package. This is shown by the piezo-electrical-sensors by means of a suitable measuring device including a charge amplifier.

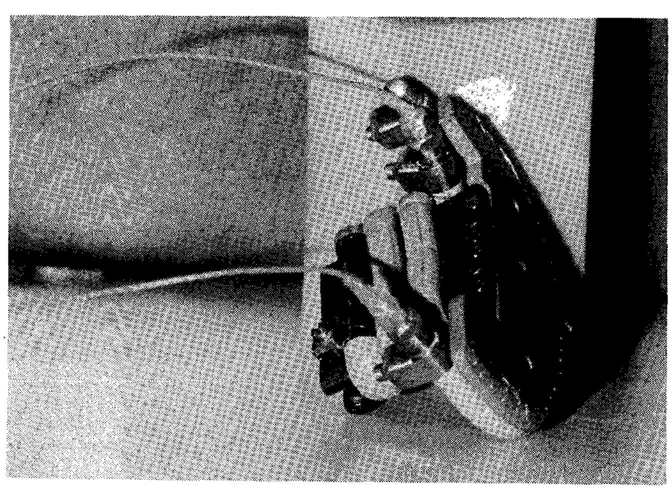

Figure 6. Hydraulic x- and z-sensor

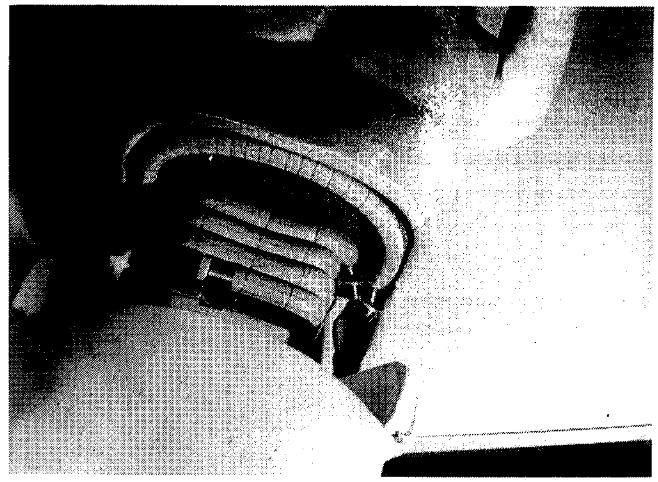

Figure 7. Hydraulic sensor implanted into the dummy

Material Specification:

Piezo-pressure-sensor:	Kistler 601A plus adapter
Tube:	soft-PVC, TREVIRA lining, bursting pressure 30 bar i.D. 8 mm o.D 11 mm
Hydraulic oil:	Renolin MR 1, Visc. 5 mm^2/sec at 40°C, density 0,85 g/cm^3 at 15°C
X-Support-Plate:	Vulkollan, flexible
Z-Support-Plate:	PVC, solid

Note: The x-support-plate is bolted to the z-support-plate. The z-support-plate is bolted to the lower thorax so that a compression of the thorax is not transferred to the z-support-plate.

Inlet-Valve:	Car tire type air valve
Plug:	Metal plug
Hose Clip:	Type UNEX high pressure hose clip.

Upon covering the lumbar spine by the x-sensor downwards in the direction towards the pelvis, it was taken into consideration that no interaction between the x- and z-sensor will occur upon extreme bending of the dummy torso. Therefore the lower thorax rim will not contact the lower loop of the x-sensor (See fig. 7).

Fig. 8 shows the schematic system of the measuring device. The charge amplifiers, Type Kistler, were installed on the test sled in order to reduce the length of the input wire. The amplified signals are transferred to a stationary digital memory scope and furtheron to the plotter.

Calibration

The calibration was done on a dynamic drop test system as used for helmet testing (see fig. 9). By means of a drop weight a defined force impulse can be applied to the sensor corresponding to the time

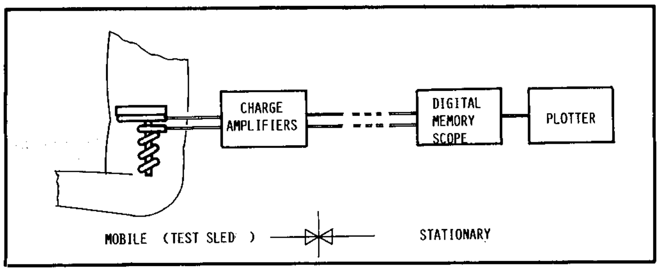

Figure 8. Schematic system of measuring device

history of the impulse upon actual test. The load can be varied by changing the drop height, the duration of the impulse can be adjusted via the foam at the drop weight. Readings were taken by means of a force transducer.

The calibration curves are shown in fig. 10 and 11. The ordinate shows both voltage and pressure obtained by varying loads. As a result it can be stated that the desired linear calibration curves could be achieved.

The influence of the pre-pressure onto the measuring signal is shown in fig. 12. The sensibility is increased with increasing pressure, however, this influence is relatively small at pressures above 2 bar. Due to sufficient sensibility and to reduce the loads onto the tube package a pressure of 4 bar was selected. A pressure drop down to 2 bar was acceptable.

Test Work and Results

The necessary dynamic tests were done on test sleds at Römer-Britax, Germany (see fig. 13) and at TNO, Holland according to ECE 44/02 requirements.

In order to prove the desired reproducibility the following test set-up was chosen:

Seat shell with impact shield and two-point lap belt (Römer PEGGY Seat)

Dummy: TNO P3

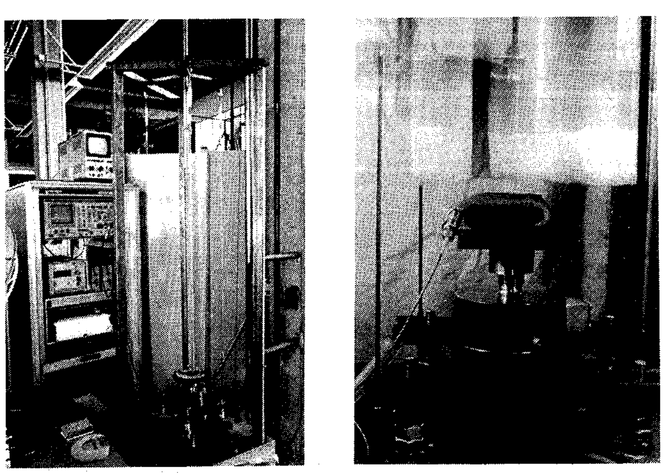

Figure 9. Drop test system for dynamic calibration

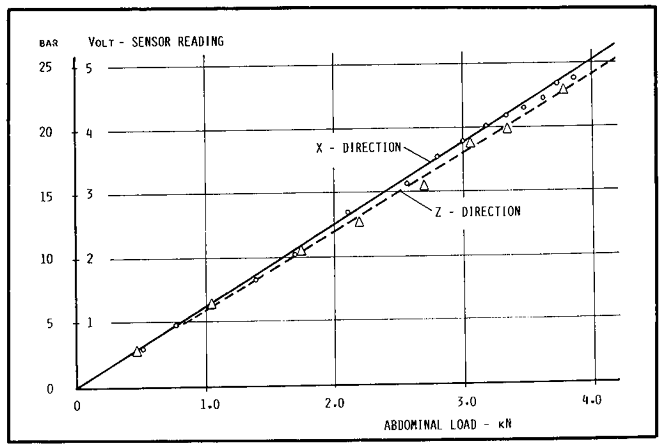

Figure 10. Calibration curves for P 3—dummy sensor

As a result almost identical curves can be stated with a max. deviation of $\pm 9\%$. This result taken from two different test sleds with different P3 TNO dummies is considered to fulfill the important requirement of good reproducibility (see fig. 14).

To investigate how well the sensor system is capable to discriminate extreme loading conditions the following tests were done. All readings of the abdominal sensor were compared with the analysis of high-speed crash films.

Note: Fig. 15 and fig. 16 do not indicate any substantial interference between the x- and z-sensor, since there are results with high x-readings and low z-readings and vice versa depending on the test set-up. At the same time these results confirm the need to separately measure the loading in x- and z-direction.

In comparison to other measurements taken such as sled and chest deceleration and head excursion there seems to be no interference nor time lagging of the recorded readings.

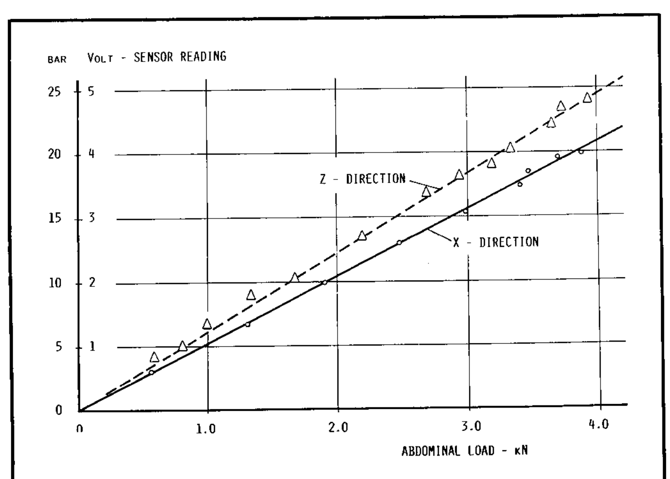

Figure 11. Calibration curves for P 10—dummy sensor

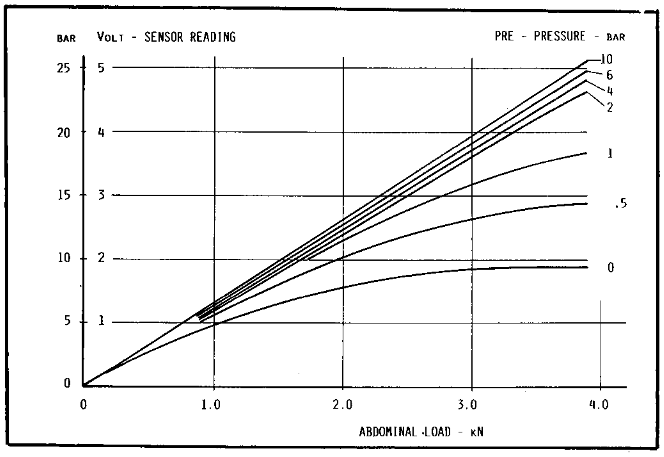

Figure 12. Influence of pre-pressure onto sensor reading

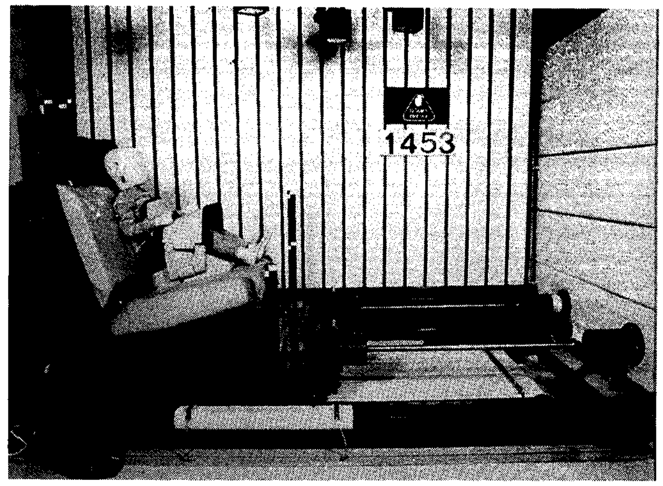

Figure 13. Dynamic test sled (Römer-Britax)

Conclusions

- The basic concept as described in [6], which was designed for the Part 572 dummy, is also applicable to the TNO child dummy family.
- The direct measurement of abdominal loading without the use of a transducer cylinder is both desirable and feasible.
- Due to the complex motion upon submarining it is advisable to separately record abdominal loading both in x- and z-direction.
- In comparison to the analysis of the high-speed crash film the proposed sensor device with its continuous measurements provide a more accurate tool to assess the degree of abdominal loading.
- The tests done with the proposed system show the desired reproducibility and can well discriminate between high-, medium- and no-loading. The proposed system seems to be superior to other Yes-No detectors such as the present modelling clay method. It records the actual abdominal loading, i.e. in kilo-

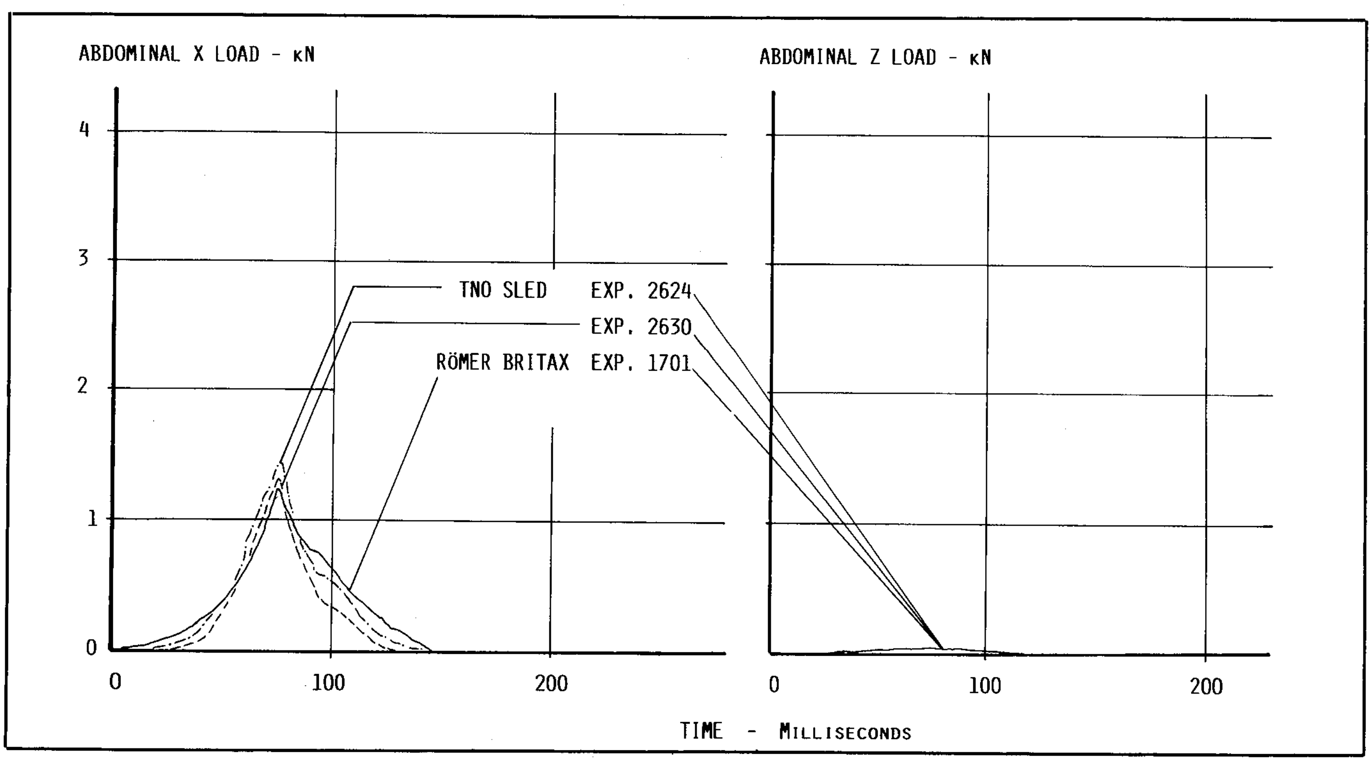

Figure 14. Abdominal load on TNO P 3 using seat shell with impact shield

Load condition	Restraining System	Result
No-load/No-reading	booster cushion with additional crotch strap	Virtually no-reading, both in x- and z-direction
Medium-load/ Medium-reading	seat shell with impact shield, restrained by 2-point-belt	full impact shield with low surface pressure has acceptable abdominal loading
High-load/ High-reading	seat shell without impact shield, lap-belt placed over abdomen	extremely high abdominal loading in x-direction
	seat shell with 4-point-harness, with slack without crotch strap; shell in reclined position	extremely high abdominal loading in z-direction

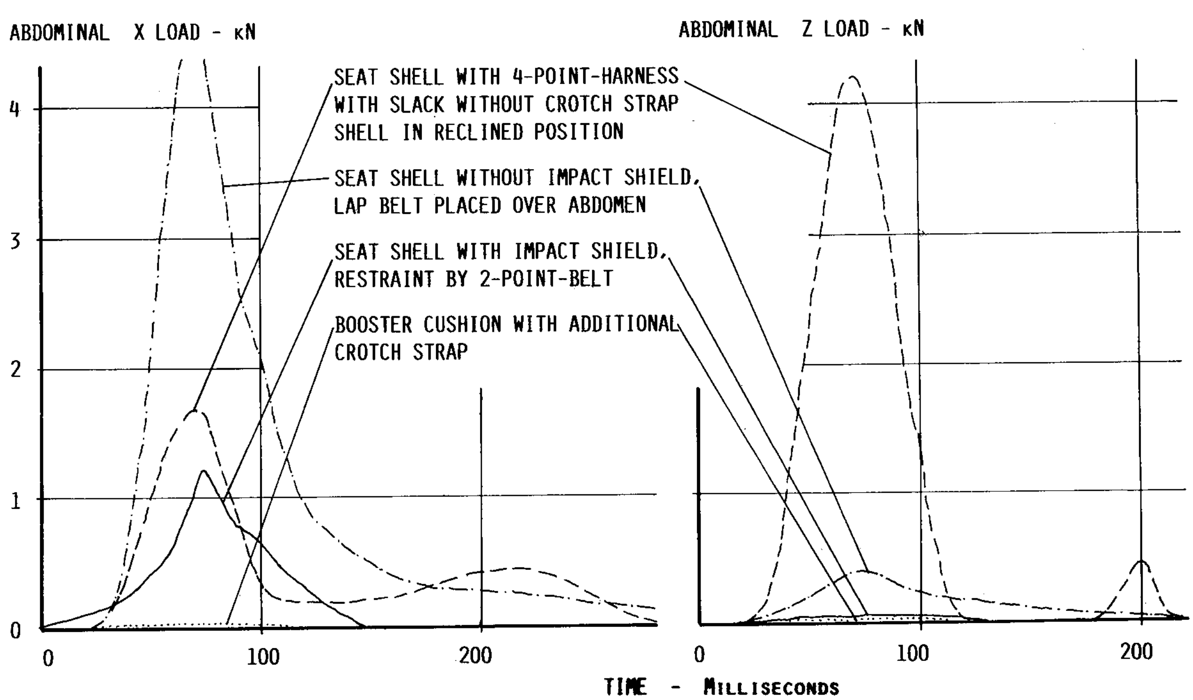

Figure 15. P 3—Dummy-sensor readings under different load conditions

Load condition	Restraining System	Result
No-load/No-reading	booster cushion with lap belt pulled down by additional crotch strap	Virtually no-reading, both in x- and z-direction
Medium-load/ Medium-reading	booster cushion Type A and B with 3-point-belt, ideally placed.	acceptable reading. [9] Due to dummy size and test set-up higher reading than for P3 (see fig. 15).
High-load/ High-reading	booster cushion with 3-point-belt, applied with slack.	extremely high reading in x-direction

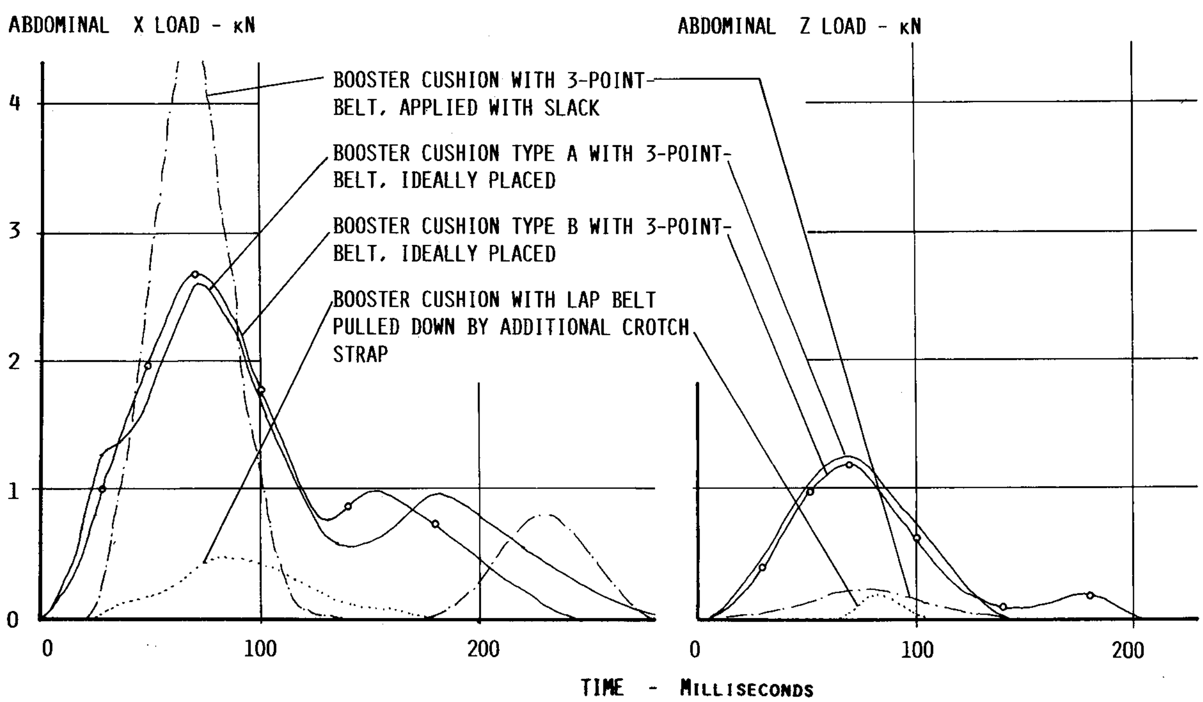

Figure 16. P 10—Dummy-sensor readings under different load conditions

Newton, thus providing the possibility for defining biomechanical limits.

- The application of the sensor system into the dummy abdomen does not result in other malfunction of the dummy behavior.

Acknowledgement

The authors like to thank the staff of the TNO test department and Manfred Bigalke and Kurt Weiss of ADAC for their valuable contribution in achieving the test results.

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Development of an Advanced Dynamic Anthropomorphic Manikin—ADAM for Military Applications

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Abstract

Ejection from aircraft at high speeds poses severe injury hazards to the crewmember. As performance characteristics of aircraft are further improved the protection capabilities of ejection systems must also be improved to assure the safety of the crewmember. The demonstration of these ejection system improvements requires extensive testing with manikins that can both effectively evaluate the performance of the ejection seat and assess the injury potential to the crewmember. The United States Air Force (USAF) is embarking on a new effort to design and develop an Advanced Dynamic Anthropomorphic Manikin (ADAM) with improved biofidelity and instrumentation over currently available escape system testing dummies. This effort will provide for the development of two prototype (one small and one large) instrumented, anthropomorphic manikins for testing, evaluating and qualifying high-performance aircraft escape systems (including the restraint and harness system effectiveness.) Discussed will be the design specification for ADAM, including the required experimental verification to demonstrate that the manikins mimic specified human biomechanical responses and are adequate for ejection system testing. Among the required responses are that it provide a human-like reactive live load into the ejection seat and possess realistic dynamics and kinematics due to windblast, impact, vibration and acceleration forces representative of those encountered during ejection from aircraft.

Introduction

The USAF's Crew Escape Technology (CREST) Advanced Development Program Office has begun a program to develop and demonstrate a prototype ejection seat capable of safe operation at all airspeeds up to 700 KEAS (Knots Equivalent Airspeed). This system will be capable of producing steering forces through the mechanism of vectored thrust rockets. By control of the vector thrust line with respect to the seat/occupant center of mass, stable flight should be achievable. By vectoring the thrust line away from the seat/occupant center of mass, a turning moment would be generated to permit steering. Due to the high multidirectional angular and linear accelerations and high windblast forces involved in an ejection, and especially those conditions to be addressed under the CREST program, any human occupant will slump and moved around involuntarily in a seat, even with an effective restraint system. This motion will shift the seat/occupant center of mass, change the seat/occupant moments of inertia and could create undesired turning moments capable of destabilizing the seat. The seat flight control system must be capable of overcoming such destabilizing moments as well as any potentially destabilizing aerodynamic moments encountered during high-speed ejections. To demonstrate this important capability of the seat flight controller, it is necessary that ejection tests be conducted with manikins designed to react dynamically as a human does. Such manikins must have individual body segments with proper centers of mass, moments of inertia, articulation flexibility and deformation similar to that of a human. Present ejection system testing manikins tend to be very crude human analogues with respect to dynamic and kinematic responses and are inadequate for evaluating ejection seats for which human body dynamic reactive forces are a factor in seat performance. Manikins developed for automotive safety testing, while having much better biofidelity, do not have high durability, longitudinal spinal response or a self contained data acquisition system. The Advanced Dynamic Anthropomorphic Manikin (ADAM), Figure 1, is a USAF program to design and fabricate an advanced instrumented manikin suitable for use in high-performance aircraft escape system testing. In addition to improved biomechanical response properties, the manikin will have a data acquisition system to measure, record and transmit its responses and the data from the escape system.

Phase 1 of the program is subdivided into Tasks 1, 2 and 3. Task 1 of the program was initiated on 16 September 1985 and was the basis for the development of a System Specification for the small, mid-size and large male manikins. Task 2 consisted of preparing detailed design studies to meet the requirements as set forth in the System Specification from Task 1. Task 3 consists of the detailed engineering design and fabrication of a small and large prototype male manikin. These prototypes will be tested at Wright-Patterson Air Force Base against the System Requirements established in earlier tasks. The anticipated acceptance date of the prototypes is August 1987.

In Phase II the contractor will fabricate five manikin sets and two sets of support equipment to be tested in an actual ejection environment. A manikin set will be composed of a small and a large manikin. The first set of manikins is scheduled for delivery in August 1987. The support equipment will be composed of all hardware and equipment which is not mounted on the manikin when it is being used in ejection testing.

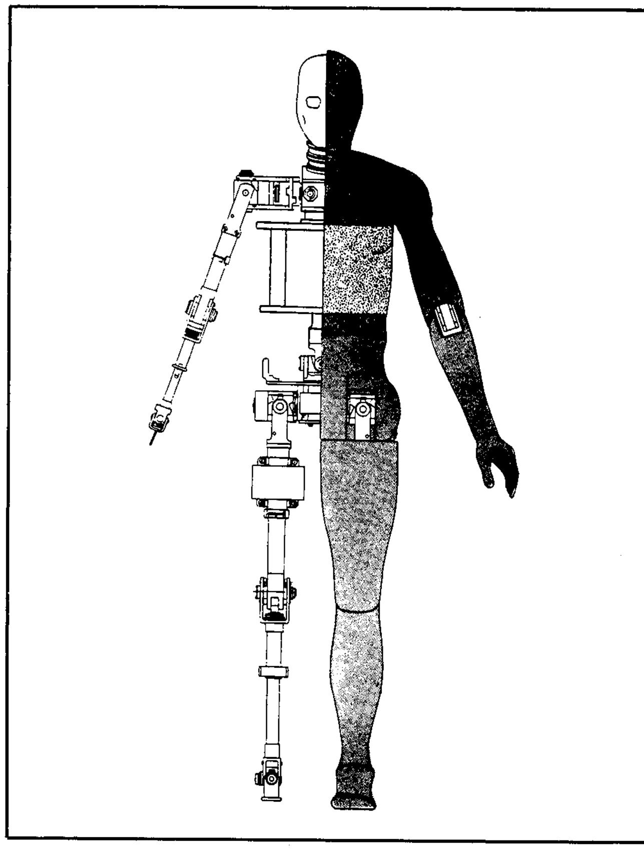

Figure 1. Advanced dynamic anthropomorphic manikin (ADAM)

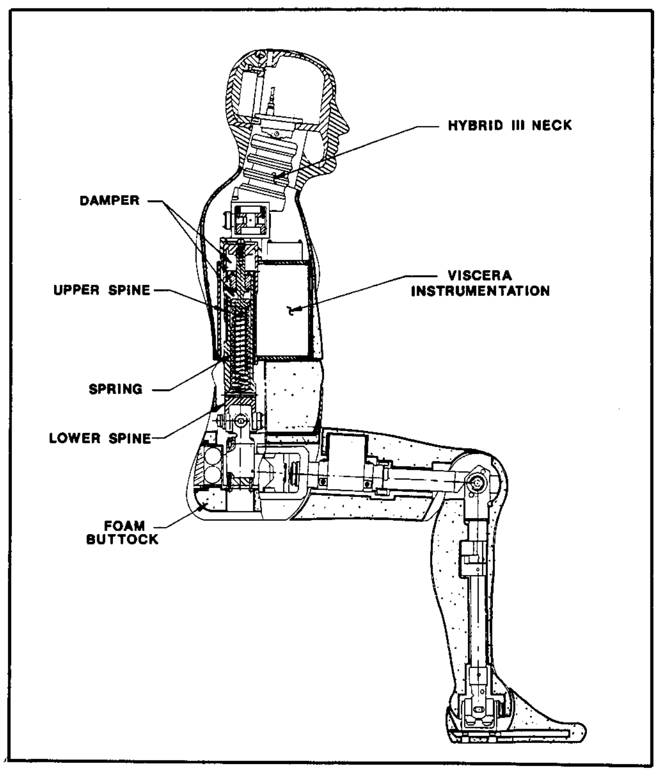

Figure 2. ADAM cross section showing spinal design

Discussion

A number of capabilities have been designed into ADAM to make it specifically applicable to testing of aircraft ejection and protection systems. Although these designs are directed towards ejection system applications it is expected to have applicability in many other areas concerned with human response to mechanical forces. ADAM must exhibit human-like dynamic response for G_x, G_y and particularly G_z exposures to assure that ADAM will react into the seat as a human would if ejected. ADAM will be based on USAF flight crew anthropometry with refined segment and total body inertial properties and proper joint articulation and motion-resistive properties. In order to analyze all the data to be collected during testing, an on-board data acquisition system will be developed to collect, store and transmit data and provide calibration for all sensors.

To provide a design baseline for ADAM a state-of-the-art study was completed in 1984. Among the manikins considered were the CG GARD dummy, which is the current standard ejection system testing manikin, and a number of automotive testing manikins including the Part 572—(which is the standard automotive safety compliance testing dummy), the Hybrid III—(which is an improved Part 572), the VIP, the OGLE manikin from the United Kingdom, Dynamic Dan—(which was the first attempt to build in spinal dynamic response) and the Limb Restraint Evaluator (LRE), developed by the USAF primarily to evaluate limb restraint system effectiveness. It was found from the study that the state-of-the-art manikins exhibited inadequate gross motion responses in the G_x, G_y and particularly the G_z directions. These manikins, with the exception of the LRE, have limited instrumentation and data acquisition systems to measure and record their responses and the data from the system being tested.

The ADAM technical requirements can be divided into two areas: mechanical design and instrumentation. The mechanical design includes, mass distribution, anthropometry, joint structure and spine development. The instrumentation area includes sensor selection; signal conditioning; data processing, control and storage; software development and telemetry design.

The anthropometry of the manikin includes traditional anthropometric dimensional measurements, individual segment masses, centers of mass and principal moments of inertia, segment surface definitions and joint centers and ranges of rotation. The approach taken in specifying these data was to reference all data to individual segments by defining anatomical coordinate systems for each segment based on at least three body landmarks identifiable on human body segments. A cosine matrix relating the segment principal inertial axes to the anatomical axis system for each segment was provided. The joint centers of rotation were given as vectors in the segment anatomical system. The anatomical coordinate systems were chosen to provide a basis for relating mass distribution and joint locations to human anatomical structure. While the dimensions are all segment based, they are compatible with overall body traditional anthropometry both for the standing and seated position.

Specifications for two manikins are being developed. These are designated as small and large. They are based on multiple stature and weight regressions on the 1967 USAF male flying personnel survey (Reference 1) data for the 3rd and 97th percentile, respectively, with a projected time growth factor for the year 1990.

The body is to be made up of 17 segments consisting of the head, neck, upper arms, forearms, hands, thorax, abdomen, lower torso or pelvis, thighs or upper legs, calves or lower legs and feet. The manikins will have articulating joints for the shoulders, elbows, wrists, hips, knees and ankles as well as distributed articulations for the spine including the neck. The mechanical design of these articulation mechanisms will emphasize human biofidelity. Since the primary vertical load transmitting structural member in the human torso is the spine, ADAM's spinal elements (Figure 2) are to be designed to have elastic and viscous properties such that it's seated dynamic response to G_x, G_y and G_z accelerations are similar to those of a seated human.

The instrumentation system involves a major challenge because it must be designed to fit within the small size manikin. The small manikin has very limited space in the pelvic, abdomen and thoracic areas in which to locate the instrumentation, therefore hybrid circuits will be required to make the electronic design as small as possible. The manikin will have 128 data channels with a sampling rate that is operator selectable from 100 to 1000 samples per second. The data to be sampled will come from strain gauges, transducers, potentiometers, accelerometers and load cells. Anti-aliasing filters will be used to filter out high frequency interference and provide a selectable bandwidth from 20 to 200 Hz. Data from the seat or vehicle can also be collected in the ADAM by use of a quick disconnect hardwired connector between the seat and the manikin.

The heart of the data processing system will be a Motorolla 68020, 32 bit microprocessor with 512K bytes of on-board storage. This will provide storage for 128 channels of data at a maximum sampling rate of 1000 samples/sec for four seconds. There will also be storage for pre and post test calibration measurements. The data processing system will also handle all I/O and includes an A/D converter for on-board storage and the control of telemetry transmission to nearby receivers. Software will handle system calibration, setting up data collection parameters, sensor calibration and system diagnostics. A telemetry system which can transmit data through a skull mounted antenna at a rate of 1.048 Mbits/sec using the standard Inter-Range Instrumentation Group (IRIG) 106-80 protocol provides data acquisition redundancy.

Performance Compliance Testing

To fully evaluate ADAM a series of tests will be conducted by the Air Force. The first test will be conducted on a Six Degree of Freedom Motion Device which can apply simultaneous multiaxis linear and angular oscillatory forces. It will correlate the mechanical frequency response characteristics of the manikin with the seated human driving point impedance or transmissibility data. Additionally, whole body impact tests will be performed on a Vertical Deceleration Tower and a Horizontal Impulse Accelerator. These tests will correlate the dynamic whole body response characteristics of the manikin for $-G_x$, $+G_y$ and $+G_z$ impacts. The primary response correlations will be head and chest accelerations and reactive seat loads. Emphasis in these tests will be primarily on Z axis impact responses. Impact sled tests will also be conducted to validate the durability of the fully functioning manikin. One series of tests will involve exposures to an acceleration pulse of approximately half sine wave shape, 120 msec duration and 45G peak values. These tests will be conducted in the positive and negative directions of G_x, G_y and G_z. A second series of tests will involve exposures to half sine pulses of 6 msec duration and 100G peak magnitude. Due to limitations in the test facility, these tests will be conducted only in the $-X$ and $+Z$ directions. In another series of tests, the manikin will be dropped onto a concrete pad to simulate worse case ground landing impact. Immediately after a successful drop test, the manikin will be placed in a thermal environment with ambient temperature of 100°F, humidity of 50% and a simulated or real mid-day solar radiant heat load for four hours. For a successful test, the instrumentation must function to record and transmit data during the drop test and must not lose the data during the thermal stress test. Additionally, the manikin itself must not suffer any structural damage. The ADAM manikin, including its instrumentation, must function without loss of data and with no damage, that may impair function, to pass these tests.

Products and Applications

The ADAM program's primary objective is to provide an instrumented manikin, with detailed engineering drawings and documentation, to satisfy advanced ejection system testing requirements. However ADAM has other applications as well. ADAM can be used to test head or helmet mounted equipment, such as chemical defense or avionics equipment. The manikin can be used to evaluate windblast protection concepts and the effectiveness of capture/haulback active restraints. It has possible applications in crash attenuation seat testing involving energy absorbing seats, such as those found in helicopters. ADAM has commercial applications as well. It can be used in car crash impact/rollover testing where gross body and limb motion are of concern and a self-contained instrumented manikin would be useful in evaluating occupant motion and interaction with the vehicle.

Summary

The ADAM program is an attempt to overcome the shortfalls and advance the state-of-the-art of escape system testing manikins. It is durable, possesses human-like dynamic mechanical responses, and has an internal instrumentation system for data acquisition, on-board storage and telemetry. The first production ADAMs are scheduled for delivery in August 1987 and will be used in CREST's high performance aircraft escape seat testing program.

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The Development of a Dynamic Human Analog (Written only paper)

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Abstract

The current efforts associated with upgrading the capabilities of in-service ejection seats and the development of new ejection seats incorporating state-of-the-art technology such as the CRew EScape Technologies (CREST) Program have resulted in a requirement for an Advanced Dynamic Anthropomorphic Manikin (ADAM) that represents the static and dynamic characteristics of the human body to a degree not present in existing designs. Systems Research Laboratories, Inc. (SRL), is presently under contract to the Armstrong Aerospace Medical Research Laboratories (AAMRL), Wright-Patterson Air Force Base, Ohio, to develop and fabricate ADAM as a small, mid-size, and large size human analog.

This paper will summarize the basic aims and goals of the ADAM development program and some of the basic features of the ADAM that set it apart from previous manikins designed for ejection testing. The unique advanced features of the ADAM to be discussed will include the manikin anthropometry, flexible spine/viscera system, body articulation, and the unique instrumentation system.

Introduction

While the first ADAM was created many thousands of years ago, we are now trying to create a mechanical replica of that complex machine—also called ADAM, which stands for Advanced Dynamic Anthropomorphic Manikin. In the case of ADAM the manikin, he is not the first to be created in an attempt to duplicate the characteristics of man to test ejection seats during an escape sequence from an aircraft. He is, in fact, the grandson of the original attempt to duplicate the characteristics of man to test ejection seats. The grandfather of ejection manikins is the GARD manikin, jointly developed by Grumman Aircraft and Alderson Research Laboratories in the 1950s, and was named after the two companies as Grumman-Alderson Research Dummy (GARD). A picture of the GARD is shown in Figure 1. It is also known as the CG dummy as it placed the center of gravity of the human analog in the seated position at the same location as a human when seated in an ejection seat. While the GARD manikin duplicated the location of the human CG in the ejection seat, the body articulations were very limited; therefore, the dynamic characteristics of the human were not simulated to a significant extent. The GARD, while having limited capabilities, has been invaluable in the development of ejection seats and, in fact, is still the primary manikin used in a majority of ejection seat testing when the mass characteristics of the human must be accounted for to evaluate seat performance.

It was more than a quarter of a century since the GARD was developed before a new attempt was undertaken to develop a more complete replica of the human for use in testing of updated versions of current operational ejection seats. The Limb Restraint Evaluator (LRE), shown in Figure 2, is the son of the GARD manikin. Reference 1 provides a detailed discussion of the LRE development. All of discussion of the LRE development. The LRE was designed and constructed to test the effectiveness of limb restraint systems being incorporated into current operational ejection seats. As such, the LRE is a highly articulated manikin that was designed to withstand aerodynamic loadings up to 700 KEAS and to incorporate an onboard data recording and telemetry system to handle up to 96 channels of sensor data generated by accelerometers and potentiometers. This 95th percentile manikin, developed by SRL, used existing molded skin shapes developed by Alderson Research Laboratories, Inc., for the VIP-95 manikin, has been tested for dynamic loadings up to 40 Gs in the Gx and Gz directions and to 30 Gs in the Gy direction in the test facilities at AAMRL and was recently successfully tested in a 600 KEAS windblast at the Dayton T. Brown facilities. It is anticipated that ejection testing of the LRE will be undertaken at Holloman Air Force Base, New Mexico, in the near future.

One might ask, with the highly articulated and instrumented LRE soon to be available for use, why is ADAM needed? Very simple—the need for increased biofidelity in the duplication of the dynamic characteristics of the human. While the LRE has highly articulated limbs, the dynamic response of the body torso was not duplicated except for that which was available through the use of a triply articulated lumbarspine. With the development of the CREST ejection system, which will be a thrust vectored rocket controlled seat, the dynamic characteristics of the entire human body must be duplicated in order to verify during ejection testing the stability of the CREST feedback control system. The challenge for the design of ADAM is to create a dynamic analog of the human, at least to the degree required to adequately test the capabilities of the CREST escape system in an ejection environment up to 700 KEAS. ADAM is considered to be a critical item in the development of an advanced ejection system as the human is an extremely large weight fraction of the combined seat/man system. As such, the dynamic motions of the manikin can develop significant forces and moments which must be overcome by the CREST thrust vectoring system.

The development of ADAM is a unique challenge, and this paper will attempt to illustrate the challenges that are being met and the manner by which the dynamic analog of the human is being designed and how it will be tested. In order to emphasize the unique challenges that must be met in the development of ADAM for ejection testing, its characteristics and requirements will be compared, where appropriate, with the Hybrid III manikin, which will soon be the standard for the testing and evaluation of safety devices in ground transportation vehicles.

Technical Discussion

General Characteristics of ADAM

The ADAM will be a highly articulated manikin in which the dynamic response of a human to dynamic loading, which is developed during an ejection at speeds to 700 KEAS, will be duplicated to the greatest degree possible. The manikin is being designed to the small, mid-size, and large sizes as defined in Tri-Service MIL-Handbook for the Standard Model Aviator (currently in preparation, Reference 2). As such, ADAM will have anthropomorphic dimensions and mass characteristics which are unique and, thus, will be the only manikin which will meet the future Tri-Service standards. In addition to duplicating the dynamic response of a human during ejection, ADAM will have an advanced onboard data recording and telemetry system that will handle up to 128 channels of data from sensors on the manikin and from the CREST seat. The redundancy of the data system, with both onboard storage and telemetry, is to ensure that a complete set of transducer data is obtained regardless of data dropout due to misorientation of the telemetry antenna during an ejection test. In the following sections of this paper, the characteristics of the unique features of ADAM will be discussed in some detail.

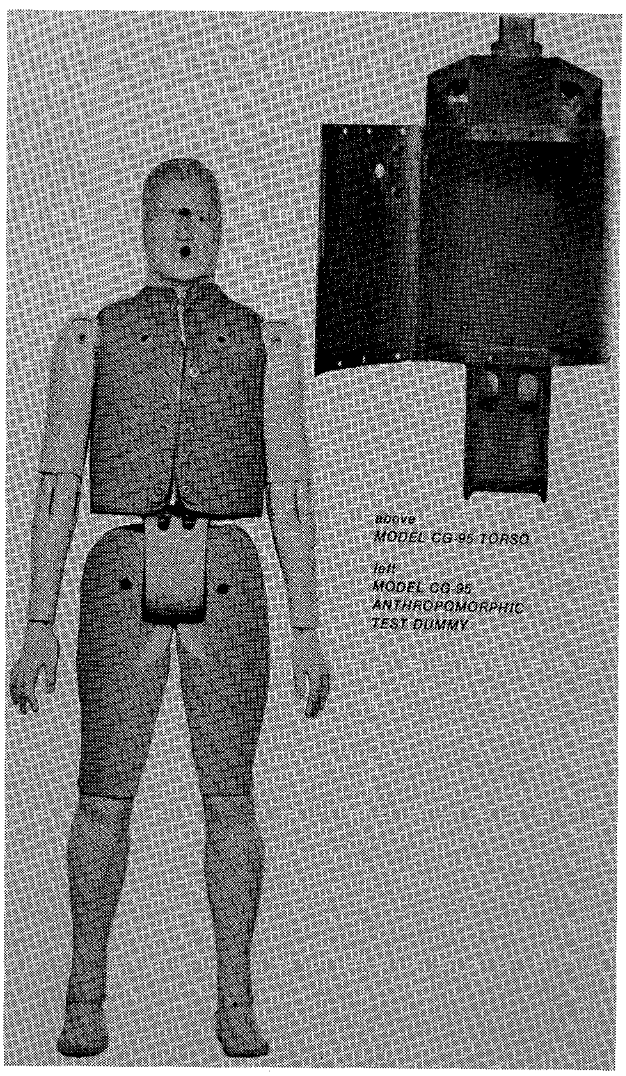

Figure 1. The GARD/CG ejection manikin

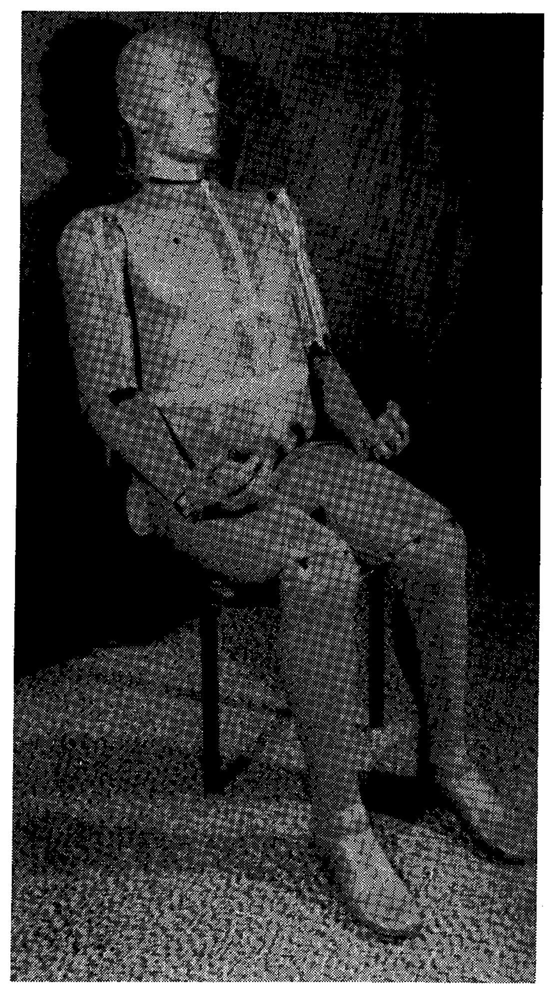

Figure 2. The LRE manikin

Requirements of an Ejection Manikin

The requirements that must be met in the design of ADAM in order to provide a manikin that will be suitable to test an ejection seat are many and, in some cases, rather demanding. In order to illustrate the challenge of meeting the requirements, some of the unique requirements of ADAM are compared to those of the Hybrid III design in Table 1.

As can be seen from Table 1, the requirements of an ejection manikin which duplicate the required features of a human in a dynamic environment can be more demanding than those of a manikin used for testing the standard safety features of ground transportation vehicles. While the comparison could be expanded to include joint articulation, inertial characteristics, and mass, it is believed that the comparisons that are made illustrate the challenge of designing a suitable ejection manikin. It is noteworthy, however, that the manikins utilized in testing the safety characteristics incorporated in ground transportation vehicles have very specialized requirements for the duplication of skin/skull compliance for evaluation of localized head impact injury. This demanding impact compliance is also generally incorporated in the knees in order to evaluate injury severity if the knee strikes fixed automotive structures during a crash. While neither of these requirements are inherently incorporated in the ADAM design, as the manikin wears a helmet during ejection tests and its knees do not interact with the structure of the cockpit, they could be easily incorporated in ADAM by using the proper head form and altering the skin covering in the knee.

One of the unique features of ejection manikins is its effect on the vehicle it is testing. The reason for this rather unique feature is that the test manikin is a large weight fraction of the man/seat combination (up to approximately 60 percent). Because the manikin is such a large weight fraction of the total system, the dynamic motions of the manikin can have a large effect on the motions and stability of the complete system. This is not true with crash manikins except possibly in the test of crashworthy or energy deforming seats. Since the dynamic motions of the human can impart significant dynamic loadings to the total system, the response of the manikin to acceleration and vibratory inputs along and about all axes must faithfully duplicate the human response to these same dynamic forces. In addition, the dynamic response of the body components to windblast must also be faithfully duplicated, as the body motions resulting from these forces can cause large forces and moments to be applied to the escape system. The duplication of human biofidelity must, therefore, be as complete as possible, particularly for the CREST escape system since it will attempt to control these dynamic inertial forces and motions with thrust vectured rockets to create a stable and directed flying system.

Comparison of Manikin Characteristics

Mechanical. Table 2 presents a comparison of the basic characteristics of various manikins designed for ejection testing as well as a comparison with the Hybrid III. It can be seen that the anthropometric characteristics of the different manikins have significant differences. It is to be noted that the GARD and LRE dimensional and mass characteristics are based on the earlier standard aviator, and the Hybrid III characteristics are based on the general population and not just aviators. These different data bases should be kept in mind when comparing the characteristics of the manikins. It is noteworthy that the characteristics of the ADAM, which reflect the Tri-Service data, indicate that the standard male aviator is not only larger, but also heavier, than previous specifications for manikins, particularly for the mid-size or 50th percentile male. Another significant difference in the manikin designs is the number of articulations which represent human articulations. This increase, of course, represents the need for realistic duplication of human biofidelity in the manikins so that they can adequately represent the human during dynamic tests of the life-saving vehicle—the ejection seat. Another major difference in the manikins is the incorporation of transducers to measure the motions at the articulation points so that in-depth measurements of the body motions can be recorded to analyze the motions of the manikin during ejection. Obtaining and analyzing these data will be of extreme importance in the development of advanced limb restraint systems and ejection systems such as CREST.

Discussion of ADAM's Special Features

The ADAM Data Acquisition System. While a complete description of the ADAM instrumentation system is presented in Reference 3, the following description will briefly describe the features of the system. The ADAM instrumentation can be divided into two distinct systems—the analog signal conditioning hardware and the microprocessor system. Figure 3 is the block diagram of the instrumentation system. It shows that there are three boards that are involved in receiving the signals from various manikin sensors and CREST signals. These three boards represent the signal conditioning circuitry used in the data acquisition system. The block diagram also indicates that there are four modules in the microprocessor system that control the data acquisition process. Each subsystem shall be described in the following paragraphs.

Figure 4 outlines the entire signal conditioning circuitry incorporated in the instrumentation design. There are 72 manikin channels that are available for

Table 1. Manikin design requirements.

Design Requirements	ADAM	HYBRID III
Proper Acceleration Response		
Gx	X	X
Gy	X	-
Gz	X	-
Proper Vibratory Response		
X Axis	X	-
Y Axis	X	-
Z Axis	X	-
Yaw Axis	X	-
Pitch Axis	X	-
Roll Axis	X	-
Respose to Environmental Effects		
Windblast	X	-
Pressure	X	-
Temperature	X	-
Effects of Dynamic Response on Test Article		
X Axis	X	X*
Y Axis	X	-
Z Axis	X	-
Yaw Axis	X	-
Pitch Axis	X	X*
Roll Axis	X	-
Localized Body Impact Response		
Gx	-	X
Gy	-	-
Gz	-	-

*Possible on tests of crashworthy seats.

Table 2. Comparison of manikin mechanical characteristics.

Characteristics (percentile)	GARD 5	GARD 50	GARD 95	LRE 95	ADAM Small	ADAM Mid	ADAM Large	HYBRID III 5	HYBRID III 50	HYBRID III 95
Height (inches)	65.2	69.1	73.1	72.2	66.2	70.2	74.3	64.8	66.2	73.6
Weight Total (pounds)	132.5	161.9	200.8	214.0	139.5	179.5	215.4	147.	155.5	207.9
Number of Articulations		18		39		43			26	
Number of Instrumented Articulations		0		33		39			0	
Range of Motions		Limited		Near Human Limits		Human Limits			Limited	
Elastomer Stops		No		No		Yes, In All Joints			Yes, In Some Joints	
Flexible Neck		No		Yes		Yes			Yes	
Flexible Spine		No		No		Yes			Yes	
Articulated Lumbar Spine		No		Yes		Yes			No	
Torque Adj. Joints		Yes		No		Yes			Yes	
Pelvis Design		Seated/Standing		Seated		Seated/Standing			Seated	

use. Of these, 36 are channels available to measure low level signals (signals from sensors that require an amplifier), and 36 channels are available for measurement of high level signals (those sensor signals that do not require an amplifier). Each manikin channel has a provision for a computer controlled shunt calibration (RCAL) signal in order to verify the proper functioning of the signal path and to verify that the sensor is attached. Each manikin channel is passed through an eight pole Butterworth low pass filter. The low pass filter is included to prevent aliasing errors from being introduced into the digitized data. The filter selected is a switched capacitor technology filter which allows the microprocessor to select the cutoff frequency by changing the frequency of a clock driving the filters. The filters are organized into four banks and each is driven by its own microprocessor controlled clock. This feature allows computer controlled channel bandwidth of groups of channels.

The signals from the CREST seat will be amplified and filtered by signal conditioning circuitry within the CREST seat; therefore, no provisions have been included for signal conditioning of these signals. The instrumentation has been designed for 56 signals from

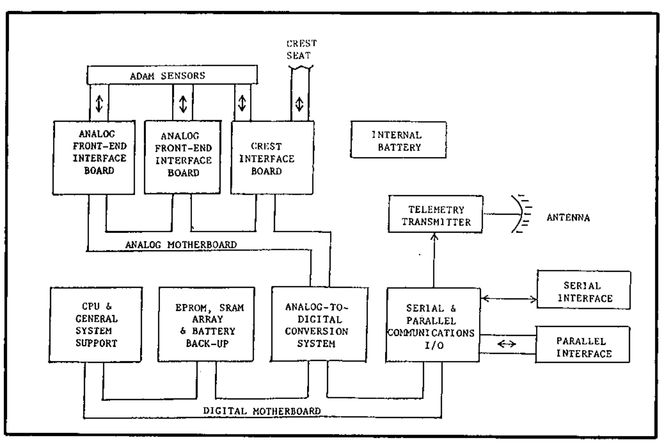

Figure 3. ADAM system block diagram

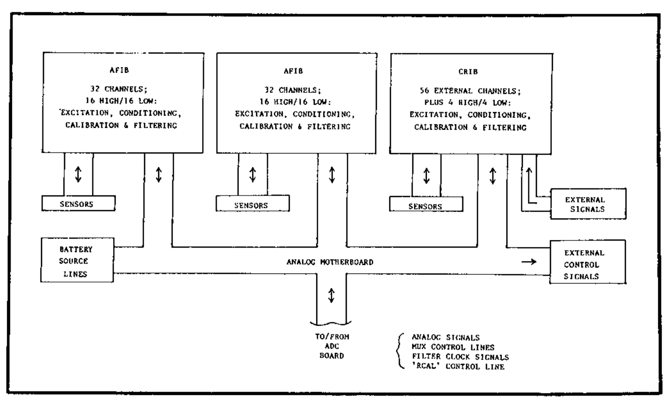

Figure 4. ADAM signal conditioning block diagram

the CREST seat to be digitized and recorded by the ADAM instrumentation.

Both the signals from the CREST interface and the outputs of the low pass filters of the manikin are directed to analog multiplexers. These multiplexers are electronic switches that are computer controlled. Each multiplexer is arranged so that it will output one channel of 32 to the input of the analog to digital converter. There are four 32 channel multiplexers to handle all 128 channels of the ADAM system.

The signal conditioning circuitry represents about two-thirds of the physical room required of the instrumentation system when standard integrated circuits are used. This represented a problem in the ADAM because there was no room in the small size manikin for circuitry of this size. In order to reduce the space required by this circuitry to meet the severe room restraints of the small size manikin, a hybrid microcircuit was designed to reduce the space required by the signal conditioning circuitry. A hybrid microcircuit is a miniaturization of a group of integrated circuits into a single integrated circuit package. This provides considerable room savings over individual integrated circuits. The ADAM hybrid combines the circuitry of 28 integrated circuits into a single integrated circuit that is 1.25 inches square (a 50 percent room reduction), and this allows the signal conditioning circuitry to fit within the manikin.

The block diagram of Figure 3 shows that the interface between the signal conditioning circuits and the microprocessor is the analog-to-digital (A/D) conversion system. This system is used to digitize the analog signals from the signal conditioning hardware to convert the analog signals into a form that the microprocessor can use. The A/D system also has the task of controlling the analog multiplexers. The A/D system block diagram is shown in Figure 5. The output from each multiplexer is fed into its respective A/D converter. Four A/D converters are used to increase the system throughput (number of conversions per second) by a factor of four. This occurs because four conversions can be done in the time that it takes one A/D to perform a single conversion. The resolution of the A/D system is 8 bits with 11 bit accuracy in the conversions. The A/D converter control logic controls the start conversion command and the address counters used to address the multiplexers. This logic is designed so that the multiplexer address counters can increment automatically with each conversion, or so that they may be loaded with an individual address. Logic is also incorporated into the system to prevent reading the A/D when it is performing a conversion, so that erroneous data is not read by the microprocessor.

The device that controls the entire data acquisition process is the microprocessor. Figure 6 shows a block diagram of the microprocessor and its associated hardware. The processor selected for the ADAM design is the Motorola 68020 microprocessor. This processor was selected due to its size, power consump-

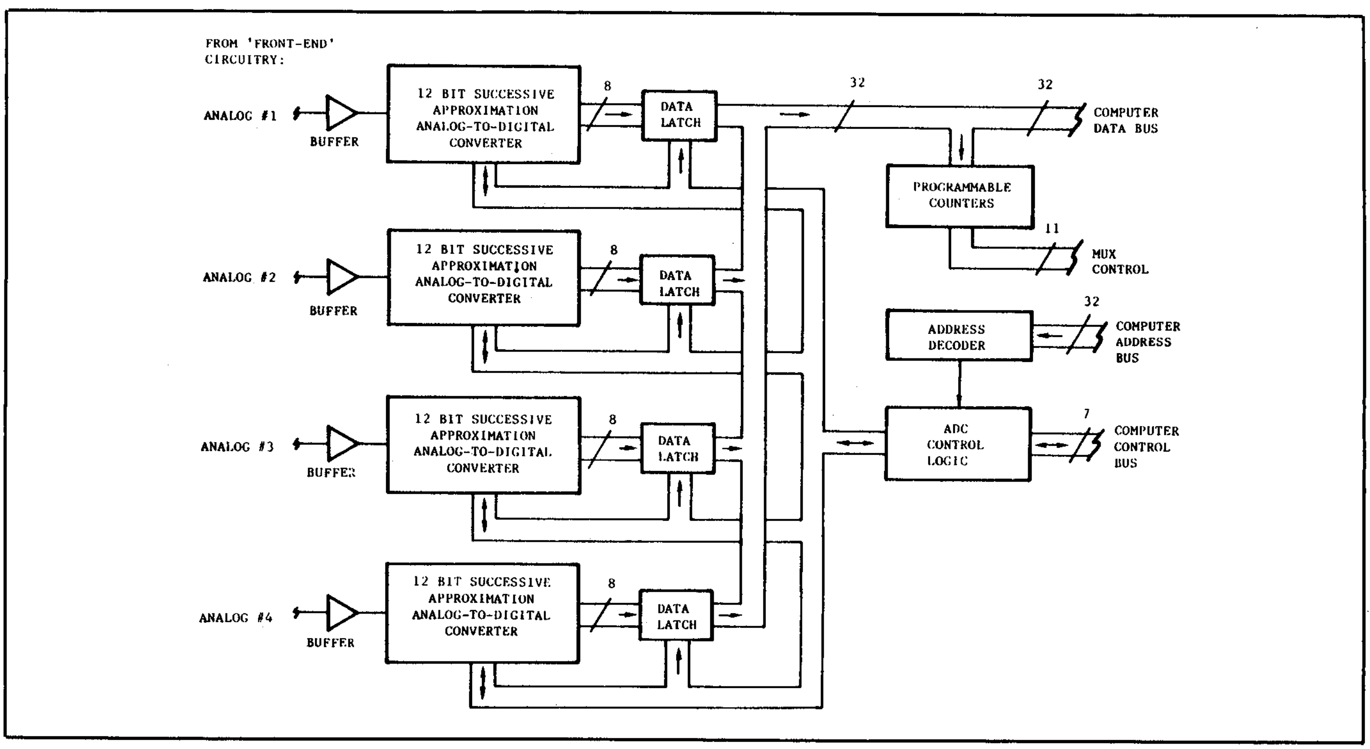

Figure 5. ADAM analog-to-digital conversion system

tion, 32 bit data bus, onboard instruction cache, and speed. The processor is driven by software to control the A/D system, store data in the onboard memory, send data to the telemetry port and offload the data by either serial or parallel input/output (I/O) ports. The processor is driven by 64 kilobytes of Erasable Programmable Read Only Memory (EPROM) which is used to store all diagnostic, run time, and posttest software routines.

The instrumentation has 512 kilobytes of static random access memory (RAM) located on a single board to store test data. A small portion of the memory is used to store pretest and posttest calibration data and to be used by the processor as scratchpad memory. A feature of this memory design is the fact that it is battery backed up to help prevent test data loss if power is inadvertently lost after data capture. The design of the memory board allows for expansion of the onboard memory as newer, denser memory devices become available.

Figure 6 also shows that communications with the instrumentation system can be achieved through the serial port. The serial port is designed for either RS-422 or RS-232 standard serial protocols. This serial communications capability is used to monitor diagnostics, select menu options for the test configuration, and various other functions which would require interactive input from the user. If desired, the data stored within the memory may be offloaded through this serial port.

The primary means by which data are offloaded from the instrumentation to the data collection device is by way of a high speed parallel port. This parallel port is a 32 bit port that is capable of offloading the data at a very high rate of speed. The entire memory could be downloaded in a matter of several seconds. This feature is very desirable since offloading the same data using a serial format would require several hours to complete, and time is not available in most test environments to accomplish a serial download.

Data can also be collected during the test using the pulse code modulation (PCM) telemetry output. This technique can be used with a landline link or with a radio link to a receiving station. This PCM output is generated internally by the processor utilizing a parallel-to-serial converter and a double buffer technique to prevent data dropout or dead spots. The telemetry output has a maximum bit rate nearing 2.4 megabits/seconds and has industry standard nonreturn to zero-level (NRZ-L) and biphase level (BIO-L) outputs at all bit rates. Figure 7 is a block diagram of the telemetry interface. Figure 7 shows that the timing that is critical to a PCM output is maintained by the use of a double buffering technique. When this buffer is empty, the processor is notified via an interrupt that the input buffer needs service. This technique allows two synchronous systems to run asynchronously.

This system provides the user with 128 channels of data acquisition—72 channels with full signal conditioning on the manikin and 56 channels able to accept preconditioned signals external to the manikin. The system incorporates switched capacitor filters with a cutoff frequency range from 50 Hz to 2500 Hz, and is capable of recording information from most common sensors in existence today. The instrumentation contains sufficient onboard memory to store 512,000 data points, and an onboard telemetry generator allows the data to be transmitted to separate receiving station. This feature provides redundancy in the data acquisition process.

The ADAM instrumentation is supplied with a comprehensive set of diagnostics to test the system operation and also provide assistance in system calibration and hardware troubleshooting. The system run time software provides routines that provides variable sampling rates of groups of channels and computer controlled sampling order. With this software, it is possible to sample the instrumentation channels at rates in excess of 1000 samples per second per channel while simultaneously storing the data internally and transmitting the data via the telemetry link. Many possible combinations of sampling order

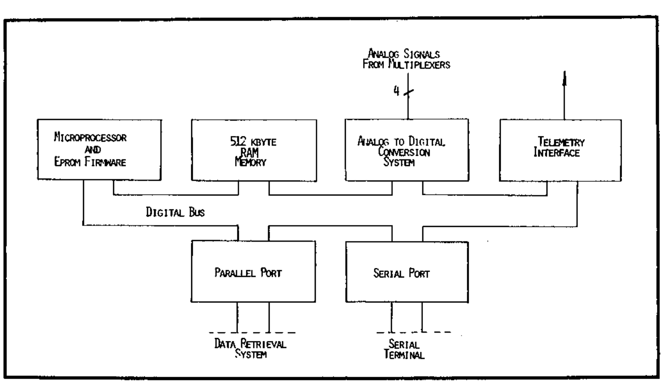

Figure 6. Microprocessor system block diagram

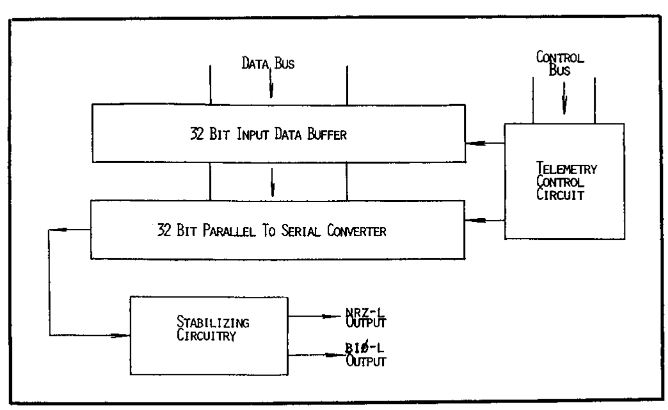

Figure 7. Telemetry interface block diagram

and sampling rate are possible just by changing the software controlling the entire system.

Modular construction techniques employed throughout the ADAM instrumentation allow for ease of expansion or reconfiguration of both the analog signal conditioning hardware and the digital computer section.

A brief comparison of the instrumentation systems ADAM with those available in other manikins will illustrate the extensive capabilities of the ADAM system.

Table 3 presents a comparison for some of the major features of the various instrumentation systems. A passive instrumentation system is one in which all the timing functions are preset as well as the frequency sampling characteristics of the various data channels. A completely computer controlled system is one in which an onboard computer can be used to control the characteristics of the data channels (i.e., sample rate, signal strength, calibration, frequency range, etc.).

As can be seen from Table 3, the instrumentation systems that were developed for the LRE and ADAM have a large capacity and great flexibility. While the instrumentation system developed for the LRE provided a quantum jump in manikin instrumentation, it lacked some of the flexibility required to satisfy the needs of ADAM, as well as the ability to provide the required number of data channels. The additional data channels required for ADAM are for processing 56 channels of data from the CREST seat which will be digitized, stored onboard, and also sent to a ground station by telemetry.

Anthropometry. It is believed that the ADAM anthropometry is a special feature of the manikin as it not only represents the new Tri-Service specification, but it has been developed from very detailed human data. The following discussion briefly describes the manner in which the data set was developed and how it was incorporated into the ADAM design.

The description of the ADAM is based on several sets of data. The Air Force 1967 survey of 2420 male flying personnel serves as the basis for the Tri-Service data set. This data set, which describes a small, mid-size, and large standard military aviator was developed using a growth factor (based on the 1980-1990 stature approximations), which was applied directly to the U.S. Air Force 1967 dimensions.

Table 3. Characteristics of instrumentation systems.

Manikin	Type	Telemetry	Onboard Recording	Umbilical	Number of Data Channels
GARD	Passive Digital	Yes	No	No	22
LRE	Computer Controlled (Partial)	Yes	Yes	Yes	96
ADAM	Computer Controlled (Complete)	Yes	Yes	Yes	128
HYBRID III	Passive Analog	No	No*	Yes	Variable (<30)

*The Naval Research and Development Center (NADC) developed an onboard data acquisition system for the Hybrid III (Reference 4).

The stereophotometric survey conducted in 1969 by the Texas Institute for Rehabilitation and Research of 31 male subjects was used to define the inertial tensors, weights, and skin contours of the ADAM body segments (Reference 5). Due to the complexity involved with building a statistical data base consisting of shapes, segments from single subjects were selected that best met the Tri-Service dimensional requirements, thus building a collection of segments from several subjects to describe a single ADAM skin envelope. This data base consists of the locations of the joint centers of rotation and the segment centers of gravity.

The data sets were initially defined with respect to a standard set of anatomical axis systems developed by AAMRL, which are used in comparing human subjects. These axis systems are based on points defined by bone protrusions on the surface of each body segment. The physical design of ADAM, however, requires the data to be related to the mechanical structures of the segments. To accomplish this, a set of mechanical axis systems, which are aligned with the mechanical axis of rotation and the long bone axes of each body segment, was defined.

A mathematical procedure was then developed to define the mechanical axis systems with respect to the standard anatomical axis systems. This procedure also performs the transformations required to describe the anatomical data base in the mechanical axis systems. This data base consists of the center of gravity locations, the joint centers of rotation, the skin shapes, and the principal moments of inertia of 17 segments for each of three sizes of manikins. With the exception of the principal moments of inertia, all data were transformed using a single displacement matrix for each segment relating the mechanical to the anatomical axis system. The rotational part of the displacement matrix for each segment was assumed to be equivalent for all three sizes; however, the translational part was dependent upon size. The inertial tensors, being located along the principal axes, were transformed into the anatomical axis systems and then into the mechanical axis systems.

Since the mechanical axis systems are easily located on the design drawings, the mass, mass moments of inertia, and center of gravity locations were then calculated for each segment and compared to the existing data base. If differences occurred, the designs were changed to comply with the data base or system specification. These axis systems also allow the user to independently define a segment by its mass properties with respect to the structure defining its kinematic properties without regard to the makeup of the entire manikin.

As stated previously, the surface shape data for each segment were taken directly from one subject rather than a statistical data base. In dealing with several subjects, the interfaces between segments were not continuous. Also, because each segment is taken directly from one subject, the shapes are not symmetrical and did not necessarily meet the dimensional requirements. Therefore, a technique which averaged each segment from right and left, fared the contours between segments, and scaled the shapes to meet the dimensional requirements was developed and used based upon this raw data base. The data base finally developed for the manikin conforms to the requirements of the specification and will result in manikins that represent statistically defined small, mid, and large size aviators. A companion paper presented at this meeting (reference 6) presents in detail the analytical procedures utilized to define the mass properties of ADAM in the various coordinate systems.

Flexible Spine

A drawing of the flexible spine that is being incorporated into the ADAM is shown in Figure 8, and the manner in which this flexible spine is incorporated in the overall ADAM design is shown in Figure 9. A linear spring/damper unit, similar to that of an automobile oleo strut, provides damping to impulsive inputs of the upper torso with respect to the pelvis. Yaw motions of the upper body with respect to the pelvis are provided by rotation of the outer tube with respect to the inner tube. Pitch and roll motions of the upper torso with respect to the pelvis are provided by the lumbar articulation mechanism. This final design evolved from an extensive series of analyses that considered not only an elastic spine, but also an elastic viscera mass suspension. Reference 7 presents a detailed description of the design evolution of the elastic spine. It was concluded that having an elastic spine in the vertical direction coupled to the nonlinear buttock spring, created by the skin/foam buttock, would provide an adequate simulation of the human response to impulsive loading in the vertical direction. Figure 10 presents a correlation of the acceleration measured on the chest of a human subject during a vertical impulsive loading test with that predicted for the manikin. The human experimental data was from Test 244 of the F-111 restraint tests presented in Reference 8. It can be seen that the correlation of the results is very good. The impedance characteristics with frequency of the ADAM system were shown to meet the desired characteristics as specified by the International Organization for Standardization (ISO).

Proof Testing of ADAM

Prior to the use of ADAM in a live ejection shot in support of the CREST development program, it is planned that ADAM will be subjected to a series of

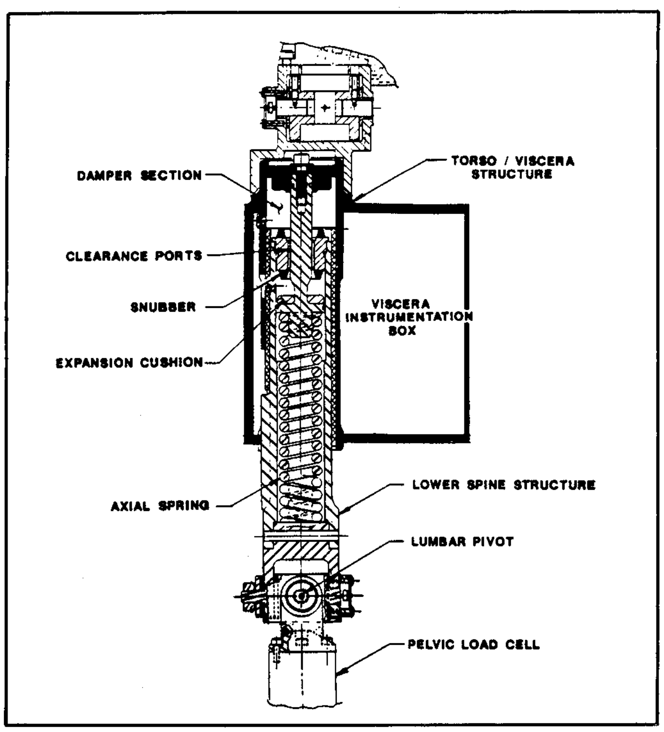

Figure 8. Elstic torso/spine system

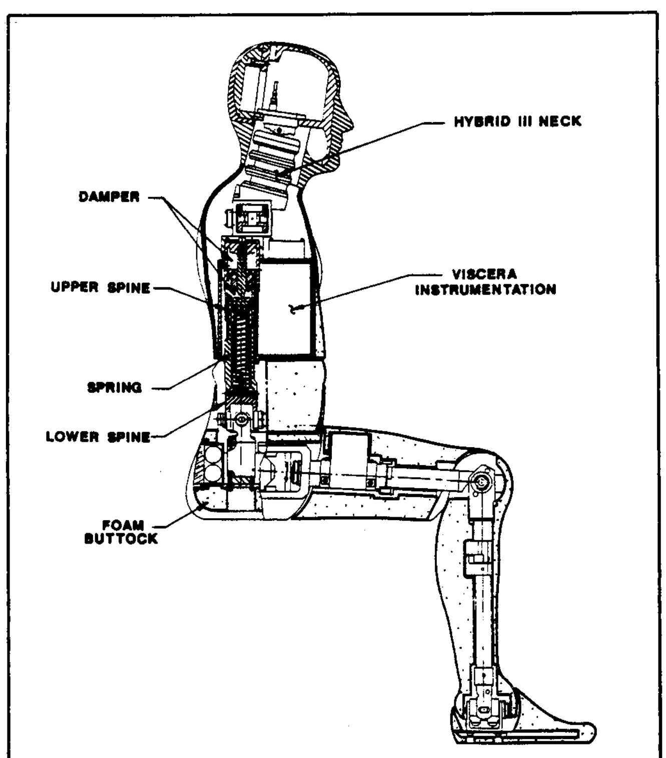

Figure 9. ADAM cross-section showing incorporation of the flexible spine system

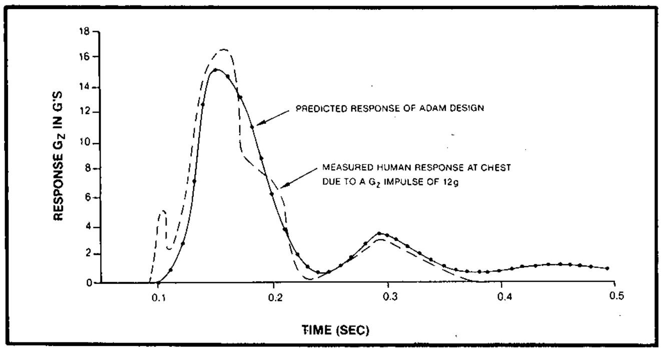

Figure 10. Comparison of predicted and measured response to Gz

rigorous tests to ensure that it performs to the specifications to which it was designed. The tests will duplicate in the laboratory, to the degree possible, the environmental conditions to which ADAM will be subjected during an ejection test at Holloman Air Force Base. The tests listed in Table 4 that are planned include vibration tests, impact tests, tests of mechanical and thermal durability, and windblast tests. All of the tests will be conducted at the facilities located in the AAMRL facilities at Wright-Patterson Air Force Base except the windblast tests which are planned for the facility at Dayton T. Brown.

While the proof tests outlined above are the most severe ever applied to a manikin, they will prove invaluable in that they will check out the complete system prior to an ejection test when a 100 percent reliability is not only desired but required.

Summary

This paper has tried to briefly summarize the need for ADAM and the stringent requirements it must meet if it is to be a successful test subject for the testing of advanced ejection seats such as CREST. It is believed that the desired goals can and will be met. At the present time, a large and a small prototype manikin are being fabricated, and the instrumentation system is undergoing component testing. It is anticipated that these two prototypes will be delivered to the Air Force during May 1987 for proof testing. The first use of the production manikin will be by the CREST program during September 1987.

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Table 4. Planned acceleration and impact durability tests.

Test Number	Type of Test	G Level	Pulse Shape	Duration	Orietation	Impact Velocity	Test Facility
1	Acceleration	45	1/2 Sine Wave	120 msec	±Gx	--	HIA
2	Acceleration	45	1/2 Sine Wave	120 msec	±Gy	--	HIA
3	Acceleration	45	1/2 Sine Wave	120 msec	±Gz	--	HIA
4	Acceleration	100	1/2 Sine Wave	6 msec	±Gx	--	HIA
5	Acceleration	100	1/2 Sine Wave	6 msec	±Gz	--	HIA
6	Acceleration	30	1/2 Sine Wave	20 msec	±Gx		VTD Para. Harness
7	Impact	--	--	--	--	40 ft/sec	VTD Concrete
8	Impact	--	--	--	--	40 ft/sec	VTD Concrete
9	Impact	--	--	--	--	40 ft/sec	VTD Concrete

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The Development of Segment Based Axis Systems for the Air Force Advanced Dynamic Anthropomorphic Manikin (ADAM) (Written only paper)

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Abstract

New advances in aircraft ejection seat technology have stimulated the development of an Advanced Dynamic Anthropomorphic Manikin (ADAM) that represents a male aviator population and possesses both static and dynamic human like characteristics. Systems Research Laboratories, Inc. (SRL), is presently under contract to the Armstrong Aerospace Medical Research laboratories (AAMRL), Wright-Patterson Air Force Base, to develop and fabricate these manikins using detailed specifications for the anthropometry, center of mass locations, and mass moments of inertia of each body segment. The data used in these designs have been initially defined with respect to a standard set of anatomically based axis systems developed from measurements on human subjects. These axis systems are based on anatomically defined points on the surface of each body segment. The physical design of ADAM, however, requires the data to be related to the mechanical structures of the segments. To accomplish this, mechanical axis systems have been defined for all body segments and transformations to the anatomical axis systems have been developed.

This paper summarizes the development of the mathematical procedure used to define and relate the mechanical axis systems with respect to the standard anatomical axis systems. The computer procedures used in performing the transformations of the anatomical data base in the mechanical axis systems are also described. Finally, the resulting mechanical axis systems developed for each of the ADAM sizes and the transformed data base are described.

Introduction

A human analog required for advanced dynamic testing, including the testing of an advanced ejection seat, must have a high level of biomechanical fidelity. The Advanced Dynamic Anthropomorphic Manikin (ADAM), as shown in Figure 1, is designed to have considerably improved biofidelity over currently available ejection system testing manikins; part of this biofidelity improvement depends on properly representing the anthropometric and mass properties of each individual segment. The human data used in this design have been specified for each segment with respect to a standard set of axis systems, designated as anatomical axis systems, which are defined by at least three points on the surface of the skin of each segment. These points were originally selected on the basis of palpability, within segment rigidity and maximum separation. The specific anatomical axis systems used for this data base were originally defined by McConville et al. (1980).

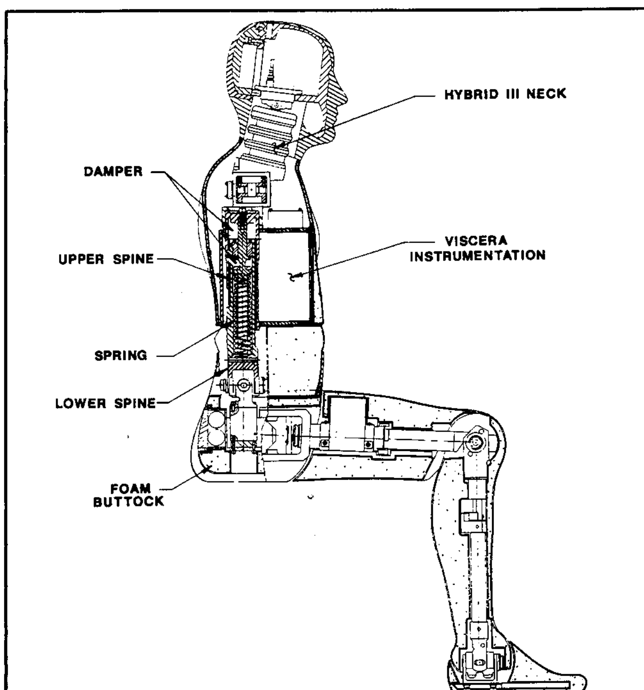

Figure 1. ADAM cross section

Data in this form can not be directly applied to the manikin design because the surface anatomical landmarks could not be directly related to the mechanical substructure necessary for fabrication. Therefore, before applying the data to ADAM, a transformation procedure was developed to obtain data in a form that could be directly applied to the design process. This transformation procedure first required the definition of a new axis system for each segment based on the mechanical elements of the segment. This report documents the definition of these axis systems, the procedure developed for transforming the data to these axis systems, and the development of the transformed data base used in the mechanical design. A brief description of the data base used for the description of the ADAM manikin and an overview of the axis systems used will be presented for background information.

Data Base Description

The specifications for the ADAM (reference ADAM Statement of Work) are based on several sets of data. The joint centers of rotation, segment centers of mass, and anthropometry were developed along with the Tri-Service data set (Proposed Military Handbook, 1986) from a common data base using similar methods. The ADAM data base was developed separately to define the joint centers and segment centers of mass in three dimensional space as the Tri-Service data are described in only two dimensions. This Tri-Service data set, which describes a small, mid-size, and large standard military aviator, was developed using a growth factor (based on the 1980-1990 stature approximations) applied directly to the dimensions from the U.S. Air Force 1967 survey of 2,420 male flying personnel (Churchill, 1978).

The results from the stereophotometric survey conducted in 1969 by the Texas Institute for Rehabilitation and Research of 31 male subjects (McConville, 1980) were used to define the inertial tensors, weights, and skin contours of the ADAM body segments. The inertial properties were derived from the volume distribution data of the survey data base; however, due to the complexity involved with building a statistical data base describing shapes, segments from single subjects were selected that best met the Tri-Service dimensional requirements, thus creating a collection of segments from several subjects to describe a single ADAM skin envelope.

Four axis systems are used in the transformations of the original ADAM data sets. Three of these are segment based and the other is a total body or global system in which the relative segment positions are fixed. The particular global system used in this analysis is defined in the stereophotometric data base and the surface data were initially compiled in this

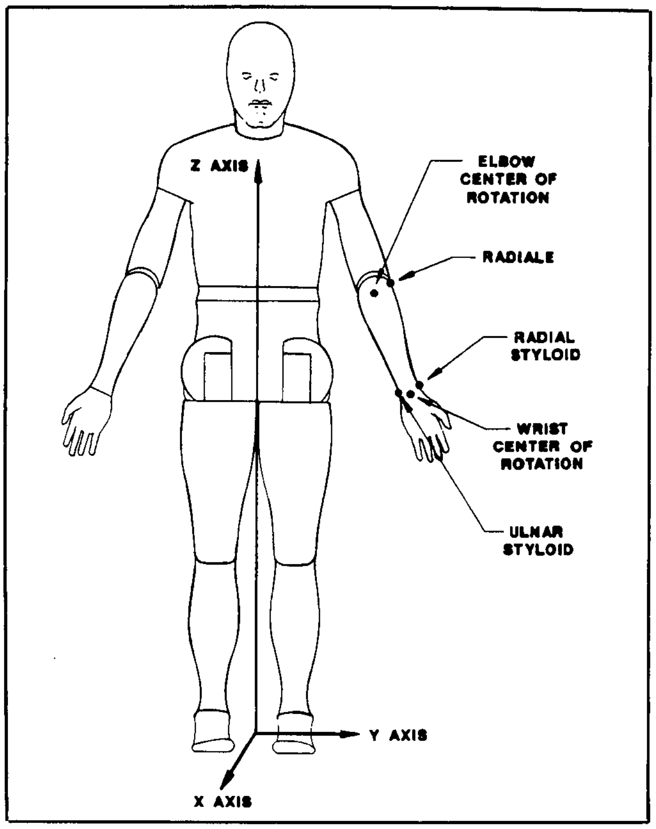

Figure 2a. Total body global axis system

axis system. These data, which consisted of body surface point coordinates and the coordinates of surface landmarks, were combined with segmentation planes as defined by McConville to create individual segment surface data sets. Body segment orientations for the 31 subjects measured in the data base were averaged to arrive at a composite body position which has been used for relative adjacent segment position and motion analyses. The reconstructed body position for ADAM, in the global coordinate system and with lower arm landmarks illustrated, is shown in Figure 2a.

The anatomical, principal, and mechanical axis systems are based on individual segment properties. As described earlier, the anatomical axis systems are based on defined points on the skin surface. Both the Tri-Service and stereophotometric data bases are de-

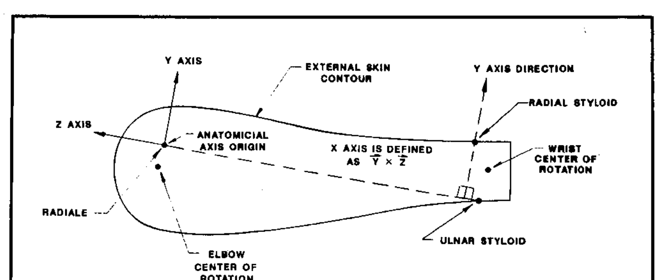

Figure 2b. Forearm anatomical axis system

fined with respect to the anatomical axis systems. The principal axis systems are derived from the segment mass distribution properties with certain approximations to aid practical implementation. They are specified with respect to the segment center of mass and are offset from the anatomical axes by a y axis rotation. The mechanical axis systems are based on the mechanical substructures within each segment and are developed for use in the design of the manikin. Figures 2b-d show the anatomical, principal, and mechanical axis systems, respectively, associated with the left forearm.

Mechanical Axes Definition

The transformation of segment data from an anatomical to a mechanical axis system requires the calculation of the displacement matrix which relates the two axis systems. The displacement matrices for the ADAM segments were calculated from the mathematical definition of the mechanical axes with respect to the anatomical axis systems. The definition procedure was dependent on the type of body segment under consideration.

The body segments were separated into three general groups based on segment geometry and type of connective joints for axis definition. The three groups were the torso segments, the limb segments, and the extremities. The mechanical axis systems were defined as described below.

Torso Segments

The torso inertial properties were initially specified with respect to three segments: the upper torso, the abdomen, and the pelvis. This implied that two discrete articulations were associated with torso deformation. The ADAM design includes one articulation point in the torso which creates two torso segments: the upper and lower torso, the plane of separation being normal to the plane of symmetry (midsagittal plane) and located at the lumbar pivot. Since the manikin torso plane of separation did not correspond to either of the planes defined in the human data, a direct application of the manikin to human torso segment data was not valid. To allow a direct comparison, the human data have been redefined to place the single plane of separation at the same location as the lumbar pivot point of the mechanical system. The human data for the upper torso were then defined with respect to the thorax anatomical axis system and the lower torso with respect to the pelvis anatomical axis system.

The mechanical axis systems for both the upper and lower torso, as shown in Figures 3 and 4, are defined with respect to the physical characteristics of the manikin. The y axis of each is normal to the midsagittal plane and is positive to the left. The z axis of the upper torso is defined by the vector from the intersection of the mechanical spine center line with the global x-y plane which passes through the shoulder centers of rotation to the lumbar pivot point. The z axis of the lower torso is defined by the vector which originates at the lumbar spine when the manikin is in the sitting position (the lumbar pivot in the 0 degrees position). The origin of the upper torso mechanical axis system is located along the spine center line at the height of the shoulder centers of rotation. The lower torso mechanical system origin is located at the lumbar pivot center of rotation.

Limb Segments

The second group of segments is that for limbs. This group includes the upper and lower arms and the upper and lower legs. The mechanical axis origins of these segments are defined at the proximal joint center for each segment, with the z axis extending from the origin to the distal joint center. The joint center locations for each segment are defined with respect to the corresponding anatomical axis systems in the human data base. The y axes for the forearm and the lower leg were chosen to be aligned with the pin rotational axes of the elbow and knee, respectively. Since the hip and shoulder joints have more than one degree of freedom, either axis of rotation can be used to define the second mechanical axis for the upper leg and arm, assuming the two axes are normal to each other. The axis which allows abduction/adduction for each joint was chosen to define the mechanical x axis direction based on the procedure outlined below.

A cross product method was used to define the orientations of the joint rotational axes. The method

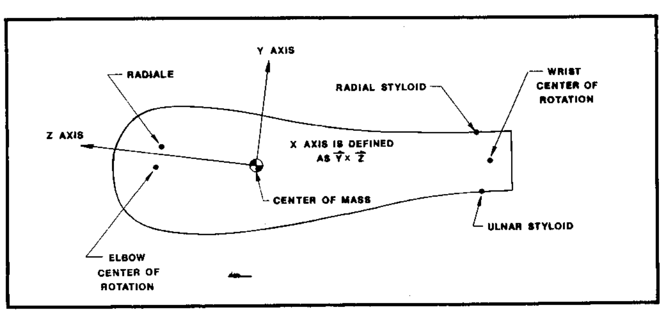

Figure 2c. Forearm principal axis system

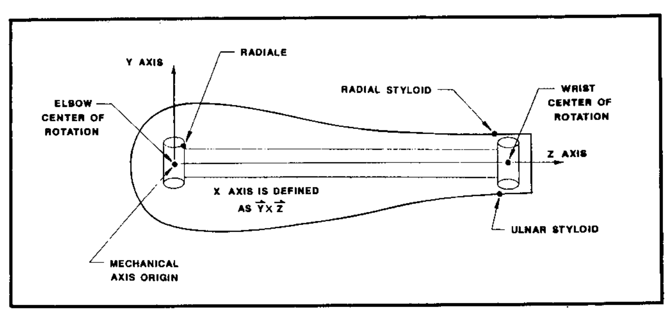

Figure 2d. Forearm mechanical axis system

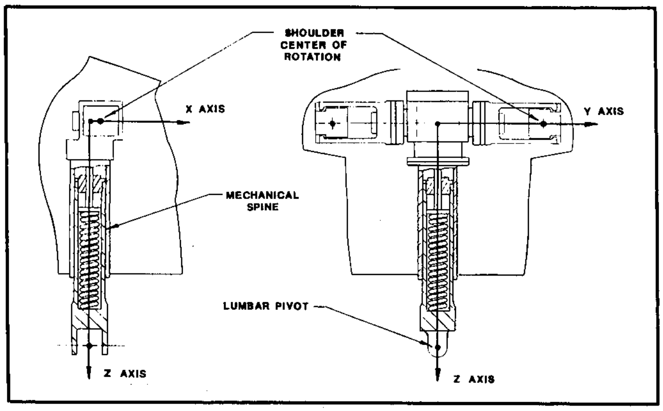

Figure 3. Upper torso mechanical axis system

is based on the fact that the two bones of a pinned joint move such that the center lines of the bones and the pinned joint center lie in the same plane throughout the range of motion. The axis of rotation of the pinned joint is normal to this plane and is taken as the rotational x or y axis. This axis direction was calculated by taking the vector cross product of the mechanical z axes of the adjacent limb segments in the global system with the body segments in the average composite orientations described previously.

Extremities

The third group of segments consists of the extremities. Included in this group are the head, neck, hands and feet. All of the skin contours except the feet and several mechanical elements of these segments used in the ADAM are standard parts from existing manikins. These segments generally met the dimensional properties however some changes were required to meet the inertial properties required in the ADAM. To make these changes, mechanical axis systems were required to relate the actual data to the human data.

The skin drawings of the existing parts with the ADAM mechanical elements superimposed on them were used to relate the anatomical and mechanical axes. By locating the landmarks on the drawings, the anatomical axis systems associated with the segments were identified.

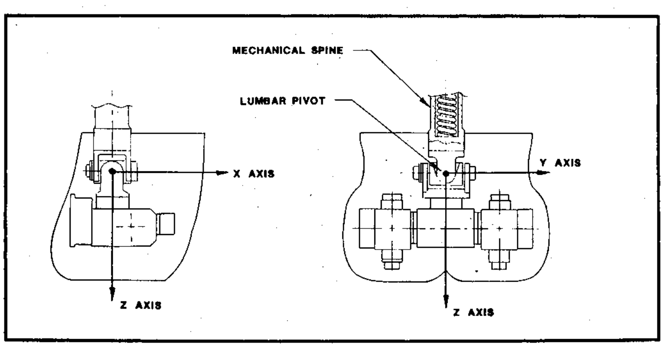

Figure 4. Lower torso mechanical axis system

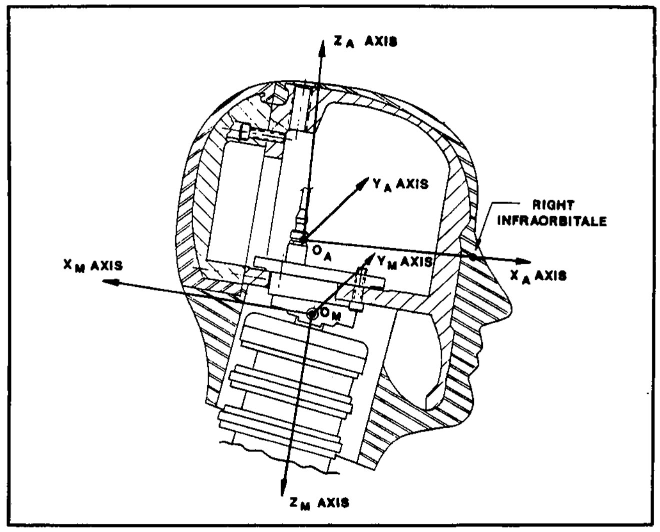

Figure 5. Head mechanical axis system definition

The mechanical axis system definitions for these segments are defined based on the joint centers of rotation and the mechanical elements within the segments. The axis systems shown in Figures 5, 6, and 7, will be briefly described. The head mechanical axis system origin, as shown in Figure 5, is located at the pin which joins the head and the neck. The x axis is the vector from the origin in the posterior direction parallel to the bottom plate of the head in the midsagittal plane. The head z axis is the vector from the origin in the midsagittal plane normal to the x axis. The neck axis origin, as shown in Figure 6, is also at the pin which attaches the head and neck

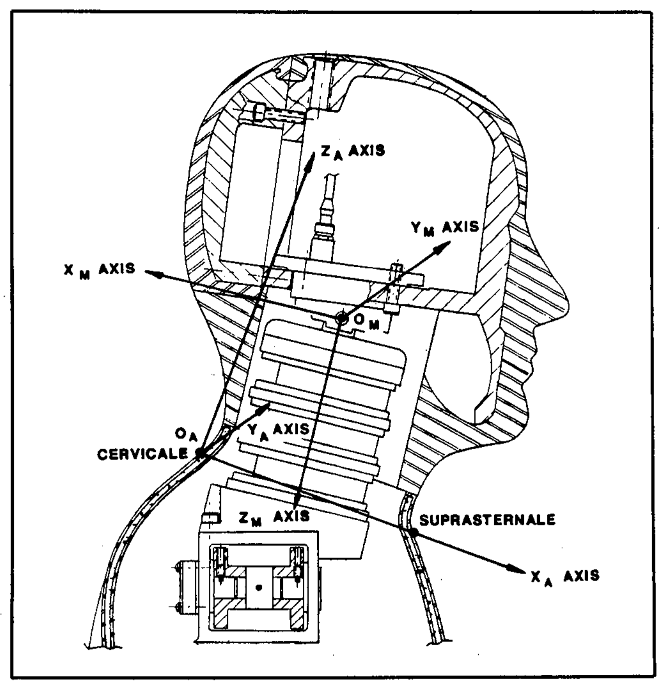

Figure 6. Neck mechanical axis system definition

segments. The z axis is chosen as the vector from the origin to the center of the bottom bolt. The x axis is the vector from the origin in the midsagittal plane normal to the z axis. The hand mechanical axis system origin, as shown in Figure 7, is located at the wrist center of rotation. The x axis is chosen to be aligned with the wrist rotational axis for inversion/eversion. The z axis is defined as the vector from the origin to a point located at the center of the hand bone.

The mechanical and antomical y axes for these segments were taken to be parallel and the translations and rotations between the two axis systems were explicitly defined on the drawings.

The foot mechanical axis system as shown in Figure 8 is defined by the mechanical elements within the segment and the known position of the foot in the global axis system. The origin is located at the ankle center of rotation and the z axis is defined as the vector extending from the origin normal to the global x-y plane. Because of the orientation of the foot in the stereophotometric data base which extended directly forward, the axis of rotation of the ankle is assumed to be parallel to the global y axis. This axis is then taken to be the y mechanical axis.

Data Transformation

The methods described for defining the mechanical axis origins and the z and y axis directions were applied to each manikin segment. The x axis directions were calculated using the cross product of the y and z axes. A summary of the mechanical axis definition schemes is presented in Table 1.

The three points used to define the mechanical system origin and axes orientations are located with respect to the anatomical axis system by the methods described. These points are used in a computer procedure which calculates the transformation matrix from the anatomical to the mechanical axis system, $[D]_{MA}$. This procedure involves the establishment of the unit vectors along each axis and the formation of the rotation matrix. The translation vector is then found and combined with the rotation matrix to generate a 4 × 4 transformation matrix.

After the displacement matrices were calculated, the data were transformed from the anatomical to the mechanical axes by the operation:

$$\overline{P}_M = [D]_{MA}\,\overline{P}_A$$

where

$\overline{P}_M$	= vector in the mechanical axis system
$[D]_{MA}$	= the displacement matrix (4 × 4) from the anatomical to the mechanical axes
$\overline{P}_A$	= vector in the anatomical axis system

The data that were directly transformed consisted of the center of gravity locations and the joint centers

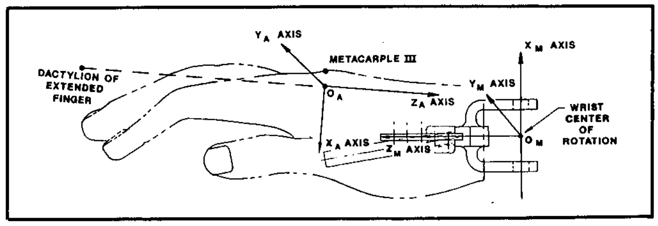

Figure 7. Hand anatomical and mechanical axis systems

of rotation. The inertial and surface shape data transformations are more involved and will be discussed in the following sections.

Inertial Transformation

The intial inertial property data provided for design consisted of principal axes and moments of inertia specified with respect to the anatomical axis system and center of mass origin. In these data, the principal axes were displaced from the anatomical axes only by a rotation about the y-anatomical axis through an angle θ. The transformation operator from the principal to the antomical axis is:

$$[A]_{AP} = \begin{bmatrix} \cos\theta & 0 & -\sin\theta \\ 0 & 1 & 0 \\ \sin\theta & 0 & \cos\theta \end{bmatrix}$$

The principal moments were transformed into the anatomical axis system by

$$[I]_A = [A]_{AP}\,[I]_P\,[A]_{AP}^T$$

where

$[I]_A$	= inertia tensor about the anatomical axes
$[I]_p$	= principal moment of inertia tensor
$[A]_{AP}^T$	= transpose of $[A]_{AP}$ matrix

The inertia tensor was then transformed into mechanical axis system alignment but with the origin still at the segment center of mass, by

$$[I]_M = [A]_{MA}\,[I]_A\,[A]_{MA}^T$$

where

$[I]_M$	= inertia tensor about the mechanical axis
$[A]_{MA}$	= transformation operator from the anatomical to the mechanical axes

Products of inertia are developed in the transformation procedure; however, the typical product of inertia in the mechanical axis system is less than 10 percent of the Ixx and Iyy moments of inertia. A sensitivity analysis showed that neglecting the products of inertia in the mechanical axis system affects the principal moments of inertia about the x and y axes of all

Table 1. Summary of mechanical axis definition.

Segment	Mechanical Z Axis is the Vector From	To	Y Axis Definition	X Axis Definition
Head	Head/Neck Pin		Midsagittal Symmetry	Cross Product of y and z
Neck	Head/Neck Pin	Bottom Center of Neck	Midsagittal Symmetry	Cross Product of y and z
Upper Torso	Point At Shoulder Along Spine Center Line	Lumbar Pivot Point	Midsagittal Symmetry	Cross Product of y and z
Lower Torso	Lumbar Pivot Point	Along Spine Center Line	Midsagittal Symmetry	Cross Product of y and z
Upper Arm	Shoulder	Elbow	Cross Product of z and x	Shoulder Abduction/ Adduction Axis of Rotation
Forearm	Elbow	Wrist	Elbow Axis of Rotation	Cross Product of y and z
Hand	Wrist	Center Point of Distal End of Hand Bone	Wrist Flexion/ Extension Axis of Rotation	Cross Product of y and z
Thigh	Hip	Knee	Cross Product of z and x	Hip Abduction/ Adduction Axis of Rotation
Calf	Knee	Ankle	Knee Axis of Rotation	Cross Product of y and z
Foot	Ankle	Bottom of Foot Bone	Ankle Flexion/ Extension Axis of Rotation	Cross Product of y and z

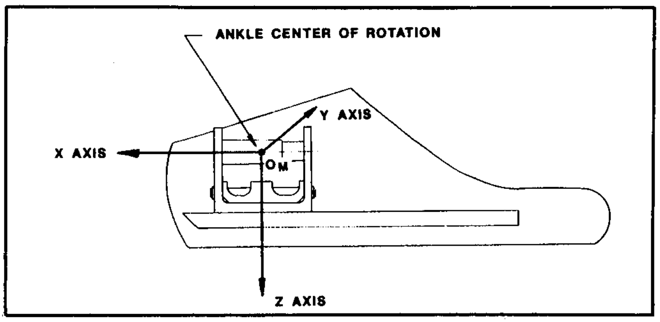

Figure 8. Foot mechanical axis system

segments and the z axes of the torso segments by less than 4 percent. Because of the small displacements of the limb skin contours in both the x and y directions from the z axis, the Izz are small compared to the Ixx and Iyy and the products of inertia are large when compared to the Izz of the limbs. However, all products of inertia have been neglected based on the assumption that only the Izz of the upper and lower torso have a significant effect on system response.

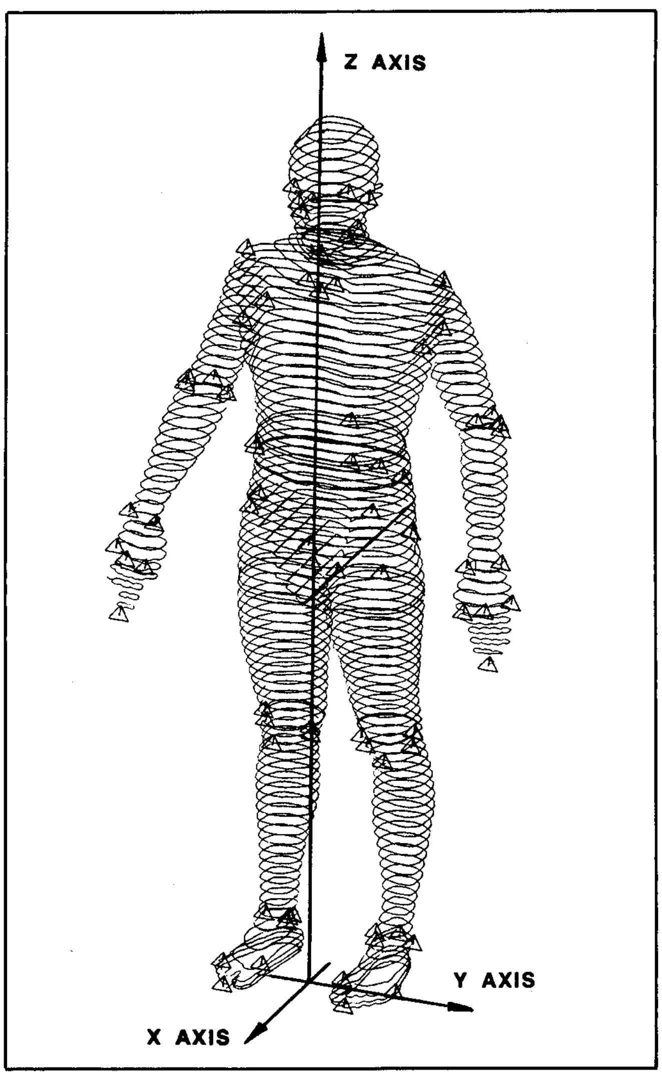

Figure 9. Stereophotometric surface data (subject #3)

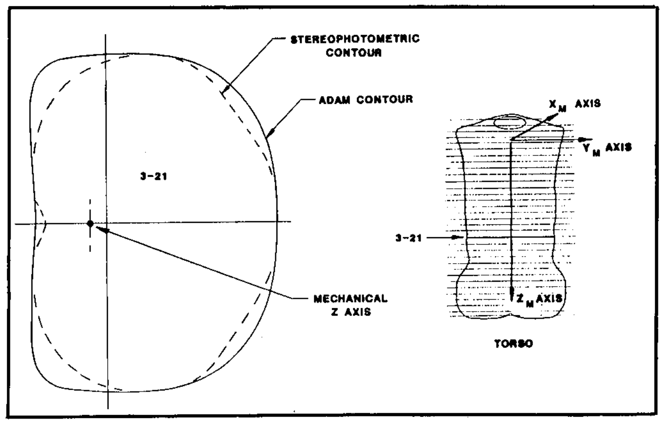

Figure 10. Surface shape definition

Surface Shape Definition

The skin surface of the ADAM is important for proper aerodynamic response, interfacing with restraint harnesses and various seat configurations, and mounting of flight equipment. To develop the skin contours for each ADAM, measured skin surface data points from the stereophotometric data base were used. The data points consisted of body surface point coordinates organized in horizontal or x-y plane cross-sectional slices at two centimeter intervals. A typical data set, including surface landmarks, from one subject is shown in Figure 9. To create a body skin contour that met the dimensional requirements for the ADAM, individual segments were selected based on these requirements from the 31 subjects.

The segment mechanical z axis was defined on each slice and the location of the slice along the z axis was calculated. The data were then graphically related to the mechanical axis system (Figure 10).

After relating the data to the mechanical axis system, three additional operations were performed on the graphical data. First, segment attachment slices were altered to create a smooth contour. Because adjacent segment data did not necessarily originate from the same subject, the data were not necessarily continuous at the segmentation planes and had to be refined. The second operation was to make the cross sections symmetrical through an averaging technique. The torso segment slices were made symmetrical about the midsagittal plane by overlaying the right and left contours and using the center line between the two to create the contour. For the limb segments, an analysis of the data showed that the line of symmetry necessary to create single right/left segments for the manikin existed in most slices. The limb data were then averaged using the same method as the torso. In the third operation, the profiles in the x-z and y-z planes were scaled to match the Tri-Service dimensional data. The slices along each segment were not necessarily scaled by the same factors due to the dimensional requirements for each segment. The finalized cross sections were then made into templates to form molds for skin production.

Conclusions

Existing manikin technology has used a generalized data base to define human body properties including only segment lengths, masses and centers of mass. In order to design a manikin which possesses a specific representation of the surface shape and inertial properties of a human population, a quantitative relationship between the manikin and human properties had to be developed.

While the transformations developed here are specific to the ADAM design, the procedures apply to any manikin development in which a one-to-one correspondence between human and manikin response is sought.

By providing these transformations that relate the mechanical structure of the manikin to an equivalent human anatomical structure, manikin performance can be accurately evaluated with regard to human response under similar conditions. Additionally, with the increasing use of occupant motion simulation computer programs, the application of manikin structures to the human data base allows for the evaluation of the effects of design variations on manikin biofidelity.

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Additional Data Sources

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Proposed Military Handbook, 1986. The Standard Male Aviator: Small, Mid-Size, and Large, unpublished data.

An Overview of Existing Sensors for the Hybrid III Anthropomorphic Dummy

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Abstract

The requirements of current automotive testing demand more data than previously afforded by the Hybrid III and Part 572 anthropomorphic dummies.

Denton, Inc. has developed a series of multi-component transducers in an effort to provide more complete answers to complex questions of force measurement in controlled and uncontrolled impact testing. In some instances the multi-axis transducers work with other previously developed sensors; in other instances they replace the earlier instruments as bolt-in components that are easily removed for calibration.

The specific areas for which multi-component sensors have been produced are: the base of the skull, the lower neck, the thoracic spine, the lumbar spine, the straight spine, the pelvis, the femur, and the lower leg (knee, tibia, ankle-foot assembly).

Introduction

In order to make best use of the research dollar, the project engineer must accumulate the most information possible from each test performed. As more engineers identify the areas where they require additional information, we at Denton, Inc. are receiving frequent requests for special transducers to be developed that have already been produced for other laboratories. Therefore, a brief overview of the trans-

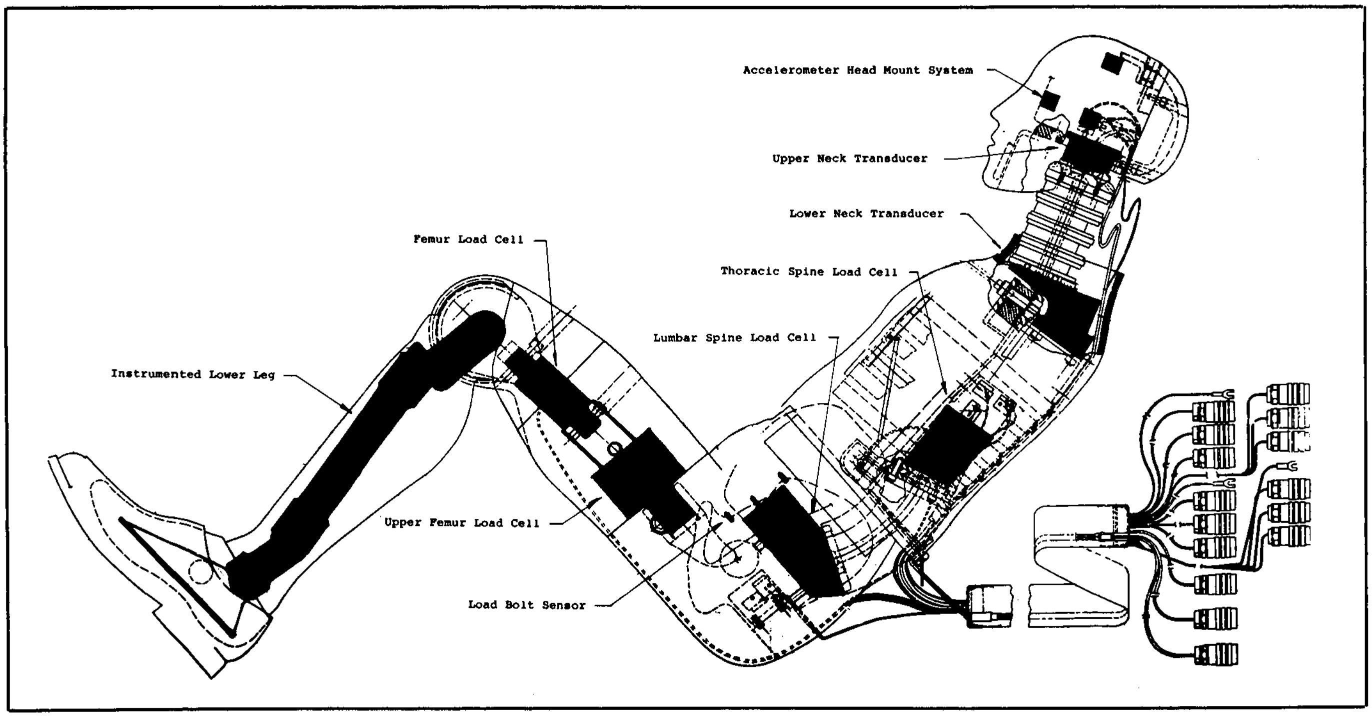

Figure 1. Overall drawing of Hybrid III dummy

ducers already available for the Hybrid III Dummy is necessary to help eliminate the redundant effort of identifying areas of needed measurement and force capacities that a project engineer must address.

In developing the Hybrid III transducers, an effort has been made to make these sensors bolt-in replacements for existing structures within the dummy. Whenever possible, the transducer has been designed to maintain the weight and cg of the structure it is replacing. In areas where the weight and cg have not been conserved, the manufacturers of the dummies have been able to make the necessary adjustments.

Skull

Figure 1 shows a head to toe drawing of the Hybrid III Dummy. The shaded areas represent the structures for which we already provide a standard line of transducers. In addition to the transducers, an accelerometer mounting system for the head is also shown in Figure 2. This accurate modification is made to the dummy skull to facilitate the mounting of nine accelerometers. The modification also provides the maximum distance between the accelerometers in the skull. This system is used as one method of measuring angular acceleration. Several methods for computation of angular acceleration exist[1].* We provide this modification only for the placement of the accelerometers themselves; the method and accuracy of computation is left to the user.

* Numbers in brackets designate References at end of paper.

Neck

The first transducer shown in Figure 1 is an upper neck transducer. This transducer, also shown in Figure 3, is a Model 1716 and measures the forces along the three orthogonal axes and the moments about these axes. Previously this area of measurement was covered by a three component transducer that did not provide sufficient data when the impact was an oblique or pure side impact. The Model 1716 Neck Load Cell provides the engineer with the complete data needed for analysis of an impact of uncontrolled and unknown forces. The manufacturers of the Hybrid III Dummy have provided a skull with the necessary mounting modifications to accept the Model 1716 Neck Load Cell. There is also a special tool available that bolts to the back of the skull. This tool is designed to compress the neck and aid in the assembly/disassembly of the dummy neck to the Model 1716.

Note: the three-axis neck load cell is still available. Therefore, when ordering a new dummy without a Model 1716 six-axis transducer installed, the engineer should specify with which skull the dummy should be equipped, dependent upon which neck transducer he anticipates using.

In order to investigate the mechanism of injury to the neck it became necessary to measure forces at the base of the neck. A transducer, Model 1794 (Figure 4), was fabricated to replace the neck mounting bracket, G.M. drawing 78051-303. The bracket is

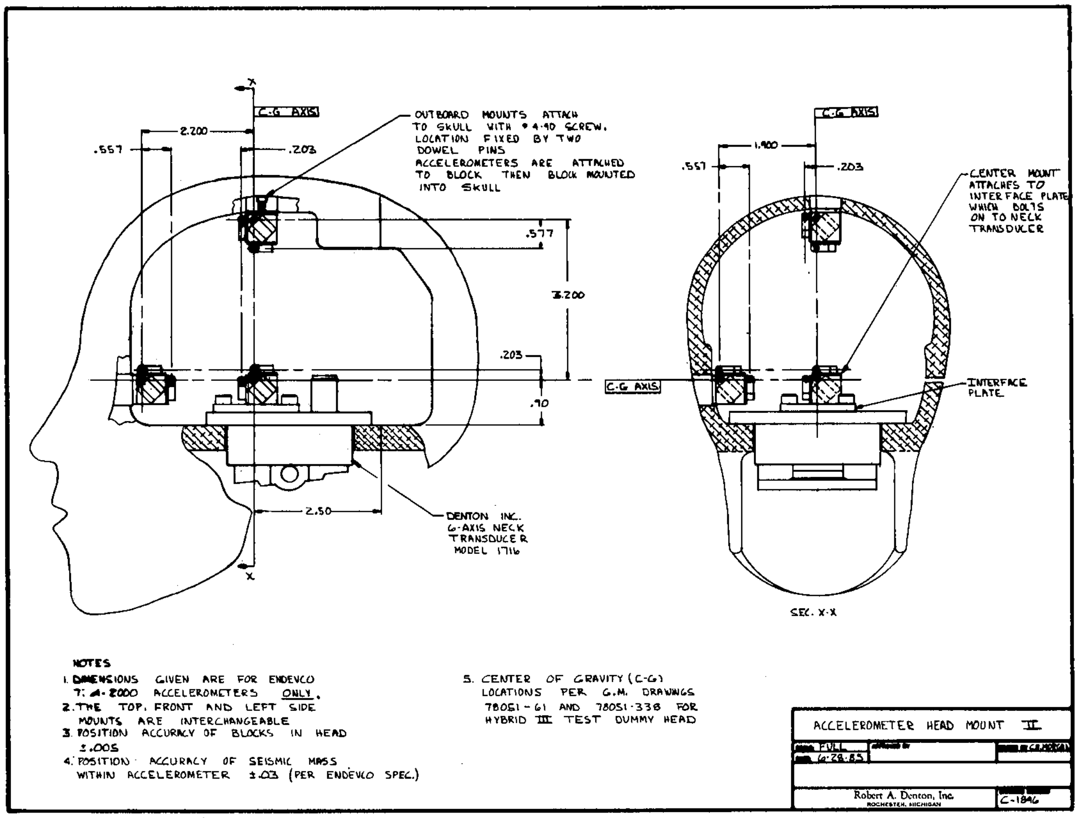

Figure 2. Accelerometer head mount system

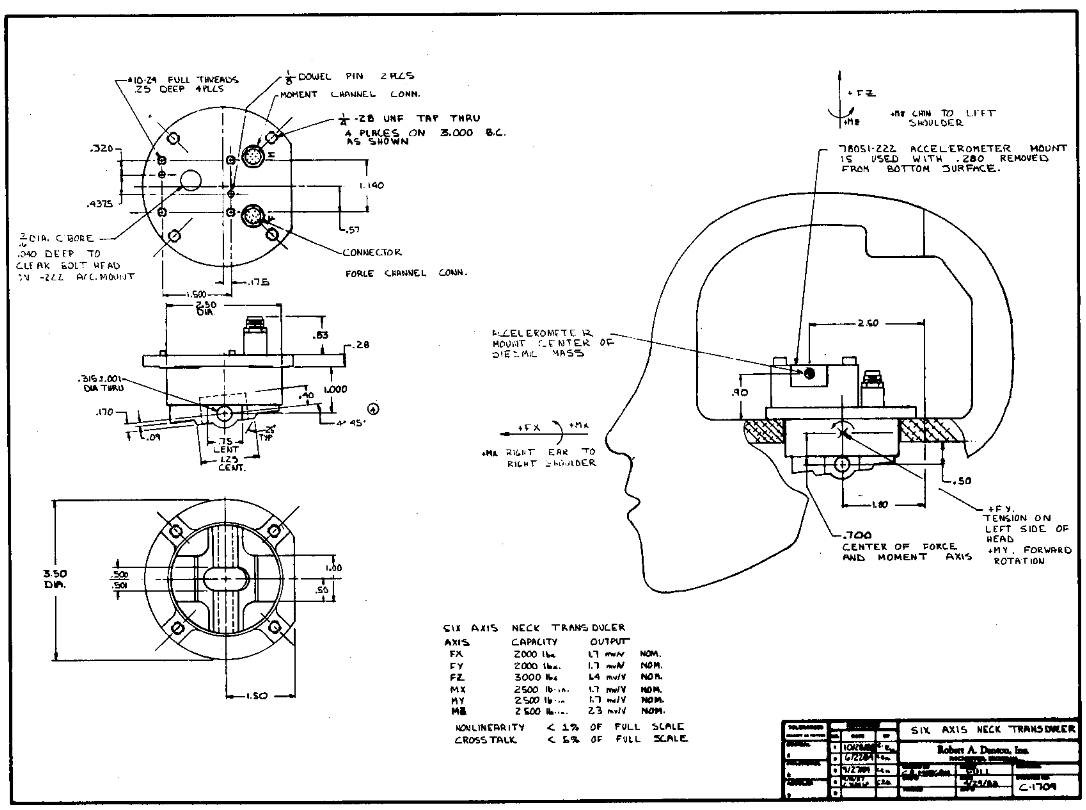

Figure 3. Upper neck transducer model 1716

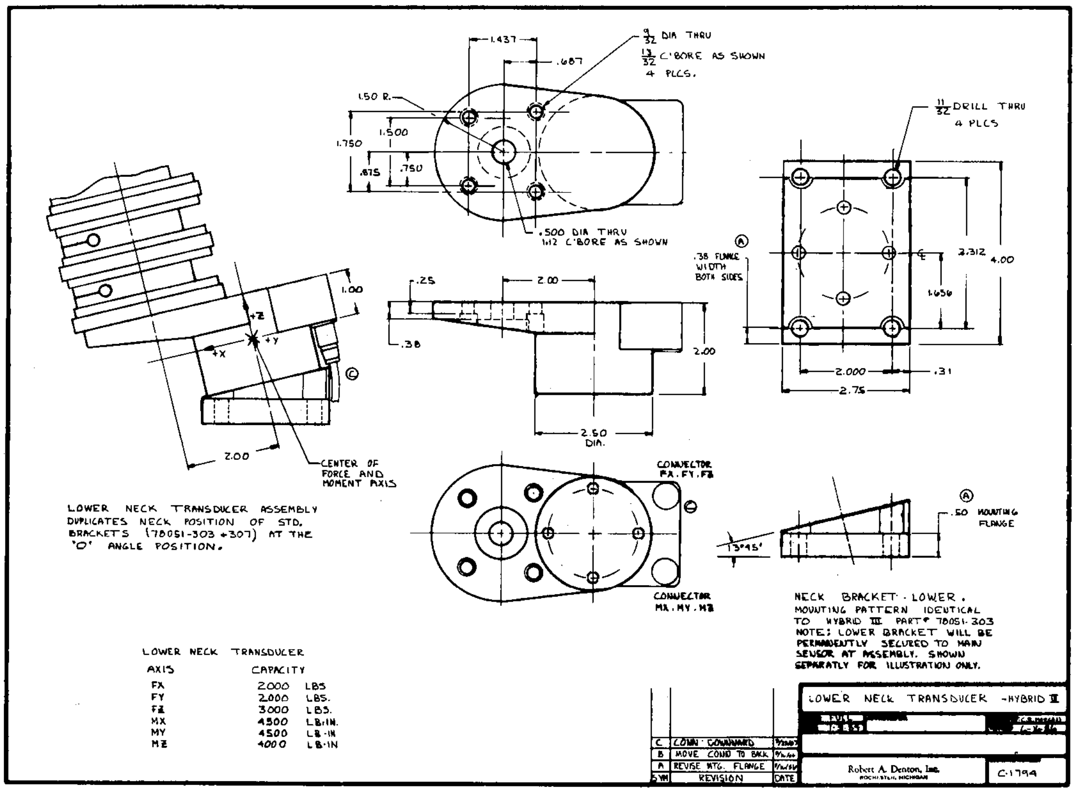

Figure 4. Lower neck transducer model 1794

normally adjustable in two degree increments and although the transducer is a bolt-in replacement for the neck bracket, the angle adjustment had to be sacrificed. The 0 angle position was chosen as the most utilized position for the neck. The Model 1794 Lower Neck Transducer is a six component load cell with the axes parallel and perpendicular to the axis of the neck.

Spine

Figure 5 is the Thoracic Spine Load Cell. This transducer measures the three orthogonal forces and the moments about the fore-aft (x) axis and the lateral (y) axis. The mounting requires modification to the existing thoracic spine, G.M. drawing number 78051-179. Once these modifications are made, the transducer bolts to the modified parts and can be mounted in the dummy.

The Lumbar Spine Load Cell, Model 1842 shown in Figure 6 is a three component load cell. The angular mounting requirement of the curved spine of the Hybrid III limited the space available for a transducer. The mounting constraints restricted this transducer to measurements of the fore-aft force (Fx), the axial force (Fz), and the moment about the lateral axis (My).

In aircraft testing the Hybrid III Dummy has been developed with a straight lumbar spine. To accommodate these users the three-axis 1842 transducer has been modified to accept the straight spine. This transducer is a Model 1891, shown in Figure 7. The straight spine configuration also allows for the use of Model 1708 (Figure 8), a six-axis lumbar spine load cell that was developed as a bolt-in transducer for the Part 572 Dummy.

Pelvis

A system for detecting pelvis movement relative to the seat belt was developed by R.P. Daniel[2]. This system utilizes a series of six load bolts in a modified pelvis casting. The Load Bolt Sensor, Figure 9, used in this system is a uniaxial load cell. The purpose of each load bolt is to sense when the lap belt is at its position, much as a contact switch functions. Each sensor can measure loads up to 1000 lbs.

Femur

Two upper femur load cells have been developed that replace the upper femur structure of the Hybrid III. The Model 1830 shown in Figure 10 is a three component sensor that measures shear force fore-aft (Fx), the bending moment about the fore-aft axis (Mx), and the bending moment about the lateral axis (My). This sensor can be rotated ninety degrees so that the fore-aft axis will measure the lateral shear force (Fy) along with the two bending moments. The

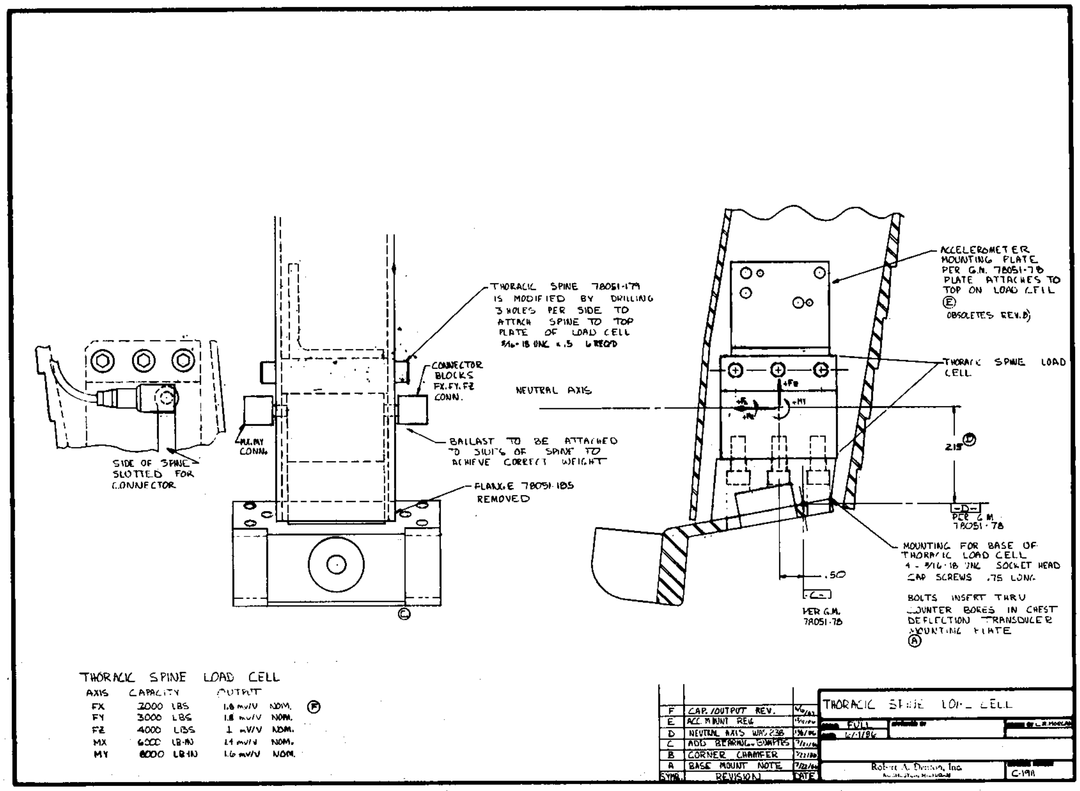

Figure 5. Thoracic spine load cell

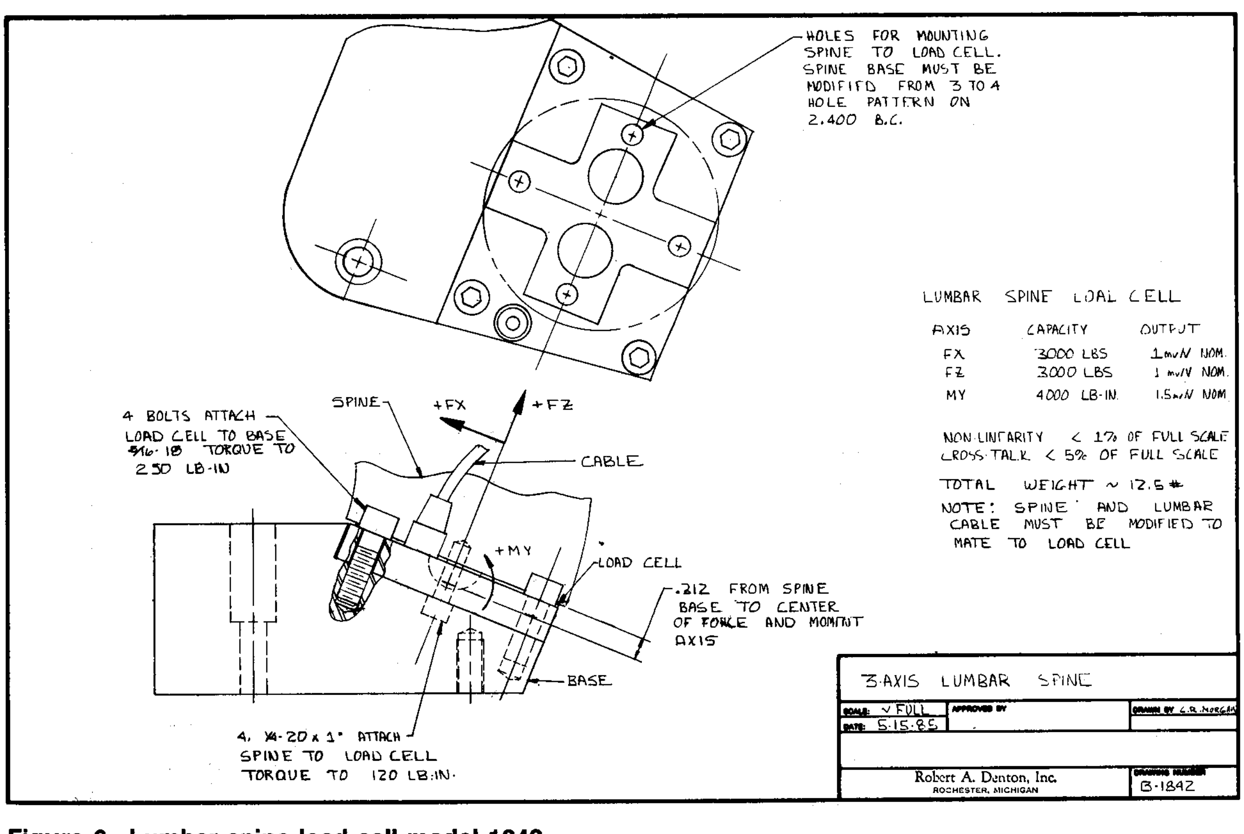

Figure 6. Lumbar spine load cell model 1842

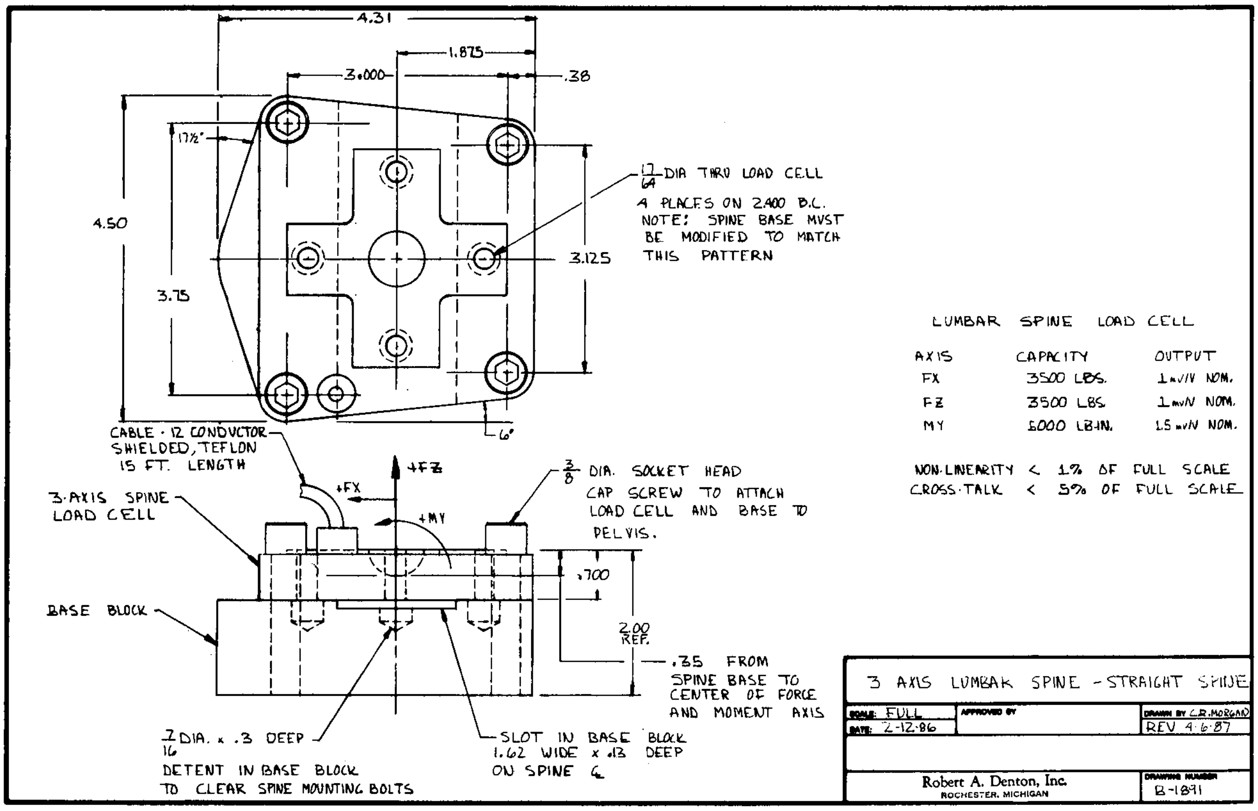

Figure 7. Straight spine load cell model 1891

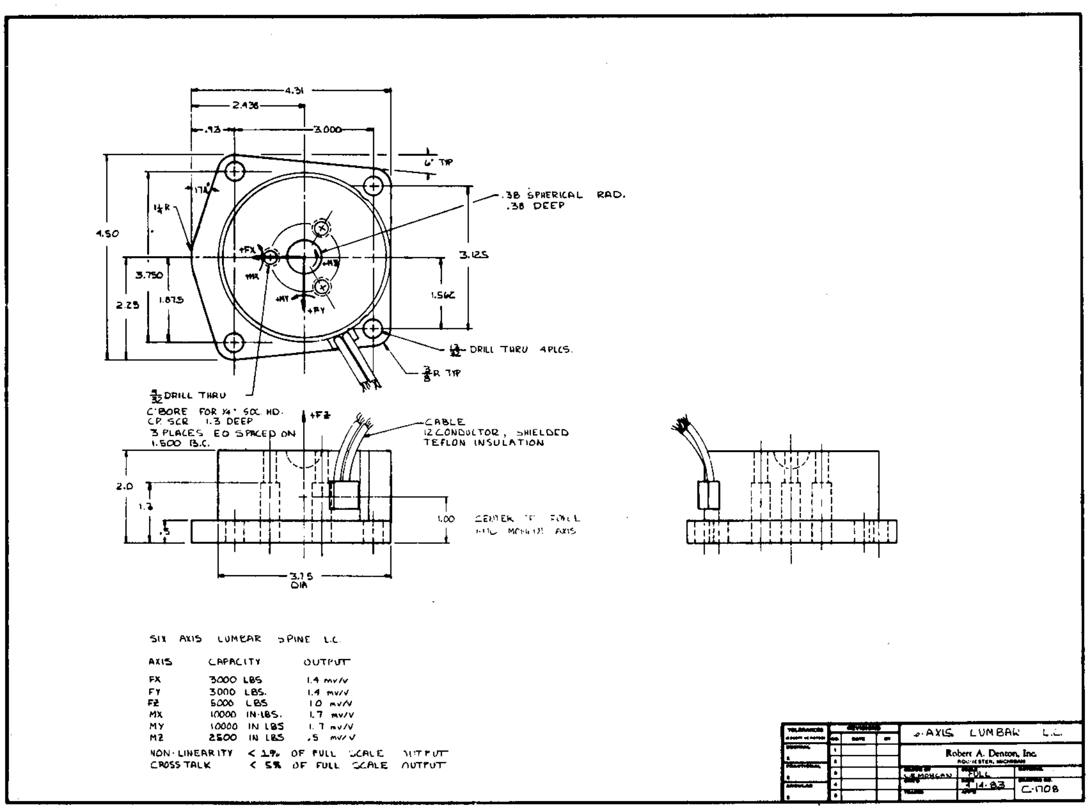

Figure 8. Lumbar spine load cell model 1708

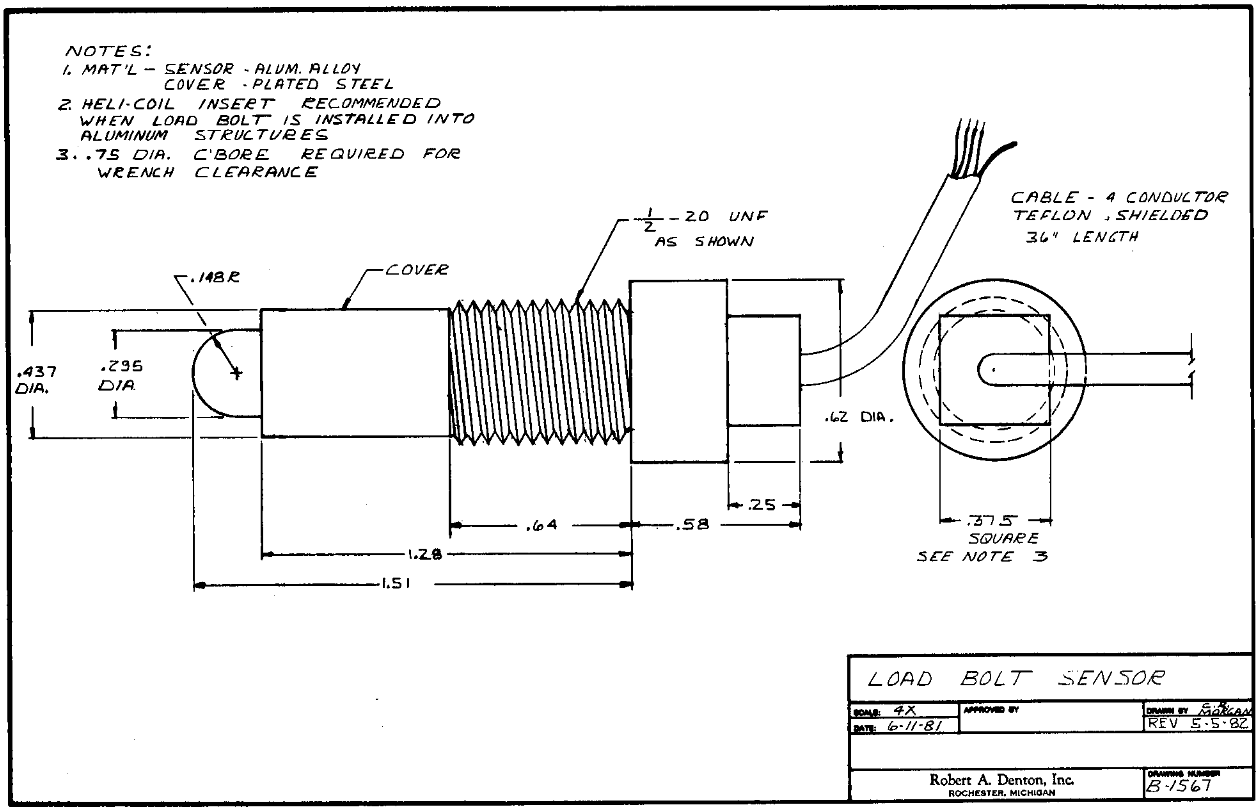

Figure 9. Load bolt sensor

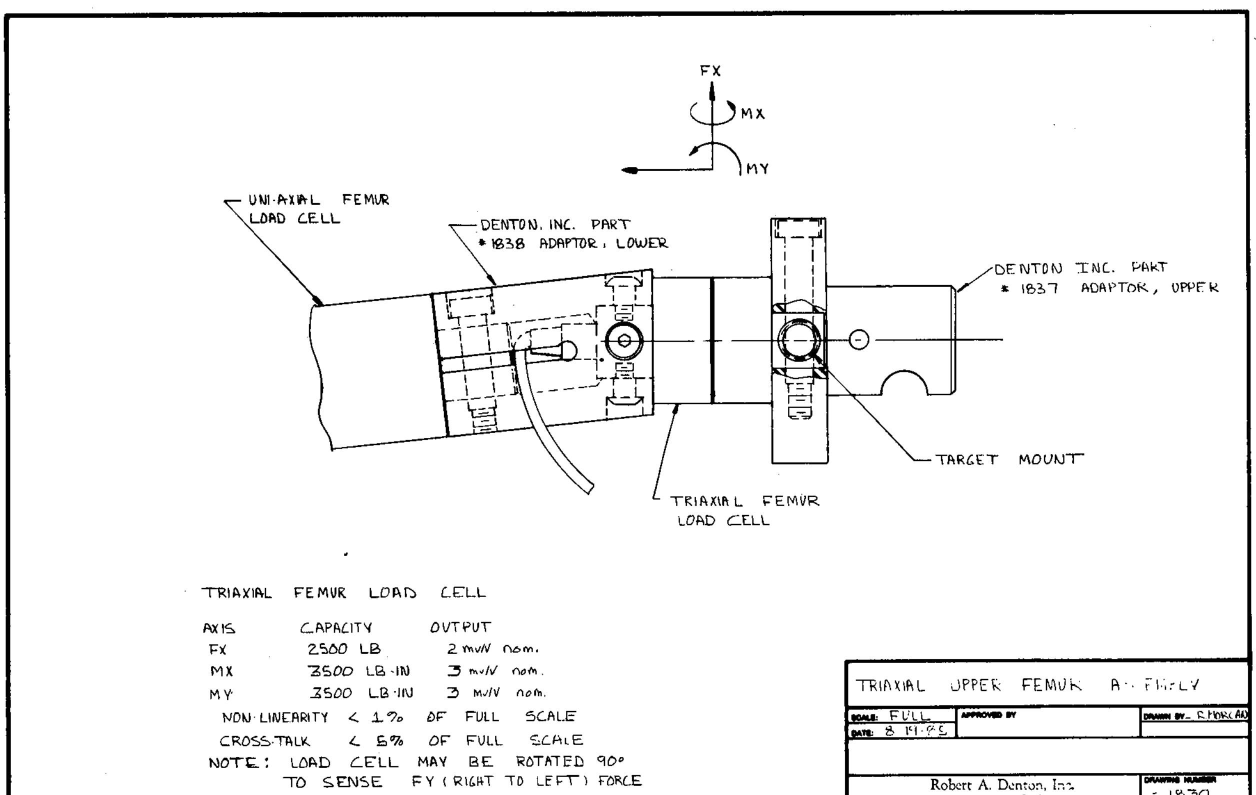

Figure 10. Upper femur load cell model 1830

sensor provides sufficient data when the impact is controlled.

When the test involves oblique impacts or unrestrained testing with unknown inputs, the Model 1974 (Figure 11) provides five channels of data. The Model 1974 measures the three orthogonal forces, the moment about the fore-aft axis and the moment about the lateral axis.

Early testing of the knee bolster demonstrated the limitation of the uniaxial femur load cell used in the Part 572 Dummy[3]. The six component load cell that was developed to mount in the same position as the uniaxial load cell revealed lateral loads as well as large bending moments that were generated by impacts to the knee bolster. This Model 1001 Femur Load Cell has been replaced by the Model 1914 Femur Load Cell shown in Figure 12. Model 1914 is also a six-axis transducer, measuring three forces and three bending moments.

Lower Leg

Initial instrumentation of lower dummy members involved placing strain gages on the Part 572 lower leg tubes[4]. Additional demand for data required that both the knee clevis and ankle clevis be instrumented. These early lower legs provided valuable information but they were vulnerable to damage during impact. The fixturing required to calibrate the system proved cumbersome as the complete leg had to be calibrated as an assembly. Despite limitations of this early system, it laid the groundwork for the present rugged instrumented lower legs.

Model 1571 is the instrumented lower leg depicted in Figure 13. It is a complete lower leg and ankle-foot assembly[5] that bolts to the displacement knee. Each transducer of the instrumented lower leg system is a bolt-in component which can be removed for calibration. The tibia load cells are designed to sustain direct impact. Their cables and connectors exit into the leg tubes and are protected from damage during testing.

Knee

The knee clevis is made up of two uniaxial load cells that measure the load on each side of the knee clevis. The sensitive axis is along the line running through the pivot point of the knee and the pivot point of the ankle.

Tibia

The tibia load cells are shown in Figure 14. The upper tibia sensor, Model 1583, mounts to this knee clevis with four bolts and is located by a dowel pin in the knee clevis. Two channels are sensed at this position: the fore-aft bending moment (My) and the lateral bending moment (Mx).

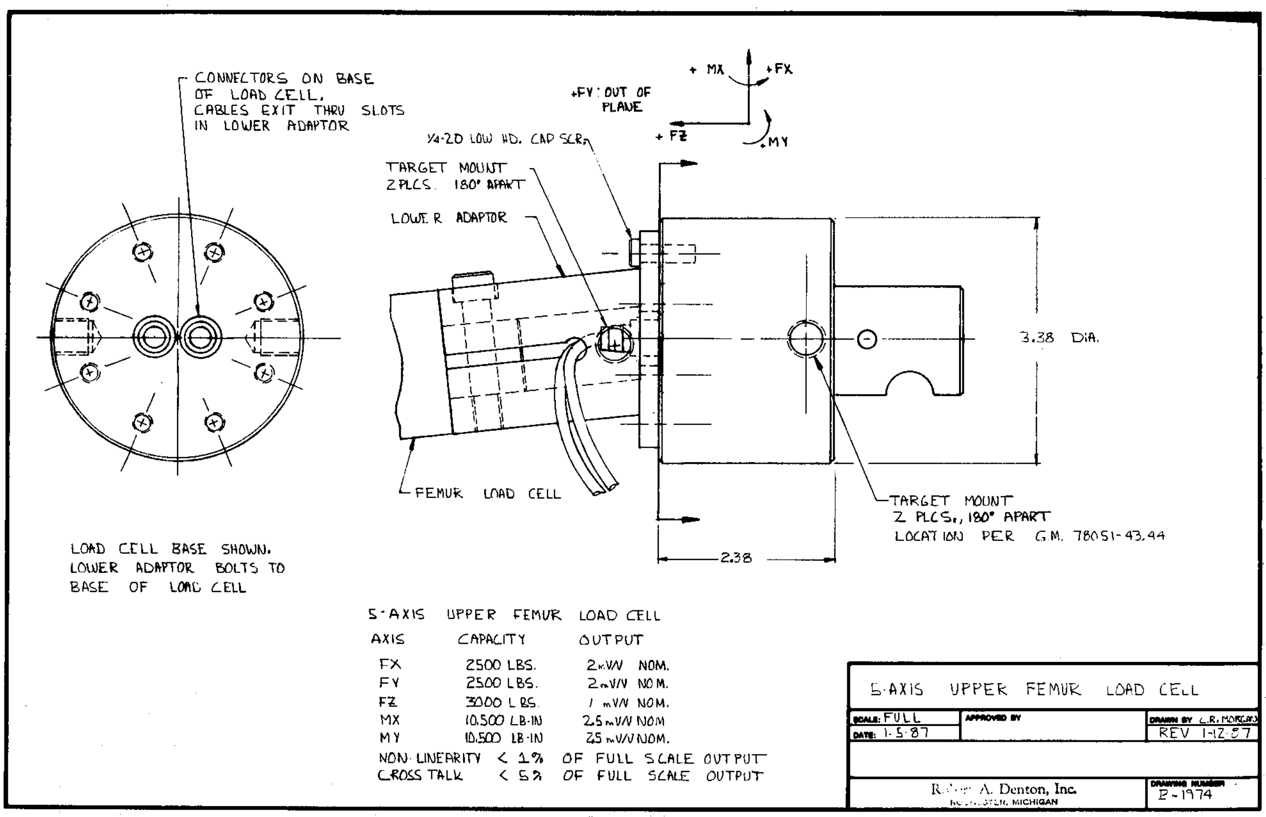

Figure 11. Upper femur load cell model 1974

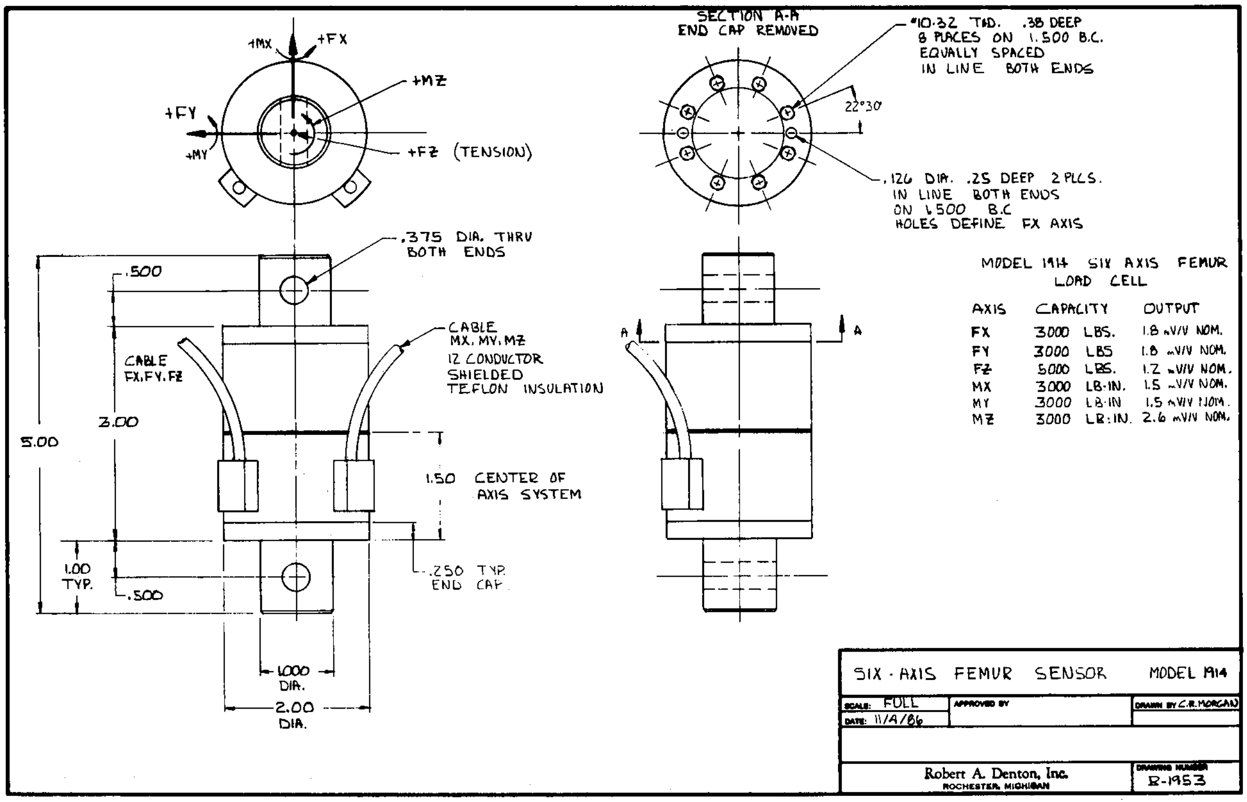

Figure 12. Femur load cell model 1914

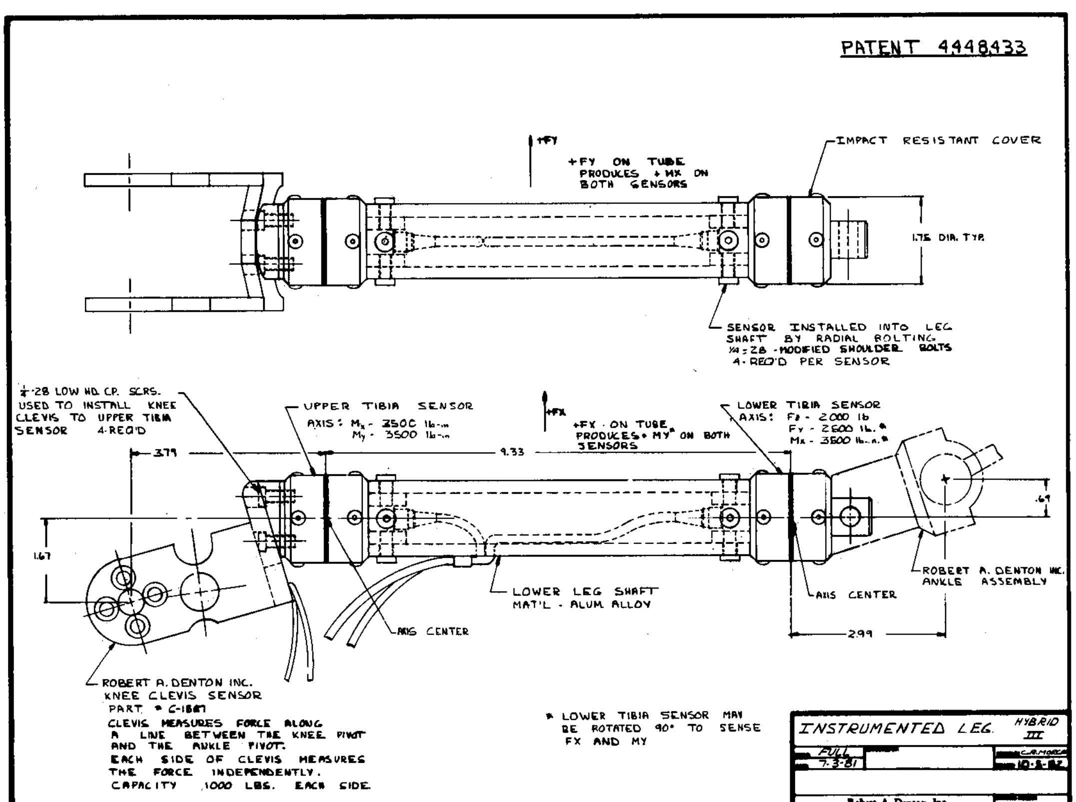

Figure 13. Instrumented lower leg model 1571

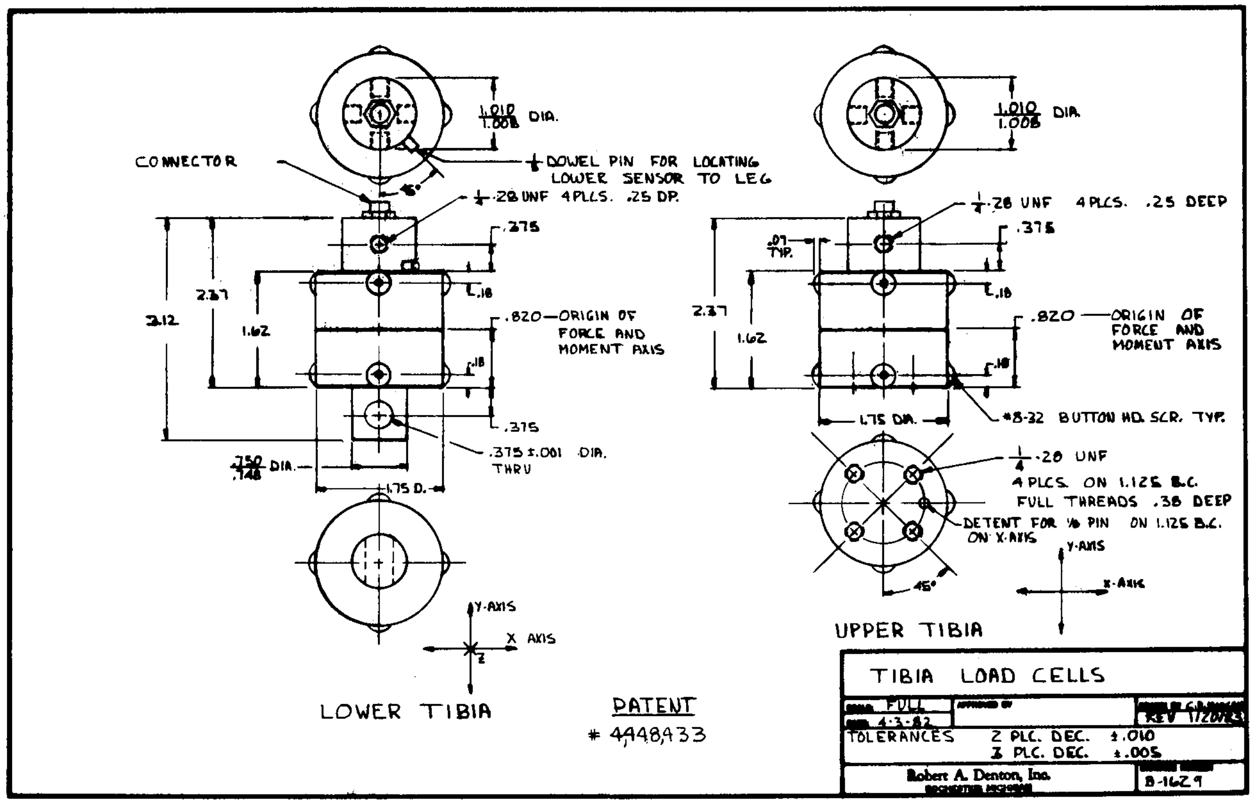

Figure 14. Upper and lower tibias, models 1583 & 1585

The lower tibia, Model 1584, mounted at the ankle has three sensitive axes. The axial force (Fz) measures the load along the axis of the leg shaft, the lateral shear force (Fy), and the moment about the shear axis (Mx). This sensor can be rotated ninety degrees so that the sensitive shear axis becomes the fore-aft (Fx) and the moment axis becomes sensitive to the bending moments from the fore-aft direction (My).

An optional axis configuration is also available for the upper and lower tibia in the lower legs. The upper tibia transducer in this optional configuration contains the two moment axes as in the standard configuration plus an axial force channel (Fz). The optional lower tibia transducer contains a fore-aft moment (My), a lateral bending moment (Mx), and a shear lateral force (Fy). As in the standard configuration, the optional lower tibia transducer can be rotated ninety degrees when installed to sense the fore-aft force (Fx) axis.

Ankle

Attached to the lower tibia is the ankle joint developed at Denton, Inc.. This ankle-foot has the full range of motion of the human foot (per DOT drawing SA 150 M002) and has been accepted as the standard ankle/foot for the Hybrid III.

J-211 Conformity

The engineering practice in developing transducers for the Hybrid III has been to use the three dimensional right hand system. The labels of axes and the output signs had been developed and agreed to on the drawing board, but in actual usage confusion arose as to whether the coordinate system applied to a seated or standing Hybrid III Dummy. There was the argument that since the Hybrid III really could not be placed in a standing position, the axes should be placed with the dummy in a seated position. Another argument was presented that, for clarity, the dummy should be considered to be standing and all axes labeled from this viewpoint. To complicate the problem of analyzing data, the sign of the output is dependent upon which portion of the dummy is considered fixed with respect to the transducer. These problems have been identified and resolved in SAE J-211[6]. All new transducers are designed to conform to these specifications.

A new problem arises with trying to make existing designs conform to J-211. Some models of already manufactured transducers may not conform to the configuration outlined in J-211. The decision was made not to change the internal wiring of these non-conforming transducers because an extremely rugged connector and cable system has been developed. This connector has proven to be highly dependable in maintaining contact during impact, yet easily removed should the cable itself be damaged. If the internal wiring of a transducer was to be changed without changing the connector, the potential for mixing cabling between an early version and a revised version of the same model exists, which would result in data with the improper sign. The lack of small multi-pin connectors that can withstand impacts without loss of signal rules out changing the connectors on the load cells. The solution for these potential problems is simply to maintain the same wiring internally to the load cell with the same cable color code designation but provide an alternate cable color code designation at the instrumentation end of the cell which makes the non-conforming models meet the sign convention in J-211.

Conclusion

In summary, many of the transducers that Denton, Inc. has developed for the Hybrid III Dummy started from a project engineer's need for more information from one test. As time went on, the information from these special transducers was requested by an increasing number of engineers as part of the data required for their tests. The special "one time" transducer became a standard part of the instrumentation package for the Hybrid III. Although not well advertised, there are numerous transducers that have been developed for the "one time" test—for the female, child and Part 572 dummies and other applications in safety testing. In this overview we have focused on the proven Hybrid III transducers that provide answers to questions of measurement that already have been raised. As project engineers identify new areas of concern, and demand more data from each test run, new transducers will continue to be developed. The feasibility of utilizing a complete system of multi-axis transducers must, of course, be decided by the needs of the individual testing facility. But it is this continuing process of transducer development that makes it possible for the instrumentation to be available when the test demands it.

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5. United States Patent No. 4,488,433 to R.A. Denton and C.R. Morgan, entitled "Crash Test Dummy Lower Leg Structure," issued Dec. 18, 1984.
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The Development and Certification of EUROSID

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on behalf of the Ad-hoc
Group of the European Experimental
Vehicles Committee on Side Impact
Dummies

Abstract

The design and development of the components of EUROSID was presented in the report of the EEVC Ad-Hoc Group to the 10th ESV Conference. Four complete versions of this dummy were assembled from components produced by their developers and standard parts purchased from the United States. These dummies, called the First Prototypes, were subject to over 500 component and whole dummy validation tests by five European laboratories, BASt, INRETS, TNO, TRRL, and APR. The experiences learned from these tests have resulted in slight design modifications which have been incorporated into the next prototype version of EUROSID, the Production Prototype. The conclusions from the validation tests are discussed in this report, and the description of the current design, the Production Prototype is given. It is concluded that the development of EUROSID is at a very satisfactory state of development at this point in time and that the final stages of development can commence by the 15 Production Prototypes being evaluated in test houses around the world.

Introduction

A programme of research on the evaluation of the First Prototype versions of EUROSID, supported by the European Commission (EC) has been undertaken by the five laboratories BASt, INRETS, TNO, APR and TRRL. Validation testing of a dummy represents an important stage in its development prior to introduction into service. In the case of EUROSID, it had been started as an EEVC specification of the desirable features for a side impact dummy primarily to be used as a means of measurement of the outcome of a side impact test into a car to check the protection provided for occupants. The EEC sponsored Biomechanics Programme included provision for three side impact dummies (ONSER/INRETS, APR and MIRA dummies) to be designed and initially developed and these were evaluated in comparison with the US SID dummy of that time. The results of this work were put together in a single unified design which was therefore named EUROSID(1). It is this dummy which has been evaluated in the programme and which this report describes.

Guidance about overall features of the design of EUROSID has been provided by the EEVC ad-hoc Working Group on Side Impact Dummies and apart from the representatives of the government (and Peugeot-Renault) laboratories, the group has been helped by several representatives of the European car industry and by one or two biomechanics and dummy experts from NHTSA.

This report describes the Production Prototype version of EUROSID, and the Validation Programme of testing, together with a summary of the results, analyses and findings. Full details of the test programme and results will be given in the proceedings of the EEC Seminar on EUROSID held in Brussels on December 11th 1986(2).

Description of the European Side Impact Dummy: EUROSID (Production Prototype)

EUROSID is a dummy which has been designed specifically for the evaluation of vehicle occupant protection under conditions of lateral impact. Measurements can be made in the head, chest, abdomen and pelvis which relate to the risk of injury to these body parts.

Description by body part

Head. The head is a standard Hybrid III head comprising an aluminium shell covered by a pliable vinyl skin. The interior of the shell is a cavity accommodating triaxial accelerometers and ballast with access provided by removal of a back cap. For more details see reference 3.

Neck. The EUROSID neck comprises three parts:

- a neck/torso interface piece,
- a head/neck interface piece,
- a central section made of rubber that links the two interfaces to one another.

Figure 1 shows the neck construction.

Each interface is composed of two plates; an exterior one (items 1 and 3) and an intermediary one

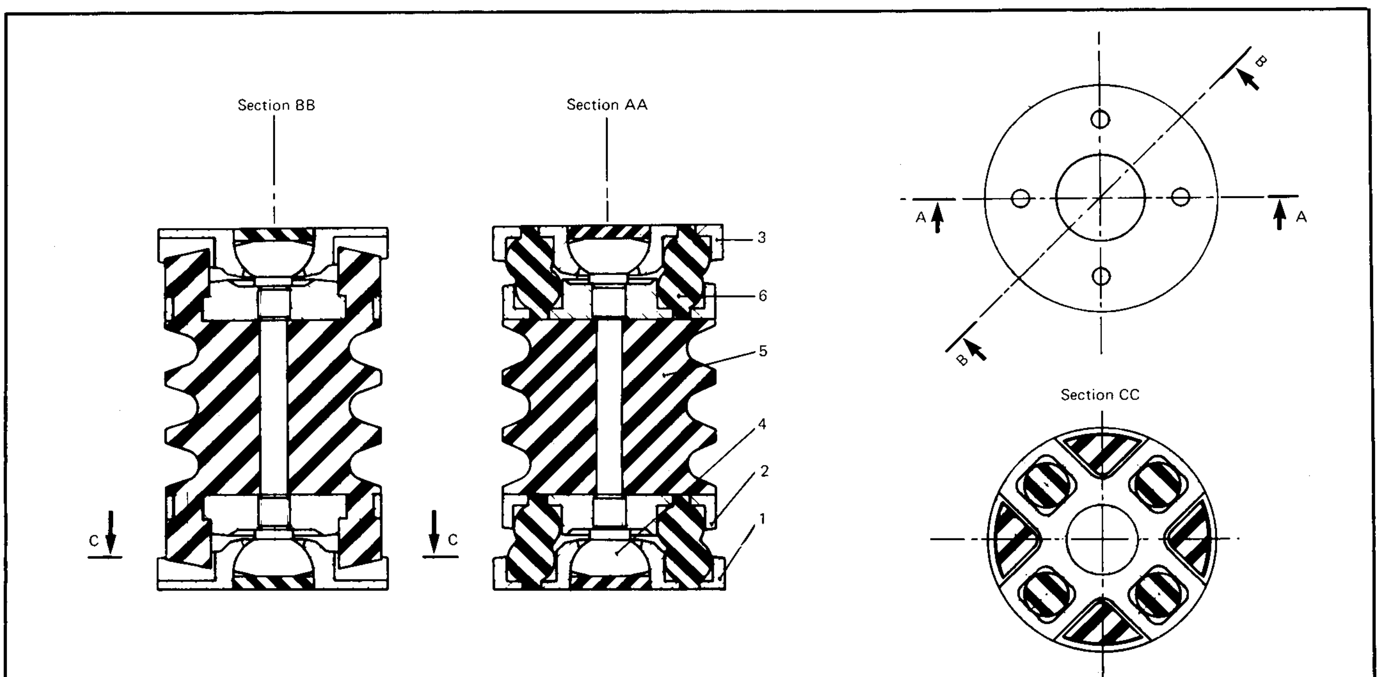

Figure 1. EUROSID neck

(item 2) bound to the central part (item 5). Linking of these plates is made by means of a screwed half-sphere (item 4), which constitutes a point of rotation.

In order to control the head-neck and the neck torso relative movèments, two types of buffers are interposed between the plates as shown in section cc of figure 1.

The triangular section buffers and the neck central part are all part of the same system (item 5), they are made of a special 70-shore hardness rubber. The circular section buffers (item 6) are made of a natural 70-shore hardness rubber.

This design represents a system having 2 pivots and 3 modes of deformation.

The two pivots are represented by the centre of the half-spheres.

The three modes of deformation possible for such a neck are:—

- simple lateral flexion (of the central part);
- translation and rotation (relative movements of the interface plates);
- extension of the central part.

The play which exists around the buffers and the compliance of these latter parts permit torsion; that is to say a rotation around the neck's vertical axis.

No instrumentation is provided in the neck.

Shoulder. The shoulder comprises two polypropylene clavicles which are mounted in a block at the top of the thoracic spine. The clavicle contact with the shoulder block is in the shape of a cam (Figure 2) such that the initial point of contact, and the centre of rotation of the clavicle, is at the posterior end of the block.

The objective of this design is to provide a force couple tending to rotate the clavicle when the top of the arm is struck so that the arm is rotated forward to expose the side of the chest in a lateral impact. The clavicle is constrained to move in plane by plates at the top and bottom of the shoulder block. The clavicle is normally held back against a stop in its neutral position by an elastic strap. No instrumentation is provided in the shoulder.

Arms. The arms have a plastic skeleton (polythene, polypropylene and nylon) with joint movements that are sufficiently realistic for lateral impacts. The upper arm flesh in the struck area is Sorbothane, while that in the forearm is solid polyurethane. The hand is solid PVC, the wrist having rotational and flexion capability. The arm is attached to the end of the clavicle and does not contain instrumentation.

Chest. The chest comprises three identical rib modules individually attached at the rear to a rigid thoracic spine box. (Figure 3). Each rib module consists of a hoop of spring steel attached at one end to the lateral control cylinder and at the other to the piston running in that cylinder.

This is the struck side of the rib module. (Figure 4). The piston runs on bearings within the cylinder and attached to the end of the piston. Between the end of the piston and the end of the cylinder is a sleeved spring, the stiffness of which can be selected to 'tune' the rib module. Lying parallel with the cylinder is a damper, specially produced by Armstrong Patents Ltd.

The damper piston is connected to the struck side of the rib by a spring. The whole module is attached to the spine box by the lateral control cylinder and can be removed intact as a unit. This concept allows the rib module to be serviced and certified separately from the whole dummy.

Each rib module is equipped with an optical lateral displacement transducer. A cyclic four-bit reflective gray code is attached to the piston and a detector unit is attached to the cylinder. The transducers from the three ribs are coupled to a single signal conditioning unit which provides a voltage output proportional to the displacement for each rib. The unit also has a digital peak hold meter which gives an instant reading of the peak deflection of each rib.

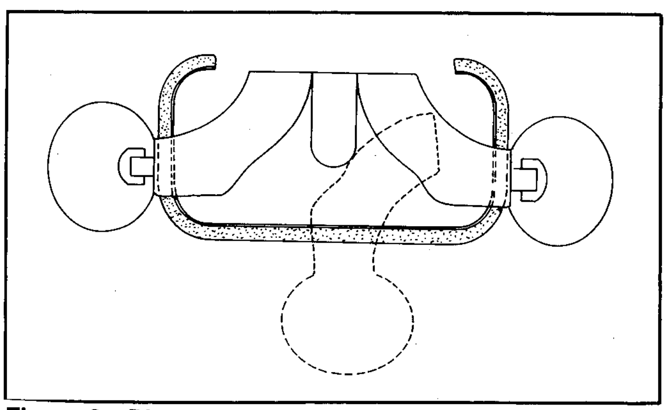

Figure 2. Diagram of shoulder cam (top view)

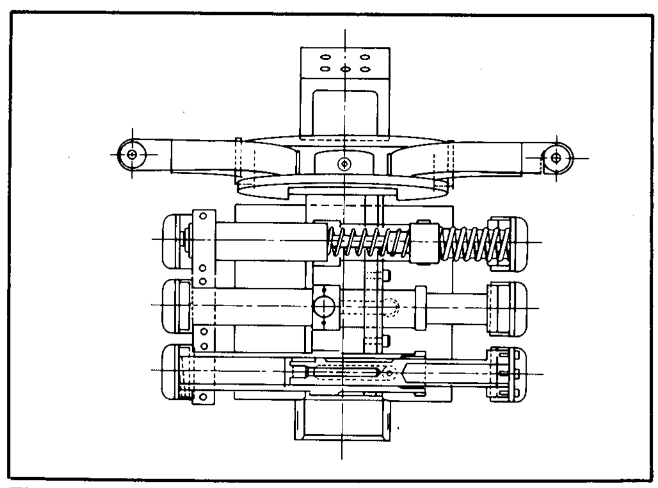

Figure 3. Thorax (shoulder and chest) assembly

Provision has been made to attach a triaxial accelerometer at the top of the thoracic spine within the shoulder block and to attach uniaxial accelerometers to the inner surface of the ribs between the piston and damper attachments if acceleration figures are required.

Abdomen. The central part of the abdomen section is a metal drum which is rigidly attached to the lumbar spine-thorax box interface of the dummy. This drum is covered by a flexible material up to the outside of the abdomen which allows a penetration of 40mm before "bottoming out". Between this flesh-simulating material and the rigid drum are located three vertical steel leaf-spring switch units. These switch units can be individually adjusted to trigger at a prescribed force and are normally set to trigger at an externally applied force level of 4500N (corresponding to AIS 3).

The inertial mass required to obtain correct biofidelity up to the point at which the tolerance levels are exceeded, is obtained by moulding a total of 1.4kg of lead, partly as small pellets in the outside contour of each side of the flesh-simulating polyurethane foam at the impacted side (Figure 5).

The three leaf-spring switches can be connected in parallel so that operation of any one of them will cause an 'event' signal. Contact of each individual switch can be detected if required.

The abdomen section is connected to the pelvis by a conventional solid rubber lumbar spine which is a standard FMVSS 208 Part 572 unit.

Pelvis. The external shape of the pelvic bone is representative of the shape of the human pelvis at the points directly involved in a side impact and at the interactions with the car seat, as well as at the iliac crests where the seat belt fits around the pelvis.

The external shape attempts to represent accurately the way in which a human sits on a car seat. The pelvis is composed of two iliac wings made of cast aluminium alloy. Each iliac crest is covered with 4mm of elastomer, to decrease the impact effects. The two iliac wings are linked together at the pubic symphysis by a force transducer. Rearward of the pelvis, the sacrum is fixed on each lateral side to an iliac wing and is the base for the lumbar spine (Figure 6).

The hip articulation of the EUROSID pelvis is intentionally different from the human one. To minimize the effect of leg position on pelvis loading, external forces are transmitted to the pelvis along an axis passing through the hip ball joint, so that the thigh position has no effect on the way in which an impact to the greater trochanter loads the pelvis, but an impact to the thigh loads the pelvis at the same point as in humans.

The design of the hip joint allows adduction and abduction angles of about 25°; however, due to interaction of the flesh in the pelvis area and the two thigh upper extremities, the adduction and abduction angles will be limited by the deformation capability of the foam in this area. A large Sorbothane cylinder is attached to a steel plate fixed on the iliac wing by an axis going through the ball joint. The Sorbothane compensates for the rigidity of the shell. The mechanical assembly is covered with a polyurethane foam which gives a dense skin over all its surface. A polyurethane film is also applied to the foam to increase its superficial tearing resistance.

The EUROSID pelvis is designed to measure pelvic compressive force in the pubic symphysis area by a force transducer and to the iliac wing by strain gauges fixed to the inside of the ilium.

In addition, the sacrum block is designed to accept an accelerometer so that pelvic acceleration may be measured if desired.

Legs. The legs are based on a metal skeleton covered by a flesh-simulation. Joints are provided to allow realistic motion at the knee and ankle. The legs are the standard FMVSS 208 Part 572 design. No instrumentation is provided in the legs.

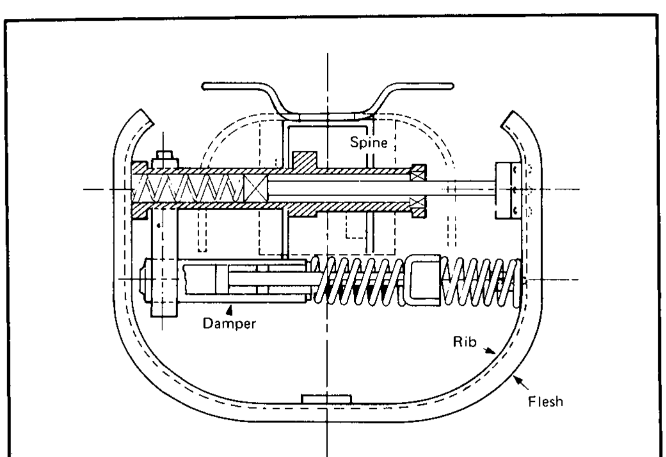

Figure 4. Diagram of rib—piston—damper module

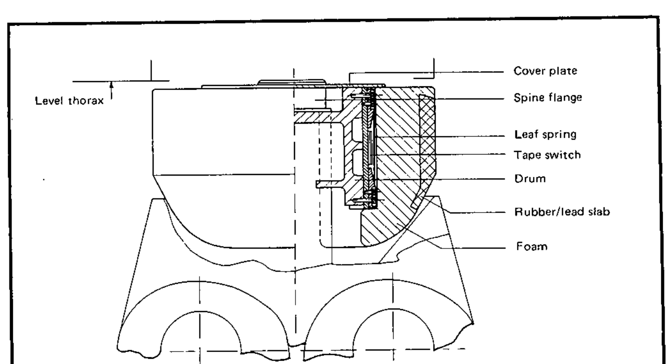

Figure 5. Abdomen section

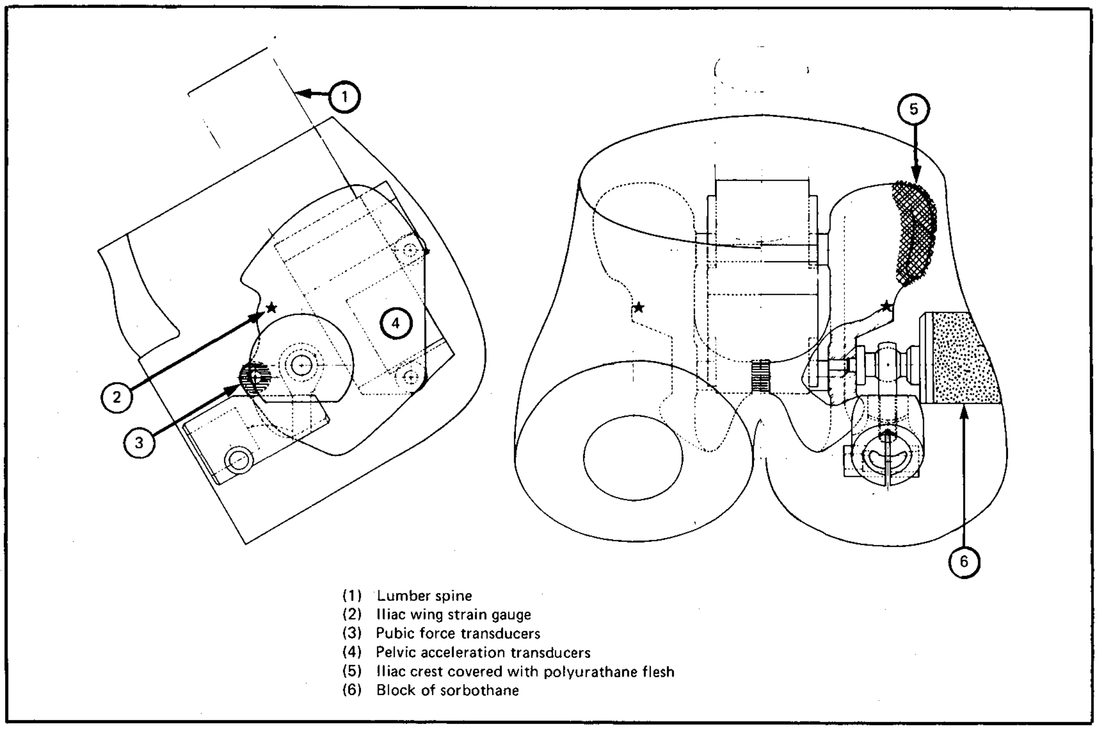

Figure 6. EUROSID pelvis diagram

The whole dummy, representing a 50th percentile adult male based, as far as practical, on recent authropomorphic data(4,5,6), is shown in figure 7.

The Validation Programme of EUROSID Testing

The programme of tests was set up by the five contractors to check a large number of aspects of dummy performance. It was arranged that most tests of any particular type to check each part of the dummy (or the dummy as a whole) would be carried out twice. One set would be by the design team for the component in question, while the second would be by another team. The contribution of BASt would be to check the dummy as a whole in a series of seven tests. APR had the special role of finally developing their neck design for this side impact application. The plan for testing is given in the Table which shows the tests performed by each laboratory (with the column headings T for Thorax A for Abdomen and P for Pelvis).

The whole extensive programme of testing to check the validity of EUROSID has been completed. With additional tests to those scheduled in the contract plan, the total number reaches about 500 tests. The additional tests were carried out to meet the required test conditions more exactly, to find more accurately the circumstances in which the abdomen switches triggered and to check slight changes to the dummies.

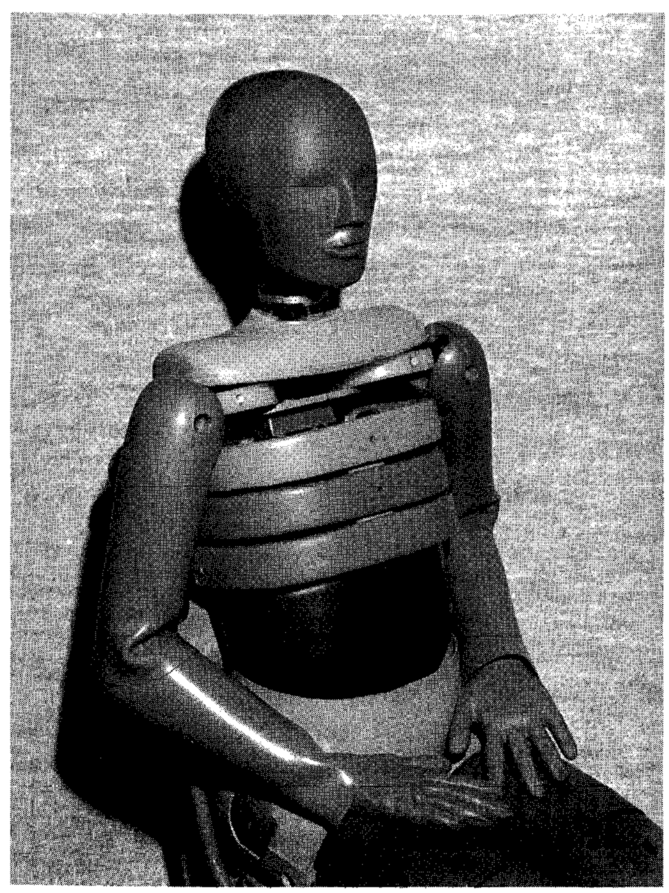

Figure 7. EUROSID first prototype

The 500 or so tests were shared between the four similar prototype EUROSIDs especially built for the programme. This means that they were all extensively tested, except perhaps for the BASt dummy which had only seven full scale tests.

The programme of tests was planned to investigate the following aspects of dummy characteristics. Comments and conclusions are given under each of the following seven headings for every part of the dummy involved in side impact.

1. Biofidelity. This was not a major aspect of the validation test programme but nevertheless it does include a series of sled tests which are basically similar to impact tests on cadavers so that responses can be compared. The consideration of matters of Biofidelity and Scaling will be considered in more detail after the completion of the Validation Programme.
2. Scaling of injury criteria values. It should be possible to relate dummy responses to the levels of impact on humans which lead to various important injuries. This can be done only when the critical loadings on human beings are already known and for some parts of the human body full data is not available for side impact situations. It is expected that sensible and appropriate estimates can however be made.
3. Repeatability of response to similar impacts. What is the mean response and its Standard Deviation when a particular impact is repeated a number of times into one particular dummy?
4. Reproducibility. To what extent do the four dummies give similar responses? In other words, reproducibility is repeatability between different dummies.
5. Sensitivity of response of dummy. This aspect is to find whether the measured responses of the dummy change by appropriate amounts when there are changes in the way it is impacted. For example how does the response change with point and angle of impact? The appropriate responses are usually those which change in a similar way to human responses. Sometimes a flat response is however needed while in other circumstances a more rapid change of response is to be preferred for a dummy.
6. Durability of dummy. To what extent is the dummy damaged? Does it need realigning and does it need recalibrating and setting up when it is impacted at severities corresponding to typical impact test levels. It should also cope with appropriate overloads without failure and loss of adjustment.
7. Certification. The certification procedure is the test or set of tests which have to be carried out to confirm that the dummy is correctly adjusted and is responding within specified limits. This is usually done before a series of tests.

Test Observations by Body Part

Neck and Head

As already stated, the head is a standard Hybrid III but the neck has been especially developed by APR to give a good biofidelity of head movement relative to the trunk in lateral impacts.

Biofidelity. The measurements clearly show the much improved flexibility of the neck which well represents the human at low volunteer levels as well as at the more severe cadaver test levels. The alternative design using the 70 shore hardness for the neck segments is preferred. These neck three dimensional flexibilities are important because they determine the way in which the head hits obstacles in lateral impacts. The head impacts are therefore much more meaningful and the influence of the mass of the head on the upper body response to impact is also improved.

Scaling. Because no measurements of neck movement are proposed, the scaling of these for EUROSID in comparison with live human beings for the development of protection criteria has not been made.

Repeatability. Results show a high degree of repeatability at the part 572 neck calibration level with a maximum coefficient of variation (SD/mean value) of 7 per cent with most results having much lower ratios.

Reproducibility. This was not checked. The sled tests restrain the head so that free neck movements do not occur.

Sensitivity. Sensitivity tests were not carried out because it was considered that changes in head movements due to small changes in impact conditions would not greatly affect the movements being taken on EUROSID. In most practical circumstances the head nods sideways through a large angle unless it first strikes some part of the side of the car.

Durability. The neck generally performed satisfactorily as far as durability was concerned but there was found to be a semi-permanent bend of 5° in one dummy one day after 25 tests. There was also a small permanent set in the one dummy. Although not of major importance this shows the need for the "re-set" procedure for the neck. It was realised during the test programme that EUROSID must not be carried around by supporting it through a hook in the head. This destroys the neck.

SCHEDULE OF PROPOSED TESTS IN EUROSID EVALUATION PROGRAMME 1985/86

	No.	BASt	INRETS	TNO	TRRL	APR
			T A P	T A P	T A P	
1. Sensitivity						
(a) Variation in output with varying energy input, Thorax and Pelvis						
(Impactor) (i) Constant mass (velocity changed)		*	* * ✓	✓ ✓ ✓	✓ ✓ *	*
(ii) Constant velocity (mass changed)		*	* * ✓	✓ ✓ ✓	✓ ✓ *	*
(b) Variation in output with change in impact contact area, (constant energy input)		*	* * *	✓ ✓ *	✓ ✓ *	*
(c) Variation in output with angle/position of impact (constant energy)		*	* * ✓	✓ ✓ ✓	✓ ✓ *	*
(d) Variation in output to varying input velocity: Rigid wall sled test 20–30 kph		*	✓ * ✓	* * *	✓ * ✓	*
(e) Variation in output to input conditions – Padded wall sled test 20 kph		*	✓ * ✓	* * *	✓ * ✓	*
(f) Temperature sensitivity (Temp Soak and Lighting effects)		*	* * *	✓ ✓ ✓	✓ ✓ ✓	*
(g) Other		3 MDB-car impacts 45 50 & 55 km/h (EEVC Face)	Hard spot wall 2 tests	*	*	*
2. Repeatability and Durability	No.					
(a) Neck 572 style Pendulum test	20	*	*	✓	*	✓
(b) Thorax Impactor Test	20	*	*	✓	✓	*
(c) Abdomen Impactor Test	20	*	✓	✓	*	*
(d) Pelvis Impactor Test	20	*	✓	✓	*	*
(e) Shoulder Impactor Test	20	*	*	✓	✓	*
3. Overall Dummy Kinematics						
2 KOB Accidents – each reconstructed twice.		*	✓	*	*	*
EEVC Mobile Deformable barrier – Car Impacts		2 MDB Car impacts at 50 km/h	*	✓	*	*
4. Reproducibility		*	*	*	3 rigid wall tests on each dummy	*
		2 crabbed MDB car impacts	*	*	*	6 sled tests to develop the APROD neck for EUROSID

Certification. A procedure is being developed to take account of the three dimensional flexibility of the neck. This is being based on the Part 572 neck pendulum test.

Shoulders

The EEVC committee which drew up the outline specification decided that injuries to the shoulder were not of major importance and so it would be preferable to have a dummy shoulder system which moved out of the way of a lateral impactor at low loads so that the thorax is not shielded in a variable manner by the upper arm.

Biofidelity. The peak forces generated in the impactor tests were in conformity with unpublished biomechanical reference data. The specification for the EUROSID shoulder lays down an initial compression of the shoulder at a low impact resistance. It is designed to deflect forwards in a horizontal plane out of the way of the thorax.

Scaling. There is no known relationship between compression and injury for the initial compression made possible by the shrugging of the human shoulders. This occurs at low loads for up to 55 mm compression.

Repeatability. Pendulum impactor tests were used to check the repeatability of the shoulder. The TNO tests at 4.28 m/s gave a mean maximum deflection of the shoulder of 84.6 mm (SD 4.4) with a peak force on the impactor of about 2.9 kN. This is regarded as being satisfactory. The tests performed on the TRRL dummy gave a mean lateral deflection of 66 mm (SD 10.8mm) with a peak shoulder force of 2.65 kN.

Reproducibility. This was not checked apart from the comparison between the results quoted under Repeatability using two different dummies which gave slightly different peak forces and somewhat larger differences in the maximum deflection. The two dummies were tested with different designs of impactor; one a pendulum suspended on wires, the other controlled by linear bearings. This difference in performance means that the certification test procedure must define the impactor.

Sensitivity. The repeatability tests had some variability of point of impact from between 10 to 20 mm forward and behind of the nominal point of impact. These results correspond to the expected freedom of the shoulder mechanism to fold forwards when struck laterally or from the rear of that position.

Durability. During the TNO impactor tests the shoulder clevis screw, which also holds the rubber strap became loose through dummy manipulation, releasing the rubber return strap. During the TRRL rigid wall tests a failure of the shoulder cam and cam spring attachment screws was observed. Design modifications have been made to avoid this. The only serious problem occurred in tests with the complete EUROSID when the arm became trapped between the incoming side and the thorax. It tended to jam at the elbow or just above and then with further loading to force the upper arm away from the rest of the dummy at the shoulder. This pulled the shoulder clevis screw out of the 'clavicle'. However EUROSID should not be used with the arm in this position and design changes for the Production Prototype should eliminate this so this damage should not normally occur.

Certification. Apart from a check on the condition of the shoulder, the only test is a standard pendulum impactor test together with a specification for the return spring tension.

Thorax

Biofidelity. The success which this thorax may have in reproducing good biofidelity of response has not yet been fully determined but it is based on cadaver responses to impact and hopefully should model these over a much greater range of speed than can simpler models. In the future it will be possible to measure various other characteristics of thoracic response, which will probably be useful for measuring the severity of impact for other modes of injury.

Scaling. Although scaling of loads to injury has not yet been decided, it is clear that severe crushing of the rib cage with the possible consequent failure of the lungs is represented by compressions of about 45 mm. More localised impacts can occur at much lesser impacts and these may cause individual ribs to fracture which may in turn puncture the lungs. Even more severe impacts are modelled up to 50 mm after which bump stops stiffen the system and compressions of up to 55 mm can be recorded in this mode. Localised injury from protruding hard spots striking the thorax can be detected by differences in the compressions and responses of the three ribs. Results clearly show that only part of the rib cage may be impacted in practical circumstances and it is important that such localised overloading is recorded. First comparisons with published cadaver results suggests that rib deflections of 40-45 mm might be equivalent to AIS 3, but a more detailed comparison will be required before more definite figures can be proposed.

Repeatability. Each rib unit can be set up fairly precisely and so it is not surprising that good repeatability in simple pendulum tests direct onto the rib cage can be measured. The observed coefficients of variation (SD/mean value) were 2 to 5 per cent by TNO and 1 to 3 per cent by Middlesex Polytechnic for TRRL. Once the EUROSID arm can get between the ribs and the impacting object, the repeatability is relatively poor and this is an important consideration in the use of the dummy as a whole.

Reproducibility. Differences were found between the ribs and pelvis of different dummies in the reproducibility tests carried out after the remainder of the test programme had been completed. The results are somewhat confusing and, in some cases, are masked by the variability of the individual dummies under the particular test conditions chosen. Some of the significant differences found were not large enough to be important. Design modifications to the pelvis, abdomen and arms have been implemented to reduce the variability and other differences.

Sensitivity. Chest deflection appears to be related to impact energy, that is to mass and to the speed. The variation with mass is less clear because the changes following from changes in impact mass were not large. The variation with speed is at least proportional to speed if not to the square of the speed. It appears that peak chest compression is dependent on impactor energy while peak rib acceleration is dependent on impactor velocity. An important feature is the sensivity to lateral impact angle and this was tested from 70° to 110° (90° being exactly lateral). The rib compressions are greatest at 90° and fall off somewhat for impact angles ahead of this. Somewhat similar effects were noted for rib accelerations. It is not known how these responses correspond to injury risks for the human thorax. With regard to the response to the size and positioning of the impactor, the three ribs showed the obvious effects on rib compressions. A small 45 mm high rectangular impactor striking only the middle rib did not deflect the other ribs, but in a 3.5 m/s impact, the middle rib deflection increased from 24 mm to over 50 mm into the overload bump stop range. There appears to be almost no effect of temperature within the range tested (15 to 25°C). Sled tests also confirmed that the dummy thorax is sensitive to hard spots in an impacting object.

Durability. The rib systems generally withstood the impacts of the extensive test programme without damage. The impact speeds were up to 55 km/h. One exception was with one rib piston in the TNO dummy which was found to be bent after the test programme. The piston design has subsequently been strengthened. There was some tendency for the damper springs to come out of their guiding mechanisms. The early prototype electronic circuits to record and process the rib piston deflections gave a number of problems and further developments are hopefully dealing with these. Detailed examination of the shape of the rib deflection curves suggests that in certain circumstances the full deflection is not quite reached. This appears to be due to some flexibility of the rib/spring/damper system at the impacted ends of the ribs and some stiffening of the rib/piston joint appears to improve matters. Further investigations should determine what action is needed.

Certification. There is a full range of certification tests for each rib assembly. The rib springs and dampers are separately checked in a series of drop tests and then each whole assembly is check tested. If the performance is found to be outside agreed limits, it is possible to select slightly different springs and, with some difficulty, to adjust the dampers. The test programme suggests that these possible adjustments are rarely needed.

Abdomen

Biofidelity. The design seeks to model the human in dynamic impact situations when the impacting object is relatively flat and is of medium to large frontal area. An essential feature is the correct lead ballasting of the model abdomen which greatly modifies the dynamic response.

Scaling. The settings of the event switches can be adjusted. Future development could possibly enable some continuous measurement of impact to be made. With the event marker switches it appears that their triggering conditions do depend quite noticeably on the switch settings and a setting of 0.7 mm is proposed. In one set of tests this gave switching at from 3.85 m/s to 4.05 m/s for an impact mass of 18 to 28 kg and a switching force from 4.88 kN down to 3.08 kN for the heaviest mass. The impactor was 70 mm high and 150 mm wide with a flat face and 5 mm edge radius. This is thought to be an appropriate setting for abdominal injuries of A1S3.

Repeatability. Although in one set of tests there appeared to be a little softening of the abdomen resistance after 14 tests, another check showed a close repeatability of the pedulum force/deflection (or compression) curves. In this set of 6.33 m/s impacts the switch contact force had a mean of 4.55 kN with a Standard Deviation of 79 N. The softening may have been due to some internal tearing of the abdomen.

Sensitivity. The dynamic response of the abdominal block of ballasted foam controls the switching conditions in a humanlike manner and the sensitivity checks generally confirm this. For example the switching is not sensitive to temperature (15°C to 25°C) or surface heating, but responds primarily to velocity of impact. As noted in the paragraph on scaling the effect of impact mass in the range 18 to 28 kg is not great. The effect of angle, forwards and behind exactly lateral, is not great with the appropriate one of the three switches actually switching first when the impact was in its direction. There was not a large effect from a change in the area of the impactor but rather unexpectedly one set of tests showed that a higher impact velocity was required to operate the switch when a smaller area triangular cross-section impactor face was used. (TRRL tests). However another set from TNO showed that the force needed for switching was lower (3.5 kN rather than 4.62 kN) when the 'triangular' face was used.

Durability. This was generally satisfactory. The abdominal flesh showed several tears after a number of tests and this did appear to affect some later results to a small degree. After the sled tests several of the event marker leaf springs appeared to be displaced. A redesign will ensure that in future EUROSID dummies the abdomen will not be able to move a little way into the lower rib space.

Certification. There would appear to be little need to check the certification of the abdomen apart from carefully checking that the various components are correctly set and are not torn or damaged. A pendulum test is to be used to confirm the triggering of the switches. The validation programme shows that the switches need careful setting up and in one set of tests it was found that the switches did not trip until 5.74 m/s with a 5 kN impact using a 23.4 kg mass when the switches were set farther apart than the standard setting.

Pelvis

Biofidelity. Geometrically the shape of the pelvis attempts to represent accurately the way in which a human sits on a car seat and it suitably links the legs to the torso so that realistic angular freedoms of movement between them are available. The human pelvic structure consists of a lightweight and flexible shell filled with a somewhat fluid sack. This is difficult to reproduce for a dummy and the dummy structure tends to be unrealistically heavy and rigid. the INRETS design reduces the mass of the shell as far as is practicable and compensates for the rigidity of the shell by placing a large Sorbothane cylinder between the shell and the point of impact into the human greater trochanter. Its characteristics are based on a model so that the correct impact response is simulated. The iliac crest on EUROSID is relatively stiff with surface padding to reduce ‘ringing’. The measurement system is positioned to record the impact at these two most likely points of lateral impact. Being measurements of load rather than acceleration, they record both impacts which throw the pelvis inwards into the car and cases in which the pelvis is crushed between the incoming structure and other car components in the middle of the car.

Scaling. The force developed in the pelvis which might be described as the pubic force best represents the human pelvic response at the greater trochanter at an impact speed of about 6 m/s. It then develops a resistance of about 5 kN. At higher speeds EUROSID is stiffer than the cadaver and at lower speeds it is softer. The iliac crest has not been matched to cadavers because data from the latter are not available. Preliminary review of the results suggest that 10 kN for each transducer would be an appropriate performance criterion.

Repeatability. With the INRETS repeatability tests at 4.39 m/s (SD 0.05) the maximum pelvic acceleration was 30.45 g (SD 0.94) at a pubic load of 3.099 kN (SD 0.188) over 19 tests. The TNO results were similar. This test does not impact the iliac crest. There was a decline in peak pubic symphysis force at TNO which could be attributed to a problem of load-cell adjustment. The full scale tests at BASt with EUROSID in a Golf car gave good repeatability with maximum loadings of the pubic symphysis being 4.5, 5.3 and 5.6 kN in 50 km/h tests.

Reproducibility. The two sets of repeatability tests on different dummies gave very similar results as noted above. The tests using Mobile Deformable Barriers into complete cars illustrate the fact that any dummy such as EUROSID is sensitive to the position of the impact into it. This means that although the total impulses for two tests may be similar it is readily possible to get different proportions of the dummy loading taken by the pelvis, abdomen and thorax. The pubic symphysis force differed significantly between the four dummies in the sled reproducibility tests, but this may have been due to differing amounts of pelvic flesh damage suffered by the different dummies.

Sensitivity. In pendulum impacts the pubic forces are highly sensitive to impact velocity as already noted. The response to impact mass was much as to be expected once instrumentation problems were resolved. There was an indication that higher pubic force loadings occurred when the impact was forwards of exactly lateral (70° rather than 90°). The size of the impactor does not have much influence as long as the iliac crest is not impacted. The pelvis measurements are not very sensitive to temperature. The largest change was noted at 15°C when the force generated at the pubic symphysis was 15 per cent higher than at the other temperature of 20°C and 25°C for otherwise similar impact test conditions. The sled tests provide further evidence about sensitivity. These confirm the rapid increase in pubic load with speed of impact into a rigid wall and similarly for the loading of the iliac wing. These tests show the effects of padding the wall that is impacted. There was a halving of pubic load with the various paddings tested at 6.7 m/s. Similarly the pelvis was sensitive to the hard spot in that the recorded pubic load was well above that for a rigid wall and this indicates a great increase in impact pressure. It was noticed that the loadings of the iliac wing were not greatly altered by the use of wall padding for the impact conditions tested. In other words the area of padding at the greater trochanter may have crushed sufficiently on the padded wall test for the iliac wing with its relatively greater stiffness to have taken a greater proportion of the total load. However in the TRRL sled tests the stiffer padding reduced the ilium force by 30 per cent and the weaker padding by 73 per cent. This suggests that the thickness of this weaker padding was large enough for it not to be 'bottomed out' at the speed of the impact test of 7 m/s.

Durability. The pelvis withstood the impacts of this programme well, apart from 3 detailed matters which have been rectified. The pubic symphysis load cell needed setting up and recalibration on one dummy. All the dummies showed tearing of the pelvic flesh and a redesign of this has been subsequently undertaken with the dummies being fitted with new flesh and skin after the test programme. The ball joints at the top of the femur were damaged and high strength bearings have been inserted to replace the original ones.

Certification. There was not a precise certification procedure for the validation programme. The results of the programme give plenty of data on which certification could be based and as a result INRETS has now provided a procedure.

Tests on complete EUROSID

Three types of tests of the complete dummy were carried out. These were a large number of sled tests in which the dummy slides laterally on its seat as the trolley decelerates to rest. The tests were with either rigid or padded walls. Then there were seven tests at BASt and two at TNO with side impacts into cars with the EUROSID dummies inside. These were impacted by mobile barriers with deformable faces to the EEVC specification. Lastly there were four reconstructions by INRETS of actual accidents studied in the KOB programme.

The conclusions of various sled tests using one or other of the dummies have been mentioned in the preceding Section where appropriate. It remains to mention the reproducibility sled tests in which each of the 4 dummies was tested three times in similar conditions into a rigid wall. These tests were the last in the programme of 500 tests and so the dummies had already been extensively tested. Some results appeared to be affected by minor damage and in changes in the settings of the instruments and components. Generally the shapes of the measurements as functions of time were closely consistent between the four dummies, but there were some variations in peak loads. The earlier tests had shown that in most respects the individual body components responded in a very similar way as each other. The differences with complete dummies seem to arise from four causes. There were very slight changes in impact velocity and slight variations in the balance between the proportions of the impacts taken by the thoracic and pelvic parts of the dummies. Thirdly there were some apparent effects of damage to components and these suggested the need for the minor design changes already mentioned in this report. There was also the strong possibility that the arm was interfering with the loading of the rib cage in a rather variable manner. The abdominal insert lifted and interfered with the lowest rib in some tests. Despite these possible effects, the rib compressions (or deflections) were almost all in the range 35 mm to 50 mm with only five out of the 30 readings outside this range.

Tests with the rigid wall showed that the shoulder did not operate as intended because the arm became trapped and the shoulder did not move forwards. This resulted in a significant force being transmitted directly to the spine through the shoulder. It may be possible to avoid this happening by placing the arm forwards before the test. Design changes have been incorporated into the Production Prototypes to avoid this problem.

It is concluded that the tests on the complete EUROSIDs indicate a generally satisfactory design of the dummy once the detailed matters outlined in this report are incorporated in the design of the dummy and in its set-up before testing. Further testing with the Production Prototype EUROSID dummies will be required to confirm this conclusion.

Performance Criteria

The Ad-Hoc Group is currently considering the performance criteria that would be applicable for use with EUROSID in lateral impact testing. Provisional proposals have been made, which need to be verified or modified in the light of past and current studies.

These preliminary proposals are:—

Head	HIC [1000] secs.
Neck	No criterion
Shoulder	No criterion
Chest	Maximum deflection of any rib [40-45 mm]
Abdomen	No switch contact, corresponding to a force level of [4.5] kN at 39 mm compression.
Pelvis	Maximum Force Level recorded at a) Pubic symphysis [10] kN and b) Ilium [10] kN
Legs	No criterion

For the thorax, the Group also recommended the measurement and evaluation of the Viscous Criterion(7,8) and the TTI(9) in any future testing as possible additional criteria for future use. Very tentative suggestions for performance criteria levels of [1] for the viscous criterion and [80 to 100] for the 'kernel' TTI were made.

Final Comments and Conclusions

This report describes the current, Production Prototype, version of EUROSID and covers the whole validation programme performed on the earlier First

Prototype version. This comprised a set of over 500 tests which were carried out by the five European laboratories BASt, TNO, INRETS, APR and TRRL with the help of a number of other oganizations in support of them.

The four First Prototype EUROSIDs were assembled early in 1986 at various laboratories from components produced by the four laboratories responsible for them (INRETS for pelvis, TNO for abdomen, TRRL for thorax and shoulders and APR for neck). The intention of the test programme was to check the performance of the dummy in some detail so that any problems should become apparent. This is what has happened because although the results were generally satisfactory, a number of detailed matters have shown up. Modification to improve these features have been incorporated into the Production Prototypes. It was not surprising that some problems were detected, because the programme provided the first opportunity for testing EUROSID in its complete form with the actual interactions of one component with another.

The validation programme looked at four main aspects of dummy performance in some detail. There were repeatability, reproducibility, sensitivity and durability. The programme checked the dummy both by impacting each important component separately and by impacting the dummy as a whole. It is concluded that EUROSID has been very satisfactory when considered component by component. All showed good degrees of repeatability as each component was checked up to 20 times. The sensitivity of response to small changes in impact conditions was measured. Changes in background conditions such as temperature were found to be very small. Changes in points and angles of impact and in the type and severity of impacts provided valuable information which is needed in the planning of the set-up and the use to be made of the dummy and in the interpretation of the measurements to be taken from the dummy. The durability of the many components was shown to be of a high order with only a few detail design improvements appearing to be needed. These have almost all been prepared for incorporation into the next batch of EUROSIDs, the Production Prototypes.

Another result of the programme is that test certification procedures can be devised with some confidence. Although some were available at the start of this validation, the remainder can now be defined and have in fact been largely developed since the end of the programme.

Looking to the future, experience with EUROSID suggests that it will be possible to adapt it readily to future requirements such as, for example, further criteria for measuring injury to the thorax. The abdomen could be adapted to recording a range of injuries.

In final conclusion it may be said that the validation testing of the First Prototype EUROSIDs has almost entirely met its objectives and the dummy has been shown to be at a very satisfactory state of development for this point in time. It is thoroughly suitable for being taken through the final stages of development by being evaluated in test houses around the world. It is ready to be specified in the near future for use as a test measuring tool for legislative side impact testing in cars for checking improvements in the protection of car occupants.

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Technical Session Four

Crash Avoidance

Chairman: Robert Nicholson, United States

Accident Avoidance: An Analysis of Inherent Vehicle and System Response Differences Between Cars and Commercial Vehicles: Are Developments Widening the Performance Gap?

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Abstract

Major differences in basic accident avoidance characteristics are almost inevitable when one compares typical private cars with various other cases of motor-driven road vehicles—and they are becoming more marked. The search for ever lower air resistance has resulted in low profile cars with fairly large areas of curved and steeply sloping glass, front and rear. When encountering low lying or drifting banks of fog or even early morning mist, the truck driver, generally sitting above such low banks of translucent fog, will be able to see reasonably clearly the general layout of the road ahead, as well as the shapes of tallish vehicles in front and to the side. He may, however, fail to notice the car in front and ride over it, crunching it in the process, for its driver, unable to see where he is and whether there are other vehicles ahead of him, will tend to put on his brakes—and they will be fast acting.

Braking performance of trucks is not as good as that of cars even when the respective systems are working perfectly.

The very real benefits to commercial vehicles which could be provided by anti-lock braking, in terms of better braking under adverse conditions and, above all, improved vehicle stability and directional control during emergency braking, have been well understood, but relatively few have been fitted to commercial vehicles, whereas it will not be long before some form of anti-lock braking will become available on high priced cars as well as modestly priced mass produced small cars. This will soon create an entirely new situation, in which the car driver can stop his vehicle in a controlled and safe manner, no matter what the weather, road or traffic conditions are or how rashly or inexpertly the driver reacts to them, whereas very few truck drivers will have the benefit of such technological quick-reacting devices.

For the wide range of possible surface textures, the tyre-to-road grip of bus and truck tyres will be around 2/3 or 3/4 that provided by car tyres intended for Western Europe. The minimum braking distance will, therefore, be 15% to 22% longer. In the foreseeable future there is little change of this basic performance gap being closed.

Current Trends

Road haulage, in its widest sense, is experiencing considerable and ever increasing competitive pressures. Hence operators will try to squeeze in an extra load or a longer trip whenever the opportunity arises. The new motto is "Time is Money" and since vehicles have "to earn their keep", many operators tend to keep their vehicles on the road, even when prudent preventive maintenance is called for.

Five key factors, arising from the need to meet ever tighter schedules, regardless of weather and road conditions, are affecting the transport scene:

1. More powerful engines and developments in transmissions and suspension allow trucks and coaches to reach and maintain quite high road speeds—also moderate gradients do no longer significantly slow them down. On many busy cross country roads in the UK, car drivers, therefore, find it ever more difficult to overtake without taking chances.
2. There are considerably more of the heavier and larger trucks about than only a decade ago—in particular there has been a marked increase in the number of articulated lorries, which in the UK and many parts of Europe have taken over from the railways as the principal load carriers.
3. An ever increasing rate of utilisation of motorways and interlinking main roads and the heavy loads carried has resulted in the rapid collapse of road surfaces and countless numbers of repair sections. In order to keep traffic flowing on motorways and dual carriageways while repairs are being carried out, many of them have been given contraflow sections, which create new accident hazards.

4. Although motorways are inherently safer than other roads, when accidents do occur, they tend to be more spectacular, with trucks and coaches crushing cars caught up in multiple shunts.
5. A significant number of heavy trucks move in tight convoy formation at speed, on trunk roads as well as through built up areas. This tends to irritate car drivers, for it makes it difficult, if not impossible, to overtake by leap-frogging trucks one at a time. It is not generally appreciated that such "appalling behaviour" by the truck drivers is a deliberate and defensive reaction to defeat organised crime. For single high value loads, ranging from registered mail or cash delivery to such readily disposable cargoes as cigarettes and spirits, attract the "villains". Driving in tight convoy formation tends to frustrate the well-established hi-jacking technique of infiltrating, often in broad daylight, a car, van or fast truck in front and another immediately behind the vehicle singled out for a hi-jack. The average truck driver has, undoubtedly and inevitably, become less tolerant of other motorists suddenly appearing out of his blind quarter—instinctively he will no longer yield ground. This is quite apart from the fact that hard braking might cause his load to shift or slew the vehicle, particularly so when the roads are wet and slippery and when spray interferes with his rearward field of vision.

Some Basic Considerations

What applies to the UK, need not necessarily be of equal relevance in the rest of the developed world, not even all of Europe, for the following reasons:

Britain has a very changeable climate and this can cause road traffic hazards to arise suddenly and unexpectedly. Though these hazards may be limited in spread, in intensity, duration and location, they can and do cause serious disturbances to safe traffic flow rates.

The methods and frequency of the Police enforcing legal limits on speed, vehicle loading and proper maintenance, as well as the penalties for flouting the Law of the Land, are such that many operators consider being caught and consequently fined to be no more than a sporting risk—and worth taking.

There is little prospect of better driver education—or exhortations by Police and Government spokesmen—improving the behaviour of *all drivers and not just a few* and thereby minimise the accident risk during the most hazardous combination of fast flowing traffic suddenly experiencing such changes in visibility as may be caused by swirling fog, heavy rain, falling snow and spray from trucks and coaches. For it is a well-documented fact that, like lemmings, many drivers have just one thought then—to get to their destination as fast as possible.

The fact that much of the trunk route network is chronically overloaded for prolonged periods is not uniquely a UK problem, nor are the morning rush hours with a reverse flow when millions of people leave work in the late afternoon and evening.

All future road building projects, design of vehicles and safety legislation ought to bear that in mind. Above all, we should, if at all practicable, make the systems of *all* classes of vehicles compensate for human failings and not improve one group of vehicles to such an extent as to make them relatively "super safe", while the others remain largely unchanged.

Major differences in basic accident avoidance characteristics are almost inevitable when one compares typical private cars with various other classes of motor-driven road vehicles, ranging from highly manoeuvrable two wheelers to often relatively cumbersome farm machinery, from vans or light trucks to "juggernauts",—not forgetting double decker buses and high-speed long distance coaches.

The key differences are in the fields of:

1. Driver's field of vision,
2. Conspicuity,
3. Manoeuvrability,
4. Stability, particularly under adverse conditions caused by wind and weather and those caused by variations in loading,
5. Occupant protection,
6. Load shifting restraints,
7. Structural integrity in various forms of accident,
8. Accelerating power,
9. Braking performance,
10. Handling characteristics.

The Driver's Field of Vision and Conspicuity

The search for even lower air resistance has resulted in low profile cars with fairly large areas of curved and steeply sloping glass, front and rear, slender pillars and maximum amount of side window glass. In general, the faster the vehicle, the lower its silhouette and hence, like a motorcyclist, such very fast cars can easily be missed by other motorists, unless they are alerted by the engine noise of the approaching fast mover. Where hedges fringe a typical cross country road, these readily screen such vehicles from view, whereas one can see that there is a high sided van, truck or coach rounding a bend and take appropriate action—steering and/or slowing down—to avoid a collision or scrape.

There are further effects of steeply sloping a windscreen:

1. The driver cannot see the four corners of his vehicle,
2. The low roof line severely restricts the driver's field of vision allround; so he can readily miss signals from a high sided truck or touring coach moving alongside,
3. Such curved laminated or toughened glass areas tend to lead to image distortion, undesirable reflections and glare, distracting and disorientating the driver, particularly so under bad weather conditions.

With an ever increasing use of non-rigid plastics to absorb or, at least, soften some of shock loads imposed on the occupants, should bodily contact occur during a collision, there will inevitably be some migration of plasticizer vapours towards the free surface and escape through it. These will then deposit themselves as a very tenacious thin layer on the inside of the glass areas. The rate and amount of build up of such undesirable thin films is accelerated by an increase of temperature of the surface of the plastic; the steeper the slope of the glass, the more of a hot-house will be generated inside the car by strong sunlight.

Smoking has a similar effect, for it will result in a thin film of a nicotine deposit; such a nicotine film will be additive to that due to the in situ final and progressive curing of the plastic and/or adhesives used in the manufacture and trimming of a car or a commercial vehicle's cab.

When a strong beam of light strikes a glass with such a thin film coating the inside, it results in a similar prismatic glare effect as that due to a smear from an ineffective or silicone loaded wiper blade on the outside of a windscreen with caked-on dirt. A series of haloes will then surround the shapes seen through the glass. It may create a distinct safety hazard not only at night with oncoming traffic causing glare, though its lights are on dipped beam, but also when the sun is low and near the horizon, or when the full beam of a truck approaching from the rear, after passing through the rear window, strikes the steeply raked front screen, turning it into a two-way mirror, with the driver of the sports car being blinded by the glare and unable to see out.

Just about the worst situation occurs when encountering low lying or drifting banks of fog or even early morning mist. The truck driver, generally sitting above such low banks of translucent fog, will be able to see reasonably clearly the general layout of the road ahead, as well as the shapes of tallish vehicles in front and to the side; this may encourage him not to reduce his speed drastically. He may, however, fail to notice the sports car in front and ride over it, crunching it in the process, for its driver, unable to see where he is and whether there are other vehicles ahead of him, will tend to put on his brakes and they will be fast acting.

The converse is also not uncommon—the classic under-ride case. The driver of the sports car, unable to see far ahead in adverse weather, will have been closely following the red rear lights of the vehicle in front of him. The front of the car can easily get wedged under the tail end of a truck when the truck driver unexpectedly and suddenly brakes hard.

The modern forward control truck or coach, with the driver sitting high up and well forward, gives much better vision—though not to the immediate rear of such a vehicle. This is a particular boon under conditions of low lying fog, rain or falling snow. There is less of the disorientation caused by the thin film deposits on the inside of the glass, for the windscreen has little rake.

Differences in Stopping Distances

The better forward visibility, given to a truck driver by modern design trends, allows him to see danger further ahead and thereby compensates in some measure for the truck's inherently inferior braking when comparing it with a modern car. But, when it comes to sideways vision, he does not have that plus factor.

This is most noticeable in the case of double decker buses. Also a new type of long distance touring coach is gaining popularity; it tries to combine the merits of single deck and double deck designs, yet is sufficiently low not to be inconvenienced by height restrictions of low bridges or road tunnels. In order to maximise the carrying capacity and profitability of such a coach, the driver is made to sit unusually low down; his field of vision can be inferior to that of a car driver's—and this could well be an explanation for some of them having become involved in spectacular multi-vehicle accidents in poor visibility.

Despite considerable efforts over the past decade by brake system suppliers and tyre manufacturers to close the performance gap, it still remains a fact that even the most advanced trucks cannot be braked to a stop in the same distance as just about any family car.

Added to this are two further factors;

1. The truck driver, conscious of the dangers to himself—and others in the cab—if the load he carries were to break loose and/or penetrate through the thin partition protecting his back, will, sensibly under these circumstances, not necessarily apply maximum brake pedal push right from the start of his attempt at stopping.
2. When such a truck runs into a car, then the remaining kinetic energy can be quite consid-

erable, sufficient to push the much lighter vehicle a considerable distance. Crushing and mangling of the car will be an inevitable consequence if there is not enough space between an obstruction ahead and the car so pushed along, despite the car's wheels being fully locked.

Discrepancy in braking performance is unlikely to be dramatically reduced by technological developments currently in hand.

Whereas at least some motorists appreciate that there might be inherent problems should the driver of a fully laden articulated truck be forced into a crash stop from over 60 mph, few realise that it might be even more difficult for the truck driver to achieve a straight-line pull up in a short distance if the trailer carries no load or only a very light one, particularly if that is located well forward.

In all situations involving emergency braking from speeds above about 30 mph, sensibly even weight distribution is important in the case of cars, but more so where trucks are concerned, particularly when there is a towed trailer, be it draw-bar or semi-trailer.

Braking: The Present State of the Art

It is worth remembering, when reviewing the current state of braking systems of millions of vehicles in regular use on our roads, that it costs a good deal of money, effort and time to fully develop effective modern systems and see them adopted by a vehicle or trailer manufacturer; thereafter they tend to have a relatively long service life and it takes many years to develop and evaluate a system which is significantly better and which finds acceptance by manufacturers and operators. A good example is the case of anti-lock braking:

The potential and very real benefits which could be provided by anti-lock braking, in terms of better braking under adverse conditions and, above all, improved vehicle stability and directional control during emergency braking, have been understood ever since the pioneering work of the TRRL and their encouragement of UK vehicle and component manufacturers some 15 to 20 years ago. The potential benefits were shown to the industry, likely users, the Authorities, Members of Parliament and the media—and that included spectacular demonstration on heavy articulated vehicles as well. So why have they not been more widely used?

Quite apart from the economic factor—currently it adds another £ 3,000 to £ 4,500 ($4,800 to $7,200) to the cost of a tractor-trailer combination—there may well be technical problems; for it requires that *both* tractor and trailer must be equipped with fully compatible systems. In practice this means dedicated tractor-trailer combinations.

Generally, it is still fairly rare in the UK and much of Europe to have comprehensive technological coordination between truck and trailer makers at the design and development stages. This omission must be seen against the background of many commercial vehicle and trailer makers finding it difficult to survive in a market where for several years a 35% to 40% overcapacity has been the norm and adopting cost cutting the preferred method of staying in business. It is not well known that despite that *the UK is setting a good example to the rest of Europe. For about 20,000 to 22,000 anti-lock braking systems have been installed on its heavy goods vehicles, mainly on such high risk or dangerous cargo vehicles as fuel and petro-chemical tankers and some fire engines. Most of the installed anti-lock systems are retrofit ones.*

When it comes to cars, it will not be long before several competitors will follow the recent example set by Ford of Europe of making some form of anti-lock braking available not only to those buying relatively expensive cars, but across their model range, right down to modestly priced mass produced small cars.

This will soon create an entirely new situation, in which the car driver can stop his vehicle in a controlled and safe manner, no matter what the weather, road or traffic conditions are or how rashly or inexpertly the driver reacts to them. On the other hand, very few truck drivers will have the benefit of such technological quick-reacting devices.

All anti-lock systems are designed to enable a vehicle to be braked in a straight line from a high speed and keep all of its full length inside a 3 1/2 m to 4 m wide lane during braking and when coming to rest. But not all systems allow the driver to steer and take avoiding action while the driver applies full brake pedal pressure.

The aim for commercial vehicles, including articulated ones and truck—draw-bar trailer combinations should be to retain steerability during emergency braking under all road surface conditions, including "split μ", rutted snow and other rough wintry situations. It is not an easy task—after all, several of the anti-lock systems fitted to car and their light commercial derivatives cannot meet such a criterion either.

Visualisation of the sequence of events, including the relative movements and the paths followed by the truck and its trailer during the braking and accident avoidance steering manoeuvre, can play a major role in shortening development time and high-lighting shortcomings and limitations of one system compared to another. Computer Aids to Engineering (CAE) may go a long way in convincing potential customers and possibly even Government Departments, but, of course, verification by actual physical testing will still be necessary—and to-date no agreements have been reached regarding realistic test procedures and an internationally acceptable set of manoeuvres and braking performance values.

So, in the meantime, the majority of commercial vehicle manufacturers design and build to *just* meet the currently mandatory brake performance tests as set out in particular National Type Approval Regulations. By their very nature, these will be a compromise, reflecting the *old-established* state-of-the-art.

Whereas, these days, no car maker could hope to get by—and actually sell his cars in the competitive market place—if the performance of his braking system were not very substantially better than the minimum laid down in such Regulations, this does not necessarily apply to commercial vehicles.

The not unreasonable fear of what adverse media comments could do to the sales prospects of a new model has done a great deal to "improve the breed" of cars. For there is no shortage of motoring journalists who are prepared to wear a flat or two on the tyres of a new car model they are evaluating, just to establish how good, bad or indifferent the car's stopping characteristics really are.

Heavy commercial vehicles are made in much smaller numbers and are not generally made available for such "evaluation" and far fewer journalists are really capable or interested in fully testing them.

The Importance of Load Sensing Brake Force Limiters

Car makers have done a great deal to achieve better, more level ride suspension, minimised bounce and wavyness in the interest of directional stability—and there are nowadays few, if any, makes with a pronounced "front end dive" during hard braking from high speed. Also they have come to accept that they must provide reliable and fool proof brake force limiters to the rear wheels.

It is, of course, relatively easy and not costly to incorporate such devices into the quick-acting all-hydraulic braking system, as fitted to a modern car, and thereby prevent premature and undesirable locking up of the rear wheels as weight transfer occurs during hard braking. Without such devices there is always the risk of the vehicle slewing when the rear wheels lock up before the front ones, for it is a fundamental feature of vehicle dynamics that the more vertical load is applied to a rolling tyre, the less prone the road wheel is to lock up during heavy braking; also when a wheel locks up, then all the sideways guidance forces developed by the tyre suddenly disappear.

Compared to commercial vehicles, the basic problems are easier to resolve on cars, because the ratio of overall weight of a car when fully laden, right up to the manufacturer's limit, to its unladen condition is of the order of 1.5:1 to 1.8:1.

This ratio tends to be 3:1 to 3.6:1 for heavy commercial vehicles.

It is even more, when considering a truck—drawbar trailer combination, for it could then be anything up to 8:1.

A variety of systems have been evolved by truck and trailer manufacturers, which can modulate the compressed air pressure to actuate the brakes on individual axles to allow for such a very wide range of possible loading conditions. But the majority of systems are not all that satisfactory in service, partly because they need more routine maintenance, but also for two other reasons:

1. Most trucks and trailers come from different manufacturers—and they often do not have the same component supplier,
2. Empty trailers have a distinct tendency to rhythmically bounce on their springs, for these were designed for maximum rather than minimum axle loading—and it is virtually impossible to achieve a proper brake effort control on a bouncing axle.

Ideally both the tractor and trailer should be equipped with a fully compatible anti-lock braking system which rapidly responds to dynamic movements of individual axles on their suspension as well as static axle loading. Currently this is still the exception rather than the norm.

An early adoption of anti-lock braking systems for at least the tractor unit of a tractor-semi trailer combination is a logical development to ensure improvements over current truck braking performance.

But it should be remembered that the anti-lock braking system does, generally, *not* give a shorter braking distance on a *dry* road surface than the current standard types of non anti-lock braking systems when the tractor-trailer combination is carrying maximum load and when all the axles carry their full share of vertical loading.

Steerability and Handling Stability of Tractor-Trailer Units

This involves so many very complex dynamic interactions that some over-simplifications are called for, to appreciate the principal behaviour patterns of current tractor-trailer combinations, including some with a form of anti-lock braking:

Steerability and handling stability can be roughly divided into four main categories.

The vehicle response characteristics are very similar for the following operating conditions:

1. Severe braking on a straight stretch of dry road,
2. Moderate to hard braking when the combined unit straddles a section of road where

part has a "grippy" surface and the rest a much lower tyre-to-road grip,

3. Moderate braking on a slippery road surface.

Category 1:

The wheels on the tractor's front axle lock first:

The combined unit remains stable, brakes in a straight line and cannot be steered.

Category 2:

The wheels on the tractor's driving axle or axles lock first:

The tractor-trailer combination becomes unstable to the point of initiating jack knifing. The driver may feel the onset of this just in time and, by easing off his brake pedal, then be able to minimise this inherently dangerous instability to just a large swerve; residual controllability on damp or wet roads can be maintained if the driver applies only moderate braking.

Category 3:

All the wheels on the tractor lock simultaneously—those on the trailer still roll:

The tractor-trailer combination is unsteerable, the trailer will swing up to 10 degrees out of line.

Category 4:

The wheels on the trailer's axles lock up—but not those on the tractor:

The combined unit remains steerable, the trailer will swing out of line. The lower the tyre-to-road grip or the weight carried by these trailer axles, the larger the angle of the trailer swing. Also the higher the initial road speed and/or the rate of deceleration of the vehicle, the greater will be the trailer swing.

If, for one reason or another, the braking effort distribution between individual tractor axles and wheels is such that the tractor will start to yaw or swing from side to side under heavy braking, then the tractor-trailer combination becomes unstable.

The undersirable behaviour of the tractor during hard braking may be due to the dynamic movement of twin rear axles, where fitted, on their suspension and/or too slow a response of the braking system to rapidly changing conditions—and that can happen even with early types of anti-lock braking systems. Poor maintenance can produce the same effect.

Braking in a Bend on a Wide Road

Only a few years ago, having to apply brakes suddenly while negotiating a bend at speed used to give real concern to car drivers, for it could cause the cars to either carry on in a straight line, tangential to the curve, or spin out of control. But steady progress with tyres and particularly with the latest types of anti-lock braking systems will substantially raise the safety margin and accident avoidance potential of cars.

Path deviation during emergency braking can already occur with tractor-trailer combinations while they are being braked from fairly modest speeds, particularly on damp or wet roads, it will be more pronounced when braking from higher speeds. *An unladen tractor-trailer combination is more at risk of that than one carrying a load—and the way such loads are distributed over the axles can have a considerable influence.*

Four basic conditions exist:

1. *The tractor-trailer combination has no anti-lock braking*: Tractor-trailer combination moves into the opposing traffic lane,
2. *An anti-lock braking system is fitted to the semi-trailer only*: Similar response to case 1,
3. *The tractor has anti-lock braking, the trailer has not*: Tractor remains stable, trailer moves into opposing traffic lane,
4. *Tractor and semi-trailer have compatible anti-lock braking*: Tractor-trailer combination remains stable, no path deviation,—shortest braking distance.

Design Improvements to the Tractor—Trailer Coupling

Improvements to the fifth wheel coupling, including strengthening, have virtually eliminated the dreaded phenomenon of the trailer breaking free during or following emergency braking and continuing on past the tractor. But in a tractor-trailer combination without anti-lock braking and with the tractor being a short wheel base type and the load concentrated at the rear of the trailer, it is then not uncommon for the tractor to nose dive during emergency braking, due to the front axle locking up, and the rear of the tractor jacking up, lifting its rear wheels clear off the road. This may or may not be accompanied by jack knifing, even a roll over.

Trailer Swing and Jack Knife Danger

Jack knifing occurs only under severe braking conditions. Trailer swing, on the other hand, can occur under both high speed towing and braking conditions.

None of the patented anti-jack knife solutions can totally eliminate trailer swing while the tractor-trailer combination travels at speed, particularly if the trailer carries only a light load—or none at all. Such devices, including the Hope anti-jack knife system, can, however, often inhibit the onset of jack knifing or, at least, minimise its severity.

Some quite ingenious systems have been evolved to minimise the tendency of lightly loaded or empty draw-bar trailers swinging from side to side while being towed at speed. But, under emergency braking, these have little effect on overall directional stability of the truck—draw-bar trailer combination, for the trailer will still swing out of line.

The only *effective* way to prevent the dangers listed above is to equip *both* the towing and the towed vehicle with a *compatible and very quick-acting micro processor controlled anti-lock braking system.*

Problems Associated with Anti-Lock Braking Systems

Some current types of truck brakes, though loosely described as of the "anti-lock type", have proved far from satisfactory in actual service. There are a number of reasons for that:

1. The moving parts of the complex braking system react too slowly in response to rapidly changing input signals,
2. The movement of axles on their suspension or of independently sprung wheels may be such as to send false signals to the electronic and/or pneumatic controls which regulate the admission of compressed air to actuate or momentarily free individual brakes. When severe braking is called for on a stretch of pot holed, broken up or loose surfaced road, the performance of such an anti-lock braking layout can be most disappointing; the same tends to apply to braking on very uneven roads and those covered in rutted snow,
3. Unacceptable and unpredictably variable delay in system response to sensor input signals may be due to the "sticking" of actuating mechanisms; this frequency occurs in cold wintry conditions and where corrosion has set in,
4. Under conditions of rain splash, slush and snow as well as high humidity—which may be due to mist and fog—some systems are prone to suffer from contact corrosion and/or short circuiting on wheel rotation sensors. The effect is that some sensors may then fail to "report" to the system's electronic brain,
5. Inability of electrical and electronic components to survive the arduous in-service shock loads, vibrations and severe temperature fluctuations,
6. Electrostatic and EMI (Electro Magnetic Interference) effects, which may originate from within the truck or the trailer's refrigeration plant, powerful radio frequency signals from a variety of stationary or mobile transmitters if the source is close enough,—when travelling in close proximity to other vehicles even some CB (Citizen Band) radio chatter can be troublesome. It is nowadays technically possible to *effectively* screen the complicated anti-lock system against most of these undesirable stray inputs, surge currents and interference—but this costs money.
7. Early applications of anti-lock brakes suffered from components giving unacceptably short in-service life, largely due to cost cutting and indifferent to poor quality.

All this—and the unfortunate history of some heavy truck anti-lock braking systems failing so spectacularly in the USA about a decade ago—calls for better engineering allround, a fail safe approach to the system, self monitoring and possibly duplication of some parts of the system. It may be easier to solve some of the problems by adopting multi-disc rather than drum brakes.

Commercial Vehicle General Braking Characteristics

Cars and their derivatives use a fairly simple and inherently self balancing hydraulic system, activated by direct application of pedal pressure, which on all modern cars is suitably assisted by a vacuum or hydraulic servo.

However, all-hydraulic brakes are not the norm for trucks. For light trucks—up to about 7 1/2 GVW—a pneumatic over hydraulic system is sometimes used.

The majority of trucks and buses are fitted with drum brakes, actuated by compressed air via linkages and these require regular and more frequent maintenance to keep them in perfect balance than do disc brakes fitted to cars.

The modern car braking system can achieve maximum braking efficiency in a small fraction of a second, for, unlike the systems used on trucks and buses, it has practically no lost motion in its mechanisms and does not require a substantial transfer of working fluid (compressed air), nor a complex system of valves to regulate the flow of the working fluid.

But in trucks and buses there will inevitably always be a delay between the driver physically depressing the brake pedal and the vehicle's brakes becoming fully effective. In many tractor-trailer combinations and where a truck pulls a draw-bar trailer, such a delay period will be quite noticeable—and it can take a second or two before peak brake performance is achieved.

Coefficient of Friction

It is conveniently assumed that the two friction coefficients which determine the rate of deceleration of a moving vehicle remain constant throughout the braking operation. It is demonstrably *not* true for rapid braking from about 60 mph or above. One friction coefficient is that between the rotating metal drum or disc and the friction lined component which is anchored to a non-rotating part of the vehicle; the other is that between the road surface and the tyre, which may or may not be rotating.

It takes a measurable amount of time before the brake linings and the metal faces they rub against reach the minimum temperature to allow them to develop their peak friction value, which may then drop slightly and reach a plateau value.

It takes longer to bring the mass of a truck's large cast iron brake drum to the required metal temperature than it does to stabilise the disc of a car.

Prolonged application of drum brakes can cause a progressive fall off in this friction value, in some cases this may be followed by a sudden and dramatic "brake fade".

Now that many countries have banned the use of asbestos in friction materials, a new situation has arisen. Many truck and coach operators will be tempted to fit readily available cheaper service and replacement parts, often originating from sources which still use asbestos. Sometimes such asbestos-based friction linings develop a higher friction coefficient than the OEM fitments, but this can be as undesirable as when they develop a lower value. It is, therefore, not uncommon to find that previously well balanced brake systems on a truck-trailer combination thereafter tend to be seriously out of balance when emergency braking is applied.

Repair garages are unlikely to refer to standard quality control tests for friction materials; though better than none, these tests are very basic, brief and, like the standard low speed brake performance dynamometer, unlikely to give a full representation of what is likely to occur under *real* life operating conditions.

The tyre-to-road grip is much more variable, largely due to (a) the weather and (b) traffic polishing and wear of the road surface; it is likely to vary substantially across the width of a traffic lane. It is quite common to find in the UK and Europe that repeated heavy truck traffic has created grooving, which in turn may hold rain water and thereby cause one of the pre-conditions for aquaplaning, the other being speed.

In really hot weather, the repeated passing of heavily laden trucks may cause the asphalt to "flow" like viscous treacle ahead of and under the rolling tyre; acting like a sticky lubricant, this can, in emergency situations, lead to extended braking distances. Under wet road conditions, such a truck tyre will plough through quite deep water, parting it and displacing it as a powerful spray with an astonishing range, most of it to the side.

Fundamental Differences Between Car and Truck Tyres

Because of the very much higher loading of their contact area, truck tyres tend not to be so disturbed by deep water as many low profile car tyres are.

The trend for ever lower profile car tyres is re-introducing problems which had been successfully tackled before their advent—namely low level of vehicle controllability when travelling at speed on rain soaked or snow and slush covered roads, for under those conditions the area of contact with the road provided by low profile car tyres is reduced to only a few percent of what it is when the road is dry.

When this "floating off" starts, the effectiveness of anti-lock braking, where fitted, is greatly reduced—and any form of accident avoidance possibility may be lost.

However, both car and truck tyres are at their worst when the roads have just a thin film of moisture, which can range from morning mist and fog to black ice and melt water on top of hard packed snow. For it results in a very poor tyre-to-road grip and hence affects accelerating and braking power, as well as steering and staying with the selected path. Despite considerable and sustained development work by the tyre makers to achieve better damp and ice grip, there is, within the foreseeable future, no prospect of a technical breakthrough—not even a marked improvement.

It is a basic fact of vehicle dynamics that the lower the downward-acting forces on a tyre—be they due to static weight carried, dynamic weight transfer during manoeuvring and/or braking, or aerodynamics—the less will be the control forces which such a tyre can provide, particularly those which determine whether the vehicle can be steered predictably.

This explains why a fully laden trailer will "track" as though running on rails and why the same trailer when not carrying a load will often tend to develop an outward drift in a tight bend or snake when being towed at speed.

In addition, there is the question of the "grippiness" of the tyre on the road. *For the wide range of possible surface textures, including traffic polished, broken and uneven or pot holed ones, those with loose gravel covering, dry, wet, damp, packed snow or combinations of these, the tyre-to-road grip of bus and truck tyres will be around 2/3 to 3/4 that provided by car tyres intended for Western Europe.* The reasons for this performance difference are the very different design and construction characteristics of car and commercial vehicles respectively and differences in rubber compounding.

In the foreseeable future there is little chance of this basic performance gap being closed, particularly as long as the all too prevalent—though little publicised—practice continues that, in order to be able to secure a sales contract, a "slight adjustment" is made by truck or trailer supplier on the final choice of tyres—which usually means substituting second line for first line tyres; these may also have a slightly narrower section.

In commercial vehicles, the loads carried per tyre and hence the internal air pressures are several times that applicable to car tyres, therefore these tyres must be given must greater sidewall stiffness and more reinforcing plies allround.

Also in a modern car tyre the flexing under load, be it due to acceleration, braking or sideways forces coming into play during changes in direction from the straight-ahead steady progress, plays a major role in giving enhanced tyre-to-road grip characteristics.

Car tyres destined for the West European markets are, generally, optimised to give good wet grip performance, a quiet ride and high comfort level, even if that means some sacrifice in terms of very high mileage being achievable. The designed-in performance characteristics of truck and bus tyres, particularly so in the case of second line tyres, come in the reverse order of priorities.

In practice the effect of all these factors is that, even if the truck or bus were to be equipped with a braking system equally as effective as that of the modern European type car, the shortest braking distance achievable would be 15% to 22% greater than that of the car under dry road surface conditions at speeds up to about 50 mph to 60 mph. And to-date trucks do not have such ideal braking systems.

At higher initial speeds and when the roads are slippery, the differences in the shortest possible braking distances to a complete stop are greater still.

These are fundamental facts and unlikely to change in the next decade.

Spray Generation

Considerable improvements can be achieved by shrouding the wheels down to near hub height and lining such wheel arches with textured material which then dissipates some of the kinetic energy of the high velocity spray or jets of displaced water as they leave the periphery of the textured tyre tread. While it cannot do much about deep water displaced sideways as a bow wave when travelling at speed and the spray this generates as high velocity jets of water mix with entrained air to form spray, it can deal with the considerable amount of surface water scooped up by the tyre tread and convert it into a steady stream of water directed to leave the trailing end of such a shroud with much diminished velocity.

It is possible to reduce by about 35% to 50% the vision obscuring effects caused by spray from trucks and coaches; but it complicates brake cooling and routine maintenance, adds a little weight and costs money, hence is not likely to be adopted by operators until compelled to do so by Regulations or their Insurers.

Enforcing safety standards costs money, so does to voluntarily fit spray suppressors. The effects of decontrolling a substantial sector of the commercial vehicle fleet—in the UK it is buses and coaches, in the USA it may cover an even wider range—can only worsen the situation, for it encourages competitive operators to do the bare minimum the Law demands of them; few will, therefore, care about a nuisance caused—after all it will affect others only. Additionally there is some evidence that several operators have resorted to cutting back on preventive maintenance;—and fitting cheaper second line tyres and cheap brake replacement parts fall into that category too.

Compatability Problems of Tractor-Trailer Brake Systems

While the peculiar way US truckers operate does not generally apply in Europe, it does have some relevance, as others may wish to follow it.

US truckers,—a large number of whom are self employed and own one or two "bobtails"—large and powerful sleeper tractor units—traverse the States, picking up a load here, dropping it off where called for and picking up another trailer or two as they progress. In terms of weight carried per axle and general brake performance, these tractors and trailers frequently are not even of similar or compatible types. The degree of braking effort matching of tractors and trailers of such combined units, to give optimum retardation and good directional stability whilst braking, can, therefore, leave a good deal to be desired, for their inherent performance characteristics can vary from one "rig" to another. The rig may be a "bobtail", which frequently is a three axle tractor unit, a four or five axle tractor-semi trailer combination, a Western Double, having five or six axles, in which a close coupled draw-bar trailer is attached to the rear of the semi trailer, to say nothing of the even longer seven axle Rocky Mountain Double and the nine axle Turnpike Double, in which the trailers are each 26 to 28 ft long; a few States allow triples, each trailer of up to 28 ft length and in Michigan State 16 axled vehicles are not uncommon.

Though the trucks which travel long distances right across Europe and on to the Middle East and the Gulf States are generally large tractor-semi trailer combinations, with a few Western Double or even Rocky Mountain type units thrown in where local legislation permits their use, there are also a considerable number of two or three axled rigids towing one or two trailers behind them.

There will be difficulty in achieving optimum braking performance even on uniformly well surfaced straight-line motorway sections. For it is not easy to get proper matching of the performance of the mutiplicity of compressed air brake actuating cylinders, their control valves and pressure modulating devices to suit *all* possible combinations of load and road surface grip conditions; additionally, under hard braking, there are the performance variations of different brake lining materials, including fade proneness of some.

When it comes to braking in a bend or where surface irregularities, unevenness and differences in tyre-to-road grip across the width of the carriageway occur, the problems are even greater.

Three Air Line Versus Two Air Line Braking Systems

As if this were not enough, a practice has arisen in which the UK has for a number of years been out of step with the rest of Europe—and it may take a further 7 to 10 years before these "odd" tractor and semi trailer units are being phased out of regular circulation.

It just does not make sense for the UK vehicle makers and/or operators to go it alone, even though some operators may, from time to time, express a preference for the three compressed air brake line arrangement rather than the two compressed air brake line system, which is the European norm.

In practice it is all too easy for an incorrect coupling of a trailer and tractor unit to occur when *either* are to the UK standard.

An improper functioning of the trailer braking, and hence possible trailer swing, are, therefore, the likely consequences. In extreme cases, it can lead to a jack knife situation, primarily when the trailer is only lightly loaded or empty. For one must never forget that, apart from Company owned dedicated tractor-trailer combinations, there are a substantial number of loose trailers crossing the Channel in either direction.

Not long ago the UK Department of Transport, the SMMT and other interested groups issued publicity material which the Department called "three into two can go", to tell truck drivers how to modify their tractor or trailer and/or introduce special pneumatic and electrical couplings to allow them to operate with either three or two air lines. Unfortunately the leaflet assumed a degree of litaracy which is hardly the hallmark of foreign or many British truck drivers.

In the case of a two-line tractor towing a three-line semi trailer, there should be no real problem, the trailer's blue (secondary) air line is simply left unconnected. In the case of a three-line tractor towing a two-line trailer, provision has to be made for the tractor's secondary brakes to operate the trailer brakes—otherwise a malfunction of the (main) service brake on the tractor would mean that the trailer brakes could also not be applied.

If the truck driver remembers to have a fourth coupling handy and so assembles it that the secondary line pressure can be fed back into the service line, then all is well.

If he does not, then the sort of catastrophic run-away accident can take place, which so enrages the media and pressure groups that they forthwith call for the banning of foreign truck drivers and/or heavy goods vehicles in their towns and villages.

Differences Between UK Construction and Use Regulations and EEC Directives

The major difference between these two is that, in the case of articulated vehicles, the EEC Braking Directives do *not* require the trailers to be fitted with a secondary braking system, for it is quite logically argued that a trailer cannot run on the road unless coupled to a towing vehicle and that, provided the prime mover and trailer braking system respectively are each protected against a failure in the other, a coupled outfit has, in effect, a dual braking system.

The tractor, which may at times run without a trailer being attached to it, must, however, have a secondary brake.

Five additional features are required by the EEC legislation, aimed at preserving satisfactory breaking performance of the tractor-trailer combination in the event of partial or total failure of the (primary) service brake, or a leak in the connector or part of the system or rupture of the air line supplying it.

There are basically three braking systems currently in regular use on our roads:

1. The traditional UK system,
2. The mixed UK-EEC system,
3. The EEC system as operated in e.g. W. Germany.

Just what does it all mean in a typical four axle articulated truck with a tractor unit having one steered and one driving axle?

In the traditional UK system, only the steered front axle of the tractor and both the trailer axles are braked when the secondary system comes into play. This leads to loss of steerability of the combined unit.

In the mixed UK-EEC system all four axles would be braked. Whilst this may result in a shorter braking distance, the driver would be unable to steer out of a potential collision.

In the West German (EEC) system the secondary braking system would *not* apply brakes on the steered front axle, thereby giving the driver the ability to control the movement of his "rig" and avoid obstructions as he approaches them.

Of equal significant is that, when it comes to parking:

In both the traditional UK and the mixed UK-EEC systems the brakes will only be applied to the two tractor axles and *none* to the trailer.

In the West German (EEC) system both the driving axle of the tractor and the two axles of the trailer have their brakes applied by compressed air. In strong winds or under slippery road conditions, this may help to hold the combined unit or the trailer against an involuntary drift.

Also when a trailer complying with the EEC regulations is left on site, i.e. is disconnected from the tractor unit, its compressed air reservoirs immediately apply the trailer brakes; this can be a valuable safety feature when, for one reason or another, a trailer becomes disconnected from its towing vehicle or has to be temporarily dropped off.

Primary and Secondary Brake Performance

Modern private cars have an all-hydraulic secondary brake system. Should the primary circuit fail, this gives a much better performance and shorter braking distance than the all-mechanical handbrake of old could ever hope to provide.

Since trucks could wreak havoc if there were a partial, let alone a total failure of the main (service) brake, some provision for a reliable secondary braking system is essential.

Braking performance of trucks is not as good as that of cars, when the respective systems are working perfectly, but there is even more a discrepancy when the primary brake has failed and the driver has to rely on the secondary braking system.

For there is a *considerable* difference in braking performance when comparing the commercial vehicle's secondary with its service or main brake.

It is a mandatory test requirement under ECE Reg 13 that in both the laden and the unladen condition the commercial vehicle must be capable of being stopped by the service brake within a braking distance of no more than 51 m (168 ft) from an initial speed of 50 mph and 71 m (234 ft) from 60 mph. When there is a partial failure in either the tractor or trailer line and the secondary brake comes into operation, then the corresponding distances must not exceed 93 m (307 ft) and 133 m (439 ft).

These minimum brake system requirements may well be marginal when the driver has to brake hard to a stop from 65 to 70 mph, let alone from the higher maximum speed which heavy trucks can and do achieve.

Even the latest amendments to the European Braking Regulation ECE 13 only demand that the tractor unit main brakes do not run out of their stored working fluid (compressed air) for about 23 seconds after the brake pedal is applied and that the secondary brakes will still perform adequately after up to 8 full-stroke applications of the brake pedal.

In the case of the towed vehicle, the compressed air reservoir capacity need only be capable of ensuring that the trailer brakes will function for at least 15 seconds.

European vs. UK Heavy Truck and Coach Driving Techniques

Partly to compensate for marked differences in terrain, some European operators tend to rely on an auxiliary retarder being fitted to their heavy trucks and particularly to touring coaches; these devices will stabilise their speed on long downhill descents. One popular device is an eddy current electric retarder, which can relieve the service brakes of much of the effort required in controlling the vehicle's speed. The driver can modulate the degree of retardation over a fairly wide range by a simple hand control of a rheostat type—not unlike the controls on trams and electric tube trains—and, by anticipating the road conditions ahead, progressively slow down the vehicle to a crawl without ever touching his service brake.

The degree of maximum rate of retardation which can be achieved by this means is about one third of that which truck and coach drivers would normally obtain by applying regular braking, or about one sixth of the maximum which can be achieved under ideal conditions on a dry level road when applying full emergency braking.

For coaches and rigids, it is convenient to locate the eddy current retarder to operate via the drive line, by offering a steady "drag", i.e. the brakes as such are not actuated.

In the case of articulated vehicles, it is more common to have a pneumatic assistance counterpart, be it vacuum or compressed air actuated, which can be made to act on one of the non-driven trailer axles. It is then not uncommon for some types of retarders to apply the trailer brakes in a similar manner to a partial application of the hand brake. This demands that such trailers be fitted with more fade-resistant friction linings than are the norm on UK built trailers.

Some of the apparently "inexplicable" run-away touring coach accidents which occur, from time to time, on mountain routes may be due to the following combinations of circumstances:

Whilst on a normal run over a familiar route, where he can anticipate when and where to slow down, the driver hardly ever applies the service brake. For he can achieve progressive deceleration by using the retarder only. This lack of use then causes the mechanical parts of the service brake mechanism to be stiff to the point of partial seizure due to corrosion. It has been established on a number of occasions that this is what actually happened and that the operators were unaware of the "creeping malfunction" of the service brake system, for it is perfectly possible to let the retarder perform the braking function.

When the driver finds that he has to slow down more quickly on a twisty or steep scenic route and therefore applies the main service brake, it may either fail to respond, or, more likely, the braking effort may be uneven, causing the vehicle to slew out of control and plunge over a precipice.

Though it is claimed as a distinct commercial benefit to fleet operators, it is false economy to "save" by extending the life of the brake linings and concentrate instead on the utilisation of the electric retarders.

Also, *if an electric retarder is retro-fitted to a vehicle that has a basically well-balanced foundation braking system, it could readily lead to overbraking of the driven axle and hence introduce directional instability, particularly when applying, in a bend, emergency braking via the service brake in addition to that already provided by the retarder.*

The Lack of Harmonisation of Truck Regulations

There is no international agreement yet on many key design and performance standards. The various countries of the EEC, not to mention other European countries and other major producers and users of commercial vehicles, all have quite a way to go before even such relatively simple but basic dimensions as maximum length, weight and permitted load per axle are fully standardised. The reasons for this lack of harmonisation are mainly political rather than technical—not least amongst them having to please pressure groups in each sovereign state; these can range from environmentalist to manufacturers lobbies.

The UK, in an effort to come in line with the rest of the EEC, increased in May 1983 the maximum weight limit from 32.5 to 38 tonnes and the minimum number of axles for that load from four to five. But Italy and the USSR permit 40 tonnes; it is 42 tonnes in Holland, 44 tonnes in Denmark, 48 tonnes in Finland, 50 tonnes in Norway and 51.4 tonnes in Sweden.

When it comes to truck and draw-bar trailer combinations or road trains, the picture is even more confused. The UK has stayed at the 32.5 tonne limit. In the majority of other European countries, with the sole exception of the USSR, the weight limit for a road train is not less than that for an articulated lorry. In fact it is 44 tonnes in Italy and 50 tonnes in Holland.

The road haulers have long been lobbying for a 40 tonne limit for articulated lorries throughout Europe and 44 tonnes for road trains.

Contrary to popular perception, allowing heavier lorries on UK roads does not mean longer vehicles. They will not take up more road space than those complying with the old UK Construction and Use Regulations—nor will they cause more road damage, for the load will be carried by more axles. *It is generally relatively easier and economically more attractive to fit to these "juggernauts"—where they will be of most benefit—the by no means cheap but latest technology and often quite complex electronic anti-lock braking systems and better axle movement controls.*

Such vehicles inherently command a higher unit price tag, since it has been demonstrated that, taken over a period of two years or more, they can give a better return on money invested, for they tend to be more efficient on a basis of cost per ton-mile of goods delivered.

Heavier vehicles are likely to be given better all-round safety features than lighter ones. Also there will be fewer vehicles on the road to carry the same load, hence the likelihood of traffic conflict situations arising is reduced.

The Roundabout "Squeeze"

However, there remains a basic traffic flow problem which will not be affected by improvements to the suspension and braking systems; it is the "jockeying" for the best position at sharp turns, traffic islands and roundabouts. Some car drivers tend to deliberately squeeze their cars in between moving trucks on roundabouts and motorway slip roads. It demonstrates a selfishness by car drivers and, above all, a fundamental lack of appreciation of the inherent handling limitations which apply to trucks, buses and coaches.

There are detail differences, in terms of wheelbase and overhang, between various makes of trucks and coaches, between articulated lorries—which predominate in the UK—and the rigid truck towing a draw-bar trailer of a length comparable to the towing truck, which is favoured by many freight operators on the Continent.

But they all "sweep" a much larger proportion of available road space than a car does. However much a coach or truck driver may wish to avoid pushing an overtaking car off the road or squeezing it against an obstruction, there is very little he can do to prevent it, once he has committed his vehicle to negotiate such a relatively tight bend.

It is really up to the car driver to hold back until coach or truck and trailer are again fully on the straight section of the road.

Any action by the car driver which might cause the truck driver to brake suddenly is likely to cause some trailer snaking, which would aggravate the situation further.

Articulated Trucks: Critical Speeds at Typical UK Roundabouts

There are some speeds which lead to a rapid and irreversible loss of control by the driver; this often occurs without the driver being aware of it even whilst it happens. For typical UK sized roundabouts these critical speeds, which lead to the trailer in an articulated truck rising off the tyres on the inside of the curve and then tipping right over, tend to be as low as 25 mph on the traditional roundabout and only 12 mph on a mini roundabout. The exact values depend on the radius of the island. In practice, a band of about 3mph either side of this calculable critical speed can lead to a trailer overturning.

Snow, slush, heavy rain and strong cross winds and, above all, the way a driver cuts a corner, tend to modify the exact values of these critical speeds. The tighter the bend, the lower the actual values of these critical speeds.

Overturning Trucks

Various attempts have been made to improve the stability of high sided vehicles in curves and when subjected to strongly gusting winds. But all have proved too costly or impracticable.

Conclusions

Currently, the level of advanced technology applied to commercial vehicles, from light trucks to road trains, from local buses to long distance touring coaches, is about a decade behind that to be found on many mass-produced cars—and the new generation of family cars will have "masterminds" to co-ordinate automatically and electronically many of the thinking and vehicle control functions, thereby giving such vehicles a greatly enhanced accident avoidance capability. But all this will *not* minimise the risk of such cars and their occupants being badly mauled by commercial vehicles, which have none of these quick-acting devices.

While no one wants to hold back technical developments on private cars, the issue must be addressed by *all* concerned with road safety of whether this will further widen the already substantial gap in terms of accident avoidance capability which exists between cars and the vast majority of commercial vehicles.

Is it really desirable to concentrate so much research funding and technological effort towards making cars go ever faster, have better traction and basically "idiot proof" braking and possibly even incorporate a form of proximity radar to trigger of the braking function, while leaving most commercial vehicles with somewhat dated "state of the art" braking and suspension systems and no effective spray suppression devices?

Should not some of the resources be redirected to make coaches and commercial vehicles better equipped to cope with the ever more demanding requirements of fast moving road traffic? Resources—and that means money and skilled people—are not unlimited.

In the end, it may well become the responsibility of politicians rather than engineers to influence the order of priorities. Could one not begin with making the existing road network, road junctions and roundabouts safer for heavy goods transport?

An Investigation of Selected Vehicle Design Characteristics Using the Crash Avoidance Research Datafile

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Abstract

Problem identification efforts have been hampered in the field of crash avoidance research by the lack of a database tailored to the information requirements of this approach to crash prevention and injury reduction. A prototype database designed to overcome the deficiencies of existing crash databases has recently been developed. Initial analyses of the information contained in this database suggest it is a viable source of information for identifying problems in the general area of crash avoidance research.

Introduction

Crash data have traditionally functioned as a source of information for identifying highway safety problems, assessing their magnitude, developing potential solutions, and evaluating their effectiveness. Perhaps the best known analyses of crash data have been those which have enhanced our understanding of the crashworthiness design characteristics of motor vehicles and their role in injury production. The knowledge derived from these investigations has ultimately led to the development of a number of improvements in vehicle design, which have in turn led to reductions in the severity of injuries sustained in motor vehicle crashes.

Efforts to employ existing crash data in support of similar activities in the field of crash avoidance research have proven less successful, in part because existing crash databases have been developed to satisfy the information needs of crashworthiness research, which differ somewhat from those of crash avoidance. In general, currently available databases have been found to be deficient in one or more of the following respects:

Size—Available databases do not contain enough observations to permit meaningful analyses of the relationship between specific vehicle design characteristics and particular types of crashes.

Representativeness—Many of the crashes in existing databases are old, and thus do not reflect the crash experience of currently manufactured vehicles.

Content—Available databases contain little if any data indicative of the movements of involved vehicles immediately prior to crash.

Any of these deficiencies is sufficient to obviate the utility of a given database for crash avoidance research. A lack of information concerning the precrash movements of vehicles does, however, represent the most significant shortcoming. Without this information, it is difficult to use crash data to explore the relationship between specific vehicle design characteristics and crash propensity. It is also difficult to establish a precise estimate of the degree to which a proposed improvement in vehicle design might affect crash involvement. For example, without some knowledge of precrash movements, one cannot examine the role of rearward field of view in crashes preceeded by a lane change maneuver. Similarly, the potential value of improvements in the detectability of front turn signals cannot be assessed without knowing one vehicle was turning left in front of another immediately prior to crash.

In an attempt to overcome these and other shortcomings in existing crash databases, NHTSA's Office of Crash Avoidance Research recently undertook an effort to develop a crash database specifically geared to the information needs of crash avoidance research. The specific objectives of this effort were to:

1. Identify the information requirements for a crash avoidance database.
2. Identify a source of crash information which could satisfy these information requirements.
3. Develop a prototype version of the database.

The result of this effort is the Crash Avoidance Research Datafile, or CARDfile, a detailed description of which is contained in Edwards (1987). The information contained in this database has been extracted from the automated police accident reports of Texas, Maryland, Michigan, Washington, Pennsylvania, and Indiana. Information pertaining to these crashes is contained in three files: Accident, Vehicle, and Driver, the contents of which are enumerated in Table 1.

Table 1. CARDfile data elements.

Accident	Vehicle	Driver
Day of Crash	Crash Type	Age
Month of Crash	Make/Model	Sex
Year of Crash	Model Year	Alcohol/Drug Use
Time of Crash	Vehicle Type	Restraint Use
Number of Vehicles	Component Failure	Helmet Use
Crash Severity	Precrash Stability	Driver Error
Accident Type	Avoidance Action	
Light Conditions	Injury Severity	
Weather Conditions	VIN	
Road Surface		
General Land Character		
Primary Impact		
Location of Impact		
Relation to Intersection		
Intersection Signaling		
Roadway Alignment		
Roadway Profile		
Roadway Separation		

Perhaps the most unique data element contained in CARDfile is Accident Type, which is used to describe the precrash movements of vehicles involved in each of the crashes contained in the database. A total of 43 accident type codes exist, describing not only the overall movement patterns associated with the crash, but the specific movements of each involved vehicle. The coding scheme employed is based on work originally accomplished by Perchonok (1972) and Terhune (1983). As has been indicated, this information is of particular importance to crash avoidance research because it makes possible the development and testing of hypotheses related to the role of specific vehicle design characteristics in crash involvement. For example, using CARDfile, it is possible to examine the frequency with which vehicles equipped with amber turn signals are struck in the rear end while turning, and compare this involvement to the frequency with which vehicles equipped with red turn signals are struck in similar situations. As another example, it would be possible to examine the extent to which vehicles equipped with front wheel drive are involved in rollover crashes. In fact, any vehicle characteristic common to a particular make and model of vehicle can be evaluated with respect to its involvement in a particular type of crash and compared with the involvement experience of vehicles with different or similar characteristics.

CARDfile Characteristics

At present, CARDfile contains data on slightly fewer than 4,000,000 crashes involving almost 7,000,000 vehicles, which represents the police reported crash experience for the most recent three years of data available. A summary of the three year crash experience (1983-1985) for each of the states in this database is presented in Table 2. In total, the crash data contained in CARDfile represents slightly more than 20% of the nation's annual police reported crash experience.

The value of CARDfile as a database for relating vehicle design characteristics to crash propensity becomes apparent when examining the number of specific types of vehicles contained in the file. As shown in Table 3, the available number is so large that even vehicles which represent a small minority of crash

Table 2. Summary of state crash experience (1983-1985).

State	No. Crashes*	No. Vehicles*
Indiana	480,399	854,571
Maryland	384,450	717,284
Michigan	1,023,297	1,724,210
Pennsylvania	414,210	694,854
Texas	1,341,415	2,326,103
Washington	338,307	617,093
Totals	3,982,078	6,934,115

* Cumulative three year experience.

Table 3. Distribution of vehicle types.

Type of Vehicle	Frequency	Percent
Passenger Car	5,007,284	72%
Light Truck/Van	1,120,866	16%
Heavy Straight Truck	152,053	2%
Heavy Articulated Truck	135,287	2%
Motorcycle/Moped	106,148	2%
Bobtail Truck	16,673	<1%
Transport Bus	17,902	<1%
School Bus	15,728	<1%
Police/Emergency	29,818	<1%
Other/Missing	335,356	5%

involved vehicles, for example, school buses, exist in numbers sufficient to permit meaningful analyses of their crash experience. Heretofore large scale analyses of the crash characteristics of these vehicles were virtually impossible.

The sensitivity of CARDfile to vehicles of recent manufacture is illustrated in Table 4, which depicts the crash experience of all vehicles contained in CARDfile by model year manufactured after 1969.

As is apparent, more than half of the vehicles in CARDfile manufactured after 1969 are no more than 8 model years old, and slightly more than 25% are no older than 5 model years. The desirability of obtaining crash data on vehicles of recent manufacture is made even more important by the recent changes in size, engine/drivetrain placement, and other design characteristics of newer vehicles.

Comparability of CARDfile

Although no attempt has been made to create a statistically representative sample of the nation's crash experience, the states comprising CARDfile were chosen to be as representative as possible of the various geographies and regions of the country. Comparisons of CARDfile with the National Accident Sampling System (NASS), a nationally representative sample of police reported crashes suggest that, on the whole, the crashes contained in CARDfile are typical of the nation's crash experience. An illustration of this comparability is presented in Table 5 which contains a comparison of selected CARDfile variables with equivalent NASS variables.

As is readily apparent, there are few real differences between CARDfile and NASS with respect to these particular variables, suggesting that conclusions reached on the basis of CARDfile analyses might be extrapolated to the nation as a whole at this general

Table 4. Distribution of vehicles for model years 1970-1986.

Model Year	Frequency	Percent
1970-72	500,127	8%
1973-75	907,550	15%
1976-78	1,554,431	25%
1979-81	1,623,744	26%
1982-84	1,392,393	22%
1985-86	216,637	4%

Table 5. Comparisons of CARDfile data with NASS data*.

Variable	NASS	CARDfile
Single Vehicle Crashes	35%	35%
Multivehicle Crashes	65%	65%
Fatal Crashes	.7%	.6%
Property Damage Crashes	61%	61%
Pedestrian/Bicyclist Crashes	4%	3%
Large Truck Crashes	4%	4%
Motorcycle Crashes	2%	2%
Alcohol Involved Drivers	7%	7%
Wet Weather Crashes	13%	15%

* Based on Information in 1984 NASS Annual Report.

level. A comparison of injury severities for both NASS and CARDfile is presented in Table 6.

An indepth effort to establish an accurate measure of the comparability of CARDfile with respect to the nation's crash experience has recently been initiated by Salvatore, Mengert, and Walter (1986). In this study, distributions of selected variables are being compared for the aggregate of states comprising CARDfile with various national estimates. To date, the results of these comparisons indicate that, for the most part, CARDfile does not differ substantially from the nation with respect to general driver, roadway, and vehicle characteristics. In addition, many of those differences which do exist involve characteristics not likely to obviate the utility of CARDfile as an aid to problem identification and countermeasure development in crash avoidance research. For example, imported light trucks are underrepresented in CARDfile. While this difference might affect the accuracy of estimates of the number of injuries resulting from imported light truck crashes, it would not likely effect the results of any analysis of the relationship between specific imported light truck design characteristics and crash involvement. More indepth analyses of the comparability of CARDfile and national crash experience are presently underway. These analyses will focus on the comparability of specific univeriate and multivariate CARDfile crash characteristics.

Pilot Applications

Although CARDfile is still under development, a number of preliminary analyses have been initiated to acquire experience with the file, and to establish some estimate of its utility as an aid to problem identification and countermeasure development in crash avoidance research. These applications have addressed such issues as rear turn signal color, front turn signal location, and other issues relevant to the general area of crash avoidance research. A detailed summary of the results of these and other analyses is presented in the following sections.

Table 6. Distribution of crashes by severity.

Crash Severity	NASS Frequency	NASS Percent	CARDfile Frequency	CARDfile Percent
Property Damage	3,772,706*	64%	884,919	64%
Possible Injury	756,049	13%	222,343	17%
Nonincapacitating Injury	756,474	13%	182,221	14%
Incapacitating Injury	369,797	6%	62,196	5%
Fatal Injury	33,396	.57%	8,542	.65%
Unknown Injury Severity	219,808	4%	488	.04%

* National estimate of police reported injury severities for 1984.

Amber Turn Signals

Two years of crash data, containing approximately 407,000 rear end crashes in which a passenger car was the struck vehicle, were utilized to assess whether or not passenger cars equipped with amber turn signals were less likely to be struck while turning than cars equipped with red turn signals. Two, rather than the three years of CARDfile data available, were used in order to match the relevant crash experience with the number of vehicles registered in each CARDfile state as a means of controlling for any differences in overall exposure which might exist between the two groups of crash involved vehicles. Registration data is available for July to June time periods only. Thus, only crashes occurring between July 1, 1983 and June 30, 1985 (a two year period) were considered in this analysis.

Only crashes involving passenger cars manufactured in 1980 or later were considered. Crashes involving these vehicles represented slightly less than 14% of rear end crashes during the time period examined. Rear end crashes considered relevant for this analysis were those in which the lead vehicle in a rear end crash was in the process of turning prior to impact.

Three measures of rear end crash experience were employed:

1. Ratio of relevant rear end crashes to total rear end crashes.
2. Ratio of relevant rear end crashes to total crashes.
3. Ratio of relevant rear end crashes to registered vehicles.

The resulting calculations of relevant rear end crash rates for the first two measures of crash experience are presented in Table 7. Data provided by manufacturers were used to determine whether the struck vehicle was equipped with red or amber turn signals. Vehicles for which turn signal color was unknown were excluded from the analysis. All other things being equal, any differences in these ratios should be indicative of a "turn signal color" effect.

Table 7. Analysis of relevant rear end crash experience.

Signal Color	Ratio of Relevant Crashes to:* Total Rear End Crashes	Total Crashes
Red	4203/21,399 (20%)	4203/192,796 (2.2%)
Amber	5638/33,673 (17%)	5638/297,936 (1.9%)

* Tabulations exclude Michigan

Both comparisons indicate that passenger cars equipped with amber turn signals are less likely to be struck while turning than cars equipped with red turn signals. Approximately 20% of rear end crashes involving vehicles with red turn signals occurred when the vehicle was struck while turning, as opposed to 17% for vehicles equipped with amber turn signals. Similarly, the proportion of relevant rear end crashes to total crash involvements is higher for this group (2.2% as compared to 1.9%). Relative percent differences for each of these comparisons are 15% and 14% respectively. Both are significantly different at the $p<.01$ level using a Chi Square "Goodness of Fit" test in which the proportion of relevant crashes involving vehicles equipped with red turn signals was used to predict the expected number of relevant crashes involving vehicles equipped with amber turn signals.

Differences in the percent of relevant crashes favoring cars equipped with amber turn signals, as opposed to red, were also found in each of the states comprising CARDfile, with the exception of Michigan which was excluded from this analysis because of the effort required to identify make/models using VIN for the extremely large number of make/models and model years considered in this analysis. Michigan does not code make/model directly. As can be seen in Table 8, the findings in each state parallel those obtained for CARDfile as a whole, indicating that the differences noted are not due to results obtained in a single state.

Disparities in the percentage of relevant rear end crashes among states appear to be due to differences in reporting criteria, states with low reporting thresholds recording a greater number of relevant crashes than states with high reporting thresholds. For example, Pennsylvania only reports crashes in which at least one vehicle was towed, whereas Texas reports all crashes with a total damage cost of at least $250. Since many of the relevant crashes in this analysis are low in severity, more would be reported in Texas than Pennsylvania.

The comparison of involvement rates per registered vehicle for relevant rear end crashes also yielded differences in favor of amber turn signals for each of the 5 CARDfile states included in this analysis, as can be seen by inspecting the data in Table 9.

A Signs Test to determine the likelihood of obtaining these results as a consequence of chance yields a probability of .031, confirming the findings from other analyses of CARDfile data that passenger vehicles equipped with amber turn signals are less likely to be rear ended while turning when compared to passenger vehicles equipped with red turn signals.

An analysis of the rear end crash experience of Chevrolet Chevettes, which changed turn signal color after model year 1979, also yielded results paralleling those obtained in the present analysis. Chevettes equipped with amber turn signals experienced a ratio of relevant to total rear end crashes of 19.70% during the time period encompassed by this analysis, whereas Chevettes manufactured with red turn signals were found to have experienced a ratio of relevant to total rear end crashes of 20.17%. While these differences are not statistically significant, they are in agreement with earlier analyses which produced significant differences in favor of vehicles equipped with amber turn signals.

Table 8. Analysis of relevant crash experience by state.

	Ratio of Relevant Crashes to: Total Rear End Crashes		Total Crashes	
State*	Amber	Red	Amber	Red
Indiana	13%	14%	1.3%	1.4%
Maryland	8%	10%	.7%	.9%
Pennsylvania	2%	3%	.2%	.3%
Texas	23%	28%	3.1%	3.8%
Washington	10%	11%	1.07%	1.18%

* Tabulations exclude Michigan.

Detectability of Front Turn Signals

The crash experience of a selected sample of passenger vehicles with differing placements of front turn signals was evaluated to determine whether or not there were differences in the frequency with which they were involved in crashes preceeded by a left turn in front of oncoming traffic. Such differences, if present, might be suggestive of a relationship between the detectability of front turn signals and crash involvement.

Vehicles selected for this analysis were those in which front turn signals were integrated into the headlamp assembly for some model years, and mounted in the front bumper assembly in others. It was hypothesized that turn signals mounted within the headlamp assembly might be more difficult for oncoming drivers to detect than those mounted in a more isolated location, and thus vehicles designed in this manner would be more likely to be involved in crashes where the driver was turning left in front of oncoming traffic.

The analysis of turn signal detectability was limited to specific make/models which had undergone changes in the location of front turn signals to

Table 9. Analysis of registration data by state.

State*	Percent of Relevant Crashes per Registrations: Amber Turn Signal	Red Turn Signal
Indiana	0.0524%	0.0577%
Maryland	0.0458%	0.0519%
Pennsylvania	0.0057%	0.0073%
Texas	0.2185%	0.2331%
Washington	0.0695%	0.0726%

* Tabulations exclude Michigan

Table 10. Distribution of left turn crashes.

Location of Turn Signal	Turning*	Crash Involvements Total Multi Vehicle	Ratio
Close To Headlamps	2465	50019	.05
In Front Bumper	2832	62976	.04

* Crashes in which case vehicle was turning

provide some control for differences in driver characteristics which might effect turn signal use, risk taking, or other aspects of driver behavior which might effect the probability of crash involvement.

Three years of CARDfile (1983-1985) were utilized in this analysis. Only crashes in which a study vehicle turned left in front of an oncoming vehicle were considered relevant. The number of multivehicle crash involvements for the 4 models of vehicles considered in this analysis totaled 112,945 for the three year period analyzed.

The number of relevant crashes and total multivehicle crashes for each of the two groups of vehicles is presented in Table 10. As is apparent, the group with front turn signals mounted in the more isolated location experienced proportionally fewer crashes (10% difference in ratio of relevant to total crashes between the two groups). A Chi Square test utilizing the proportion of relevant to total crashes for the "headlamp" group to predict an expected value for the "bumper" group yielded a value of $p<.01$.

Field of View from Vehicle

The crash experience of vehicles with differing fields of view was examined to ascertain if such differences might be reflected in crash rates. Field of view measures were obtained for 14 vehicles using a method developed by Ziedman et al (1987).

This method derives a figure of merit (FOM) for visibility which is a measure of the degree to which a target vehicle is visible to the driver in certain prescribed scenarios where field of view might be important. All openings available to the driver for seeing, including rear view mirrors, are considered in deriving this measure of visibility. Field of view values for each of the vehicles considered in this analysis appear in Table 11. The greater the numerical value, the better the field of view.

The correlation between field of view and total multivehicle crash experience for this group of vehicles was - .73, indicating the better a vehicle's field of view, the lower its crash involvement. Analyses of the relationship between field of view and more specific precrash scenarios in which field of view is thought to play an important role are planned for the near future.

Although these results provide evidence of the validity of this method for quantifying field of view, they should not be interpreted as providing conclusive proof of the relationship between field of view and crash involvement.

Table 11. Field of view metrics.

Make/Model	Model Years	Field of View Metric Windows	Windows & Mirrors
Ford Thunderbird	1983-85	83.4	85.6
Chevrolet Camaro	1979-80	87.8	87.3
Mercury Capri	1979-83	84.5	82.0
Ford Thunderbird	1977-79	81.2	82.7
Mercury Cougar	1983-84	82.2	86.4
Oldsmobile Toronado	1982-84	84.9	84.6
Mercury LN7	1981-83	83.2	83.6
Chrysler Cordoba	1975-79	82.6	83.1
Mercury Capri	1985-86	81.9	85.5
Chrysler Cordoba	1980-83	86.1	88.3
Chevrolet Camaro	1982-85	89.1	90.4
Buick Regal	1977-81	87.1	87.1
Ford EXP	1982-83	78.8	81.6
Pontiac Grand Prix	1982-86	85.8	88.2

Conclusions

Given the analytic experience to date, it would appear that CARDfile contains sufficient information to permit detailed evaluations of the relationships between selected vehicle design characteristics and crash experience. While the results obtained do not provide conclusive evidence of such relationships, they do nevertheless suggest areas of research in which it may be possible to obtain improvements in the crash avoidance design characteristics of vehicles which result in lowered crash frequency and reduced injury severity. It was for this purpose that CARDfile was established.

Although analyses of CARDfile data are not appropriate for establishing cause, the large number of variables available for analysis does allow for the statistical control of major variables known to effect exposure, and consequently, crash involvement. The value of CARDfile lies in the capability it provides for examining the statistical relationship between specific design characteristics and crash propensity as a means of identifying and directing research programs in crash avoidance research where issues of cause may be more appropriately addressed.

Immediate plans are to continue the development of CARDfile with emphasis on assessing its comparability with the nation's crash experience, and developing more sophisticated analytic techniques which permit the widest possible control of potentially confounding variables. Efforts are also underway to refine existing data elements, improve descriptions of roadway characteristics at the crash site, and add additional states.

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Study in Avoidance of Road Accidents with the Aid of Computer Simulation of Accident-Relevant Driving Manoeuvres

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Abstract

On the basis of control loop components, it is explained which occurrences of accident-relevant driving manoeuvres can lead to road accidents. These manoeuvres are simulated with the aid of the driving dynamics simulation model MEDYNA. In extreme situations, the variations in the vehicle parameters lead to an unstable behaviour, which as a rule precedes an accident, meaning that limit values and/or areas representing the criterion for study of an accident, or rather, of its avoidance, can be established. The accident-relevant driving manoeuvres, the described quantities of the actions introduced by the driver, such as braking, accelerating and steering, as well as the spectrum of vehicle-specific parameters are deduced from the analysis of the material provided by the accident research group of Medizinische Hochschule Hannover and Technische Universität Berlin (MHH/TUB). The initial results on the avoidability study, which were achieved in a research work sponsored by the Bundesanstalt für Straßenwesen, are presented.

Presentation of the Problem and Approach to Avoidability Considerations

All activities in the field of vehicle safety are carried out from the aspect of influencing the accident occurrence. The following questions therefore arise in considerations on the avoidability of accidents:

- how do accidents come about (research into accident causes)?
- what circumstances lead to accidents and what parameters can be used to describe these circumstances?
- how can the circumstances which cause accidents be avoided; i.e., what value must the parameter not be allowed to assume?

The answers to these questions inevitably lead to the necessity of analysing the accident occurrence with regard to its causes, and of following the development of accidents, so that the sensitivity and limitations of the parameters describing the actual state can be determined by introducing a widely applied variation of these parameters (Figure 1).

The accident occurrence only includes a negative selection of all those cases in which an accident-relevant driving manoeuvre did not lead to the mastering of a critical situation, but resulted in an accident in the form of incurred damages and injuries, respectively. Those cases in which accident-relevant driving manoeuvres were able to be mastered are near-accidents, but these are not included in any compilation of accident data (even though every driver of a motor vehicle can report from his own experience on one or another near-accident). In our study on the avoidability of accidents we are dealing, however, with the question of what conditions lead to the occurrence or non-occurrence of an accident when a critical situation arises—i.e., what parameters distinguish an accident-relevant driving manoeuvre from a pre-crash phase?

From this, we can conclude that indications of accident-relevant driving manoeuvres can be deduced from an analysis of the accident occurrence but that the questions concerning the mastering of a critical situation can only be answered by simulation of the accident-relevant driving manoeuvres. This can, however, only be carried out using individual cases as examples. To do this, it is necessary to deduce development patterns of accident-relevant driving ma-

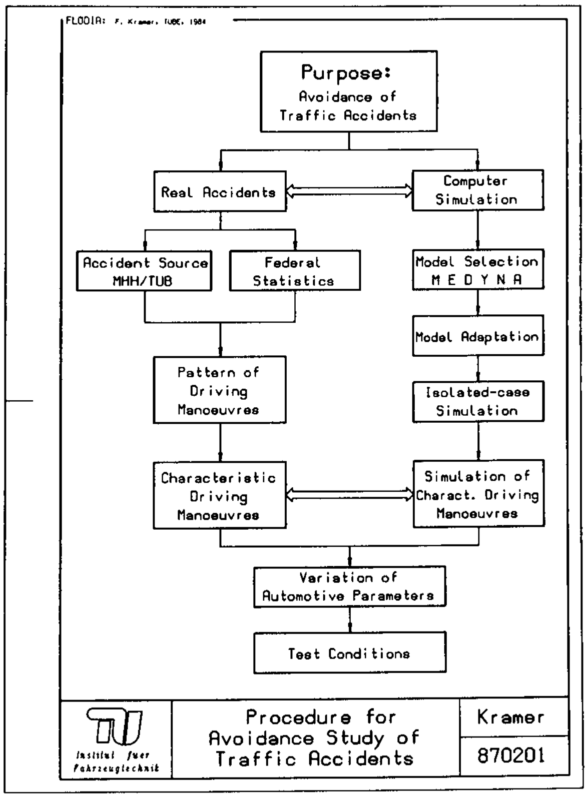

Figure 1. Procedure on avoidability study of road accidents

noeuvres on the basis of accident data and then combine these into characteristic driving manoeuvres. These driving manoeuvres are then simulated.

Experimental simulation is undoubtedly useful or even necessary, but in the variation of the automotive parameters it is very material and time-consuming. Apart from this, only the actual state as determined by the respective construction can be examined. For this reason, the method of mathematical simulation was chosen in the course of studies up to now. This requires the availability of a suitable computer program and also extensive adaptation work. Under these conditions, which can be regarded as fulfilled in the first iteration step, the mathematical simulation is an extremely useful aid in reconstructing experimental trials; it also provides a cost-favourable application for trend statements in the variation of accident parameters. In addition, one can deduce indications on the characteristics of vehicle parts and components of which only the mathematical description exists and have not been realized as a manufactured component.

For this, we must first determine what the accident-relevant driving manoeuvres are and then condense these to characteristic driving manoeuvres in order to be able to achieve effective and economical simulation. By simulating these characteristic driving manoeuvres we aim to deduce automotive parameters and develop test procedures for evaluating the primary safety of motor vehicles[1].

The Development of an Accident

The Accident within the Control Loop "Driver/Vehicle/Environment"

Here, we shall add yet another control loop description to the large number which already exist; this time to explain an accident-relevant driving manoeuvre: the relationship between vehicle and environment is conveyed to the driver by means of optical, acoustic and palpable information. On the basis of his perception, and backed by his experience, the driver decides to carry out an action if this seems necessary to him on the basis of the information he has received. Upon recognizing a deviation from the driving objective, the driver will therefore introduce an action in order to minimize this same deviation or in order to contain the deviation within certain limits which still allow him to fulfill his driving objective. This control process will take place once or several times depending on the requirements (Figure 2).

During this process, each single component of the control loop can be affected by disturbances characterised by the fact that they are unpredictable, and thus occur unexpectedly, and apart from this, affect the control task with an intensity which is at least noticeable. The occurrence of such an impairment is described therefore as a critical situation.

In order to avoid a critical situation, the driver undertakes a suitable manoeuvre (steering, braking, load cycle of engine or a combination of these) which at first seems suitable to him for compensating the impairment. If the disturbance cannot be compensated, the critical situation leads to instability of the control loop and an accident occurs. The manoeuvres undertaken by the driver in order to eliminate a

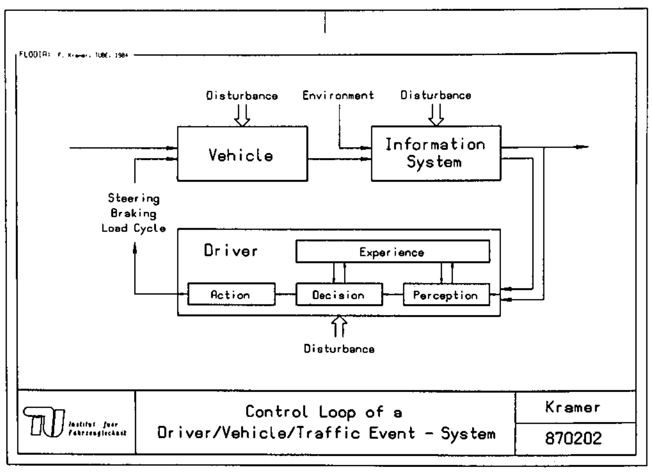

Figure 2. Control loop of the system "driver/vehicle/environment"

critical situation, and irrelevant of the success or failure of the actions introduced, are described as accident-relevant driving manoeuvres.

If, however, an accident occurs, then the time between recognition of the critical situation and the actual accident is described as the pre-crash phase.

Accident Development Patterns

The pre-crash phase is introduced by the perception of an impairment or by the reaction demand of the driver. This can take place due to

- self-overestimation of the driver (e.g. the speed chosen is too high for the following road section),
- the behaviour of the vehicle (e.g. loss of tire adhesion or technical defect) or
- environmental influences (e.g. character of the road or traffic situation).

The fact that the human being possesses no reflex which can be described as being compatible to the vehicle control elements makes it clear that every impulse has to be conveyed to the human brain. Here, the impulse is identified, compared with any existing experience, evaluated and then converted into an appropriate muscular activity. The time between perception and action can be reduced by training and repetition of danger situations. In the following general description of the processes, we shall, however, assume a cortically-controlled processing of impulses. (If necessary, individual quantities can be reduced to zero if certain automatic functions are presumed to exist.) Immediately after the time span for the processing of the impulses, the driver exerts a muscular reaction to operate a control element. Due to the strongly differing and also varying condition of the driver, the muscular reactions are not only limited by the speed of contraction of the muscles but equally by the muscle power which has to be exerted.

The consequence of the muscular reaction is the operating of a control element. Because of the principle of the control element, the reaction of this to the action of the driver is not direct, but is subject to a time lag due to elasticity and to fabrication tolerances and wear tolerances. In the further development, the reactions of the vehicle, which are a result of the changes made to the control elements, are conveyed to the driver, also with a time lag, since the dynamic qualities of the transmission elements and also the resiliences which result, for example, from the tires as connecting element between vehicle and road surface, lead to delays in feedback. This means that, for the driver, the recognizable consequence from the completed action may be regarded as a new reaction demand, this making a repetition of the processes necessary.

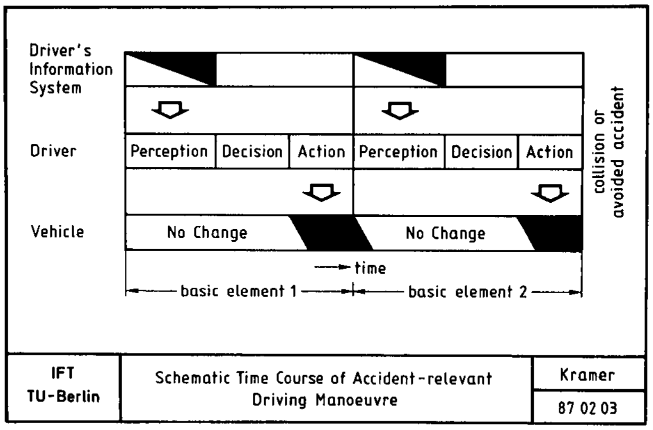

Figure 3. Schematic time course of an accident-relevant driving manoeuvre

The sequence of events from one reaction demand up to the renewed demand is defined here in the form of a model, as the basic element of the development pattern[2]. Two different developments in the actions of the driver can be derived, depending on the respective situation, this being determined by the characteristics of driver and vehicle as well as by environmental influences: One possibility of averting a critical situation can be the decision to take action, i.e. to activate a control element. Following this, either the driver waits for the effect of the introduced measure in the assumption that he has prevented the collision, or the occurrence of a collision makes any further avoidance action superfluous. An emergency braking measure could, for example, be regarded as a situation with this development. The course of the accident consists of one single basic element.

The second possibility is that the basic element is repeated once or more due to the accident situation. This possibility is conceivable in cases where the driver tries to avert the critical situation by corrective steering manoeuvres. In Figure 3, the developments pattern of an accident-relevant driving manoeuvre consisting of two basic elements is shown as a diagram.

Analysis of the Accident Occurrence

The analysis of the accident occurrence serves two basic purposes:

- to determine and select relevant driving manoeuvres in order to define characteristic driving manoeuvres, and
- statistic evaluation of parameters required for simulation and their range of values.

As we have already established, the sequence of events of accident-relevant driving manoeuvres cannot be shown directly, since the general use of crash recorders in vehicles has proven to be extremely problematic, mainly for legal reasons. For this reason, the developments have to be derived from the pre-crash phase of documented accidents. This necessary step of equating an accident-relevant driving manoeuvre with the course of a pre-crash phase can be regarded as permissible, due to the presumption that the same course of action is taken by the driver from the initial situation up to aversion of the accident, or, in the second case, up to the accident actually happening.

From the Accident-relevant Driving Manoeuvre to the Characteristic Driving Manoeuvre

In the Federal Republic of Germany, all accidents reported to the police are compiled in the form of a table in the so-called Bundes-Statistik—this is the numerical data on the road traffic accidents which is evaluated and published by the Statistisches Bundesamt (Federal Statistics Office)[3]. In these statistics the traffic accidents are differentiated into seven types and these are divided into two groups: within city boundaries (intra-urban) and outside of city boundaries (extra-urban) and also with regard to their severity. On the basis of this, it is possible to determine, in economical terms, the costs resulting from injuries in the various types of accident, which can be shown in comparison to each other. Since, however, more detailed circumstances and information on the course of the accidents—which we shall go into in more detail—are lacking, actual analysis of accident occurrences is based on the considerably more detailed documention material on accidents provided by the accident research group MHH/TUB (financed by the Bundesanstalt für Straβenwesen).

The correlation of these two sets of material by means of the common criteria: type of accident and severity of injury (expressed as costs resulting from injuries), for the purposes of comparative study, as

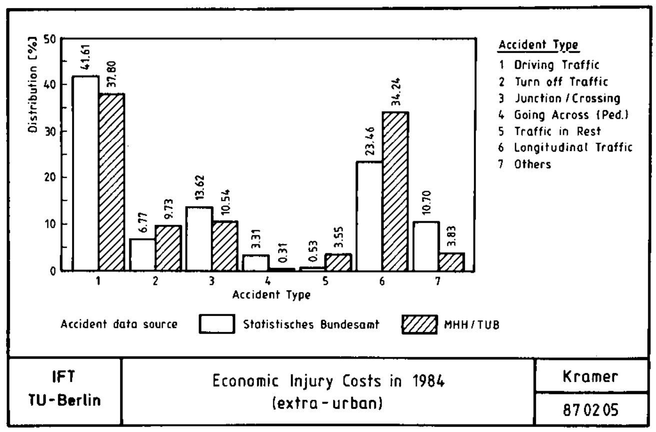

Figure 5. Distribution of the economic costs resulting from injuries in 1984 (extra-urban)

well as a limitation to the accident year 1984 makes it possible to establish the relevant type of accident inside and outside of city boundaries. Figures 4 and 5 show the percentual distribution of the costs resulting from injuries in various types of accidents inside and outside of city boundaries. Despite the considerable difference in the number of cases recorded (Bundes-statistik 1984: 476,232, MHH/TUB-material 1984: 646 injured persons) and the philosophy concerning data collection practised at this time, but which has meanwhile been revised, by the accident research group MHH/TUB (mainly serious accidents were recorded), the correlation between the respective definitions can be considered as sufficiently exact. The comparison quite clearly allows the conclusion to be made that

- inside city boundaries, the categories of accidents caused by "crossing pedestrians" and "junction/crossing" are prevalent, and
- in areas outside of city boundaries the categories of accident "driving accident" and "longitudinal-traffic accident" cause the highest costs resulting from injuries.

The further statistical evaluation was based on total material comprising 2,753 accidents. However, in order to be able to ensure a comparable safety standard, the accident material had to be leveled out and older vehicles (up to the year of construction 1972) eliminated; apart from this, only four types of vehicle of an exemplary character were selected.

In an evaluation of the remaining accidents material with a total of 290 accidents, the respective actions of the driver, such as steering and braking, can be coordinated with the relevant types of accident; the vehicle reactions swerving and deceleration, which can be determined from accident marks, serve as comparison and control quantities. Engine-load cycles could not be established as driver activity in the accident material. Apart from this, the evaluation includes the

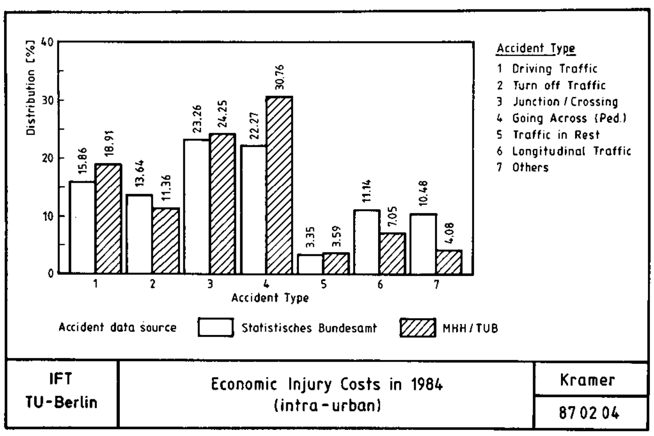

Figure 4. Distribution of the economic costs resulting from injuries in 1984 (intra-urban)

Table 1. Relevant driving manoeuvres in critical traffic situations (source of the accident research group MHH/TUB).

Relevant Accident Types	Braking [%]	Steering [%]	Braking & Steering [%]	Speeding [%] [km/h]
Driving Traffic (extra-urban)	39,5	65,1	20,9	44,2 up to 60
Longit. Traffic (extra-urban)	59,4	34,4	21,9	24,0 up to 60
Going across (intra-urban)	76,3	15,8	2,6	53,6 up to 65
Junct./Cross. (intra-urban)	56,1	21,1	14,0	38,6 up to 65

reconstructed vehicle speed as well as the amount by which speed limits were exceeded, if this was the case.

The accident-relevant driving manoeuvres (in the actual sense: the development of the pre-crash phase) therefore consist of the driver activities of braking and steering (singly or in combination) during critical situations, which can be described by means of a total of four types of accident at certain speeds (a survey is given in Table 1). Measured on the basis of the costs resulting from injuries, these form 64.6%, or approximately two thirds of all types of accident (driving accidents: 21.4%, longitudinal-traffic accidents: 19.4%, pedestrian crossing accidents: 13.3% and junction/crossing accidents: 10.5%). Here, we presume that the frequency distribution of the accident-relevant driving manoeuvres in which we are interested and the development of the pre-crash phases as documented in the accident material are approximately alike.

By adapting the accident-relevant driving manoeuvres to one or more basic elements of the development pattern, it can be seen that the mastering of critical situations with the individual stages

- straight-line motion
- transition curve and
- circular motion

superimposed with the driver activities

- braking
- steering or
- braking and steering

and with time shares which nevertheless differ strongly to some extent, can be described sufficiently accurately. The accidents of the relevant categories stated above vary merely in their intensity and, where applicable, in the sequence of the elements mentioned, e.g. straight-line motion and braking without curves or curves with varying transitions and radii. With respect to existing test procedures, but also the mathematical simulation of driving manoeuvres, provision of the characteristic driving manoeuvres in the form of

- the steady-state circular motion, with or without braking, and with varying road excitation, adhesion, curve radius and driving speed, and
- the steering angle input due to varying steering angle functions, e.g. step function, single sine or continuous sine functions with varying steering angle speed and amplitude, and with varying road excitation, adhesion and driving speed,

can be considered to be sufficient in the evaluation of the automotive parameters. The fact that the straight-line motion is not taken into consideration here is due to the fact that, for example, pedestrian crossing accident, primarily parameters other than those relating to the technology of the vehicle might prevent the critical situation, since, in most cases, this accident takes place within a very short interval, namely between the reaction demand and the occurrence of the accident.

In determining the characteristic driving manoeuvres we consciously abandoned the wish to take into consideration the regulatory influence of the driver. The reason for this is to be found in the objective of the current study, in which first of all the critical automotive parameters and evaluation criteria for the primary safety of motor vehicles are to be determined and/or confirmed.

Road-building Parameters and Automotive Parameters

Purposeful simulation requires that the parameters to be varied correspond to reality.

The road-building parameters form part of the environmental influences describing the characteristic driving manoeuvres. Thus, from the accident material, we obtained curve radii between 100 and 1,000 metres, whereas the radii of the vehicle paths are considerably smaller; these lie between 40 and 200 metres. The supposed connection between the radius of the vehicle path and the driving speed could not be established; the speed at the point of reaction demand lies at between 30 and 80 km/h inside of city boundaries and between 80 and 120 km/h outside of city boundaries. As expected, the collision speed is usually lower due to the driver activities and is up to two thirds lower within city boundaries and up to one third lower outside of city boundaries. As opposed to this, in the case of a crossing-over accident, the speed reduction is considerably less than in other accidents occurring within city boundaries, despite an over-proportionally large number of braking operations; this again indicates the shortness of the time span available for driver activities.

It is not possible to deduce the influence of the road surface on the course of the accident, since in approximately 85% of all cases the surface is described as "plane". The location of possible disturbance factors in the form of bumps, grooves, tracks etc. relative to the accident spot are not shown in more detail, so that the possible influence of such factors on the occurrence and on the course of accidents has to be ignored. Finally, the adhesion between tires and road should be mentioned: in approximately 70% of all accidents, the road surface was dry and in 25%, the surface is described as "damp" or "wet". The remaining approximately 5% of the accidents is distributed among road surfaces with a covering of ice, snow and frost.

The quantities which can be regarded as automotive parameters are those which are to be varied in the simulation of certain characteristic driving manoeuvres. This concerns mainly, in the first step, the vehicle design concept, the size and type of tire, the number of occupants of the vehicle as well as the loading. In order to be able to take into consideration the wide variety of documented types and models of vehicle, four types of vehicle class were selected according to net mass, drive layout and frequency of vehicles involved in accidents, to represent the total number of passenger vehicles included in the accident material. These are illustrated in Table 2.

In the evaluated accidents, no specific striking features could be established in the data on the tires. The effects of the difference in type of tire used in the front axle and back axle, and which could be determined in individual accident vehicles are negligible.

Table 2. Selected vehicle types which represent all accident vehicles (source of the accident research group MHH/TUB).

Vehicle's Layout	Distribution [%]	Unloaded Weigth [kg]	Powered Axle [--]	Engine Arrangement [--]
Typ A	17,7	750 - 810	front	front end
Typ B	17,1	860 - 920	front	front end
Typ C	14,7	1.340 - 1.485	standard	front end
Typ D	5,7	990 - 1.180	standard	front end

As regards the number of occupants of the vehicle, differences could be established, as expected, between the accident inside and those outside of city boundaries: whereas within city boundaries the vehicles are occupied in 62% of all cases by one person and in 28% by two persons, outside of city boundaries only 52% are occupied by one person but 29% by two persons. The greatest variation is to be found in the extent of occupation of more than two persons: the percentage outside of city boundaries is 18% as opposed to 10% inside of city boundaries. The frequency of loading in the front of and on the roof of the accident vehicles is negligible both inside and outside of city boundaries. As opposed to this, the loading in a baggage compartment located at the back of the vehicle amounts to up to 200 kg in 37% of the accidents occurring extra-urban; intra-urban, the loading amounts to up to 150 kg in 18% of all cases. The loading in the passenger compartment also varies between the accidents occurring inside and outside of city boundaries, namely in 32% of the cases up to 150 kg, and in 27% of the cases up to 60 kg, respectively. The amount of fuel in the tank which is given in gradations of 0.25, 0.50, 0.75 and 1.00, varies insignificantly intra- and extra-urban.

Computer Simulation

The Vehicle Dynamics Program MEDYNA

The used program system MEDYNA was developed by the company MAN Technology, the Deutsche Forschungs- und Versuchsanstalt für Luft- und Raumfahrt (German Research and Experimental Station for Air and Space Travel) and other partners, with participation of the Technische Universitt Berlin commissioned by the Bundesminister für Forschung und Technologie (Federal Minister of Research and Technology)[4]. This program serves to simulate three-dimensional mechanical multi-body systems of which the individual bodies can be linked by means of various connection elements (spring, dampers, rubber elements, hinged rods, bending rods, expansion rods and torsion rods)[5]. As application of the program was at first limited to rail-borne vehicles, a tire model and a vehicle-guidance model was introduced (based on the theories according to [6] and [7]), so that the range of application could be extended to the simulation of road vehicles.

The mechanical substitution systems to be simulated can be determined in an interactive dialogue; in this way, rigid and also flexible bodies and coupling elements can be combined in a modular form to create an overall model. With the aid of an internal equation modeller, the required equations of motion are constructed by the program by means of a multi-body formalism. As model excitation, deterministic time and path functions, stochastic processes and excitation processes displaced in either time or space can be entered, and these must then be converted into user specific subprograms in the form of force relationships. The excitation can be applied to the bodies, kinematic guidance and to the connecting elements.

On the output side, static and dynamic analyses in the time and frequency domains can be carried out with the aid of the program system MEDYNA. To obtain a solution with respect to time, the complete system of differential equations is solved and transferred into the next time interval by means of numerical integration (RUNGE-KUTTA method with automatic step length control). Output of the results of the various motion quantities are available in graphic mode, both as function of time and frequency. Since the recent incorporation of the high-resolution graphics system with calligraphic display (vector screen), a "Multi Picture System" by the EVANS & SUTHERLAND company, the calculated results can also be shown as an animated display.

Description of the Modelled Vehicles

The passenger vehicles to be examined are modelled as systems with six elastically sprung masses: a mass of the fully equipped car body, four wheel-carrier masses and the mass of the steering system. The mass elements for the vehicle occupants, for the loading and for the tank contents remain as variables with regard to position and size and are connected rigidly to the superstructure (Figure 6).

By superstructure we mean a combination of the completely-equipped bodywork, the engine/gearbox block and of the percentual suspension parts. Each individual wheel-carrier mass consists of the mass of the wheel support, the brakes, the rim and the tire as well as the percentual mass of the drive shaft, the differential, the suspension link, the spring and the shock absorber. In the model, the individual masses are idealised in such a way that they are assumed to be concentrated at one point, namely at the centre of

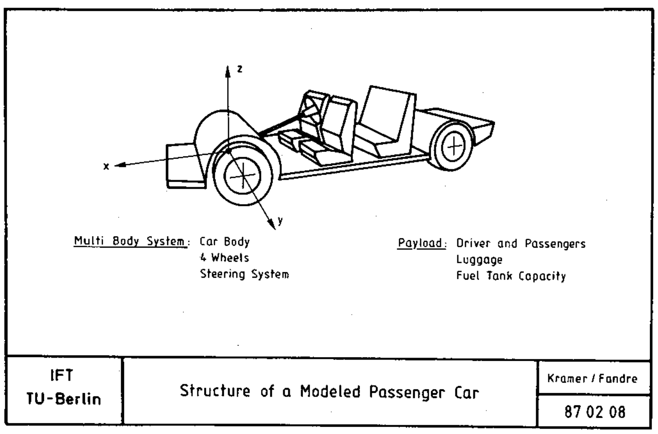

Figure 6. Structure of the modelled passenger cars

gravity. Centre of gravity, mass, moment of inertia and the geometric data of the nodes for the linking of the connecting elements as well as the description of the motion possibilities of the node, i.e. the degree of freedom, must be stated for each body. Coupling of the mass-bound bodies takes place either by means of linear spring- and damper characteristics or non-linearily by means of special force relationships, which must be specifically defined. The connecting element "tires" presents a special case here due to its complex behaviour in comparison to other connecting elements, and has therefore been introduced into program system MEDYNA as a separate subprogram[8]. In procuring the required data for the model, we were able to make use of the kind support of the VOLKSWAGENWERK AG and CONTINENTAL Gummiwerke AG companies, who made their internally determined test data and measuring data available for the vehicle simulation. These data were adapted to the requirements of the model and compiled in data sets which the program could read.

Time-dependent input data are provided as simulation functions for powering, braking and steering. The drive torque acts on the tire and, if necessary, can assume a different value for each wheel. A further input parameter is the moment of inertia of the drive train which participates in the motion and which depends on the respective transmission ratio. Similar to the treatment of propulsion, different time functions can be allocated to the effective braking torque on the individual tires, so that in the simulation, for example, brakes which are pulling unevenly, or at a later point antiblock systems can also be taken into consideration.

In order to carry out a steering manoeuvre, a steering torque which is a function of the vehicle path must be determined to act about the steering axis (kingpin) of the front wheels. The calculation of this torque is carried out in a subprogram specially developed for this purpose and in which the vehicle guidance model is converted by means of the program.

The road excitation is described by the roadway topology and the surface quality expressed in values for upward or downward slope of the road and by unevenness of surface in the form of spectral power density, respectively. Simulation with stochastic roadway disturbance can at the moment only be carried out for a linear vehicle model. For calculation in the time domain, it is necessary to introduce the unevenness as the determining excitation.

Figures 7 and 8 show a number of simulation results for driving in a left-hand curve with a radius of 100 metres and at a speed of 36 km/h as a function of time: Figure 7 shows the steering torque acting on the kingpin as a result of which a steering angle is established at the steering axis with a small time delay. Following this, vehicle roll angle and sideslip angle as system response to the steering torque are shown.

In Figure 8, again based on the kingpin torque, the torque about the z-axis as well as the force in y- and z-direction in the contact area of the front, left-hand tire are traced.

Work is being carried out at the moment on a comparative study between simulation and experimental results with the aim of proving the quality of simulation. Since we have not yet carried out any tests of our own, the comparative study must for the time being refer to published results.

Simulation of Characteristic Driving Manoeuvres

The simulation of characteristic manoeuvres is to be illustrated on the basis of steady-state circular motion at a speed of 36 km/h. A smooth road with a dry surface is assumed (adhesion coefficient: 0.80). The vehicle model, type A (refer to Table 2) is occupied by the driver and one front-seat passenger (mass: each 68 kg) and at first with no additional loading; the tank is half-full and therefore has a mass of 20 kg. In order to maintain the steady-state circular motion, the radius of the vehicle path is set to 100 metres; the driving speed shall be varied yet up to 100 km/h. Because the guidance model is regulated by the vehicle path, the steady-state circular motion is attained at the conclusion of a curve entry within a certain permissible range. This also corresponds furthermore with the conditions which apply in the experimental execution of this test procedure.

In order to be able to evaluate the curve behaviour of the vehicle model, Figure 9 shows the steering torque about the king pin, the steering angle, the roll angle and the sideslip angle as a function of simulation time. The same evaluation criteria are applied to these results, however with different loading conditions of the vehicle model. The vehicle now contains four occupants with a mass of 75 kg each, two sitting in front and two at the back. The mass of the additional load in the baggage compartment is 200 kg, and in the passenger compartment are another 150 kg. The contents of the tank have a mass of 40 kg; this is the equivalent of a full tank. The comparison shows that, due to the increased number of occupants and the higher load, the driving behaviour shows a tendency to oversteer. This becomes particularly apparent

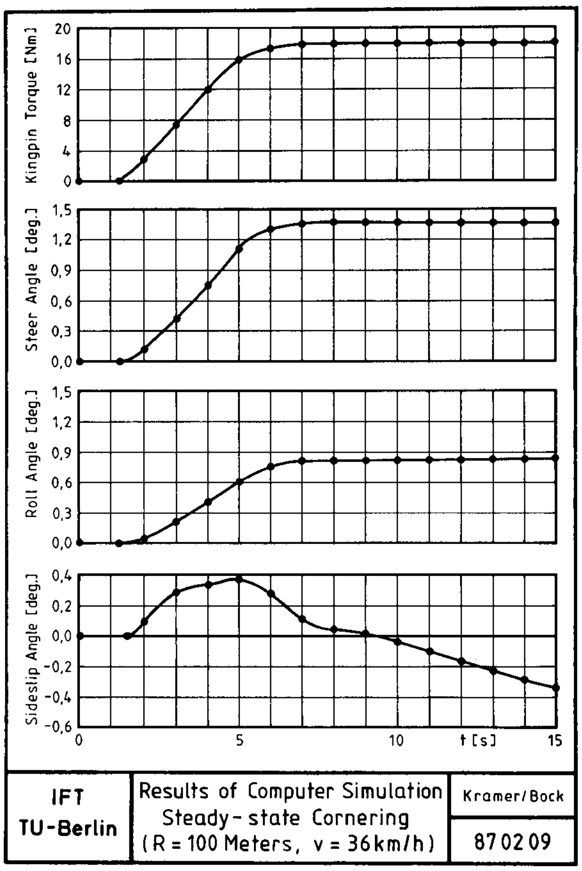

Figure 7. Results of computer simulation, steady-state cornering (R = 100 metres, v = km/h)

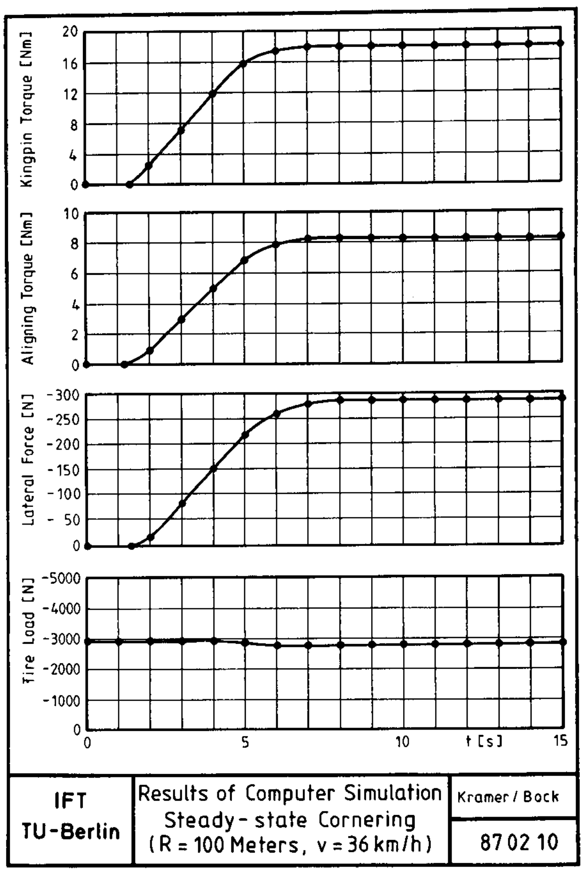

Figure 8. Results of computer simulation, steady-state cornering (R = 100 metres, v = 36 km/h)

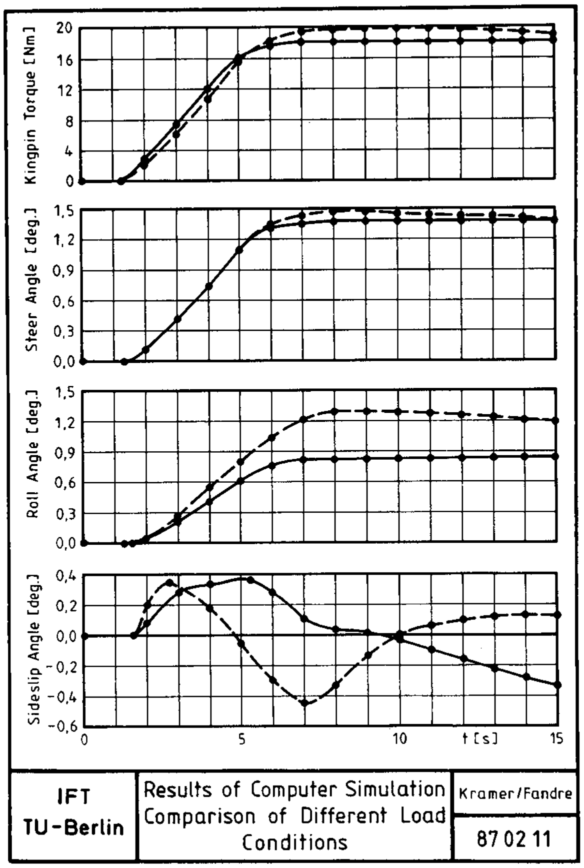

Figure 9. Results of computer simulation, comparison on different vehicle load conditions (R = 100 metres, v = 36 km/h)

on the curves of the two sideslip angles. At this point it must be stated emphatically that the slip qualities of the tires have a distinct influence on the course of simulation.

At the moment, along with further adaptation work on the vehicle model, simulation procedures are being carried out with all four modelled vehicles also taking into consideration the further characteristic driving manoeuvres, namely the input of various steering angle functions. The results will be reported in due time.

Conclusion

The application of the multi-body program system MEDYNA for calculating the vehicle dynamics allows the simulation of characteristic driving manoeuvres in accident situations after extensive model generating work. The mathematical simulation thus opens up the possibility of providing criteria for an avoidability study of road traffic accidents and to the evaluation of the primary vehicle safety by directed variation of the large number of automotive parameters. For reliability of the statements which can be obtained, a comparison of the calculated result with measurement data obtained by experiment and a subsequent adaptation process is required. In order to make this possible, it is necessary to carry out tests on vehicles and vehicle components.

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Operational and Design Features of the Steer Angle Dependent Four Wheel Steering System

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Abstract

Research on a directional control technique which steers the rear wheels along with the front ones has been reported in a series of papers since the 7th International Technical Conference on Experimental Safety Vehicles.

These fundamental studies have led to the development of a new four wheel steering system that varies the steer direction and angle of the rear wheels, depending on the steering wheel input. Among its features are decreased delay of steering response at high speed and decreased minimum turning radius at low speed.

This paper describes the basic principles of this new four wheel steering system, as well as its design and operation. It also discusses the results of tests of system function conducted under various operating conditions over a range of speeds, to demonstrate the benefits of installing this system in a vehicle.

Introduction

Since the 7th International Technical Conference on Experimental Safety Vehicles, we have reported on a series of studies on a control technique which steers the rear wheels in the same direction as the front ones[1]-[7]. These research efforts found that steering the rear wheels in the same direction as the front ones could reduce the delay in lateral acceleration response of the vehicle to steering input, resulting in more responsive steering characteristics under some conditions.

To achieve a shorter turning radius during very low speed maneuvers, specifically for parking the vehicle, it is desirable that the rear wheels should be steered in the opposite direction to the front ones. In an effort to successfully combine these two requirements for controlling the rear wheels, we proposed a steer angle dependent four wheel steering system, called the "Honda 4WS," that could change the direction and angle in which the rear wheels were steered, depending on how much the steering wheel was turned. Through continued research and development activities, the Honda 4WS has recently been completed technically and introduced into the market. The present report discusses the basic operational mechanism and design features of this steering system.

Operational Principle

It is known that steering the rear wheels in the same direction as the front ones results in a shorter delay in lateral acceleration response to steering input. This can offer more responsive steering characteristics when the drive is given such tasks as a lane change during highway cruising[1]-[5].

However, always steering the rear wheels this way increases the minimum turning radius of the vehicle. Especially during a sharp turn at low speed, it is preferable to steer the rear wheels in the opposite direction to the front ones. To meet these two conflicting requirements—one at high speed and the other at low speed, we have developed a variable control system that can steer the rear wheels in either the same direction or the opposite direction to the front ones, depending on the operating conditions of the vehicle.

The steer angle dependent Honda 4WS here is a new steering system which can meet the high and low speed control requirements by mechanical devices alone. As is apparent from the typical example given in Fig. 1, this system steers the rear wheels in the same direction as the front ones when the driver turns the steering wheel in a small angle, but if he turns the wheel in a large angle, it steers the rear wheels in the opposite direction.

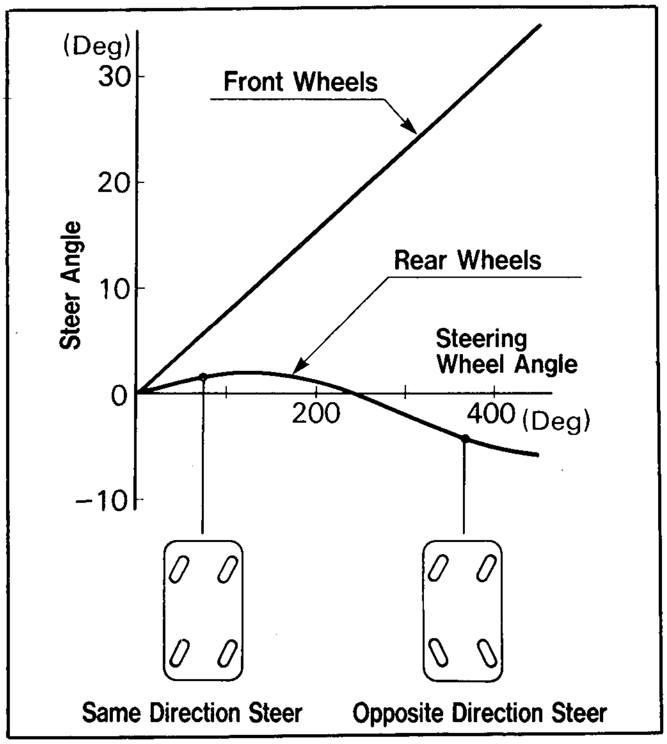

Figure 1. Steer characteristics of Honda 4WS

Major details of how this steering system works in high and low speed ranges are discussed below:

High-Speed Cruise

During a high-speed cruise, the driver usually turns the steering wheel in a relatively small angle. Fig. 2 shows the results of theoretically calculating the relationship between the steering wheel angle and vehicle speed of an automobile equipped with the Honda 4WS when making a steady-state circular turn. The diagram uses lateral acceleration as a parameter, which was selected at 0.2 g, 0.4 g and 0.6 g—typical values which could be used in normal highway cruises.

When the vehicle is going at 50km/h or faster, as indicated in the diagram, the steering wheel angle for making a steady-state circular turn with a lateral acceleration of 0.6 g or less comes within a 100-degree range. In the typical example of front and rear wheel steer angle characteristics given in Fig. 1, the rear wheels are steered in the same direction as the front ones in a speed range above a certain level. This results in a smaller delay in lateral acceleration response of the vehicle to steering input, offering quick, smooth steering response characteristics.

Low-Speed Maneuver

The driver turns the steering wheel in a large angle during short, sharp turns at very low speeds particularly for parking the vehicle, making a U-turn, or turning to the right or left at an intersection of narrow back streets. For such low-speed maneuvers, the Honda 4WS steers the rear wheels in the opposite direction to the front ones, reducing the minimum turning radius of the vehicle.

A steady-state turning radius of a vehicle equipped with the Honda 4WS is virtually equal to that of a conventional two-wheel steering car, the front wheels of which are steered by the difference between the front and rear steer angles of the Honda 4WS. As shown in Fig. 3, therefore, the equivalent steering gear ratio of the Honda 4WS depends on the relationship—the difference between the front and rear wheel steer angles in response to the steering wheel angle.

In a Honda 4WS vehicle, the direction and angle in which the rear wheels are steered depend on how much the steering wheel is turned. This means that its equivalent steering gear ratio varies with steering wheel angle. In the smaller range of steering wheel angle, the gear ratio becomes slow, while in the larger steering angle range, the ratio becomes more quick. During roughly straight-ahead driving, therefore, the system has a relatively small gain in yaw response to steering input, offering a moderate directional response. As the turning radius of the vehicle becomes shorter, the yaw response gain increases resulting in a larger directional response.

Construction and Operation

System Construction

The Honda 4WS can be constructed by simply adding a mechanical subsystem for steering the rear wheels to the conventional front-wheel steering system. As shown in Fig. 4, this four wheel steering system is essentially comprised of two subsystems, the front and rear steering gear boxes, which are mechanically linked with each other by center steering shaft.

Now let us briefly describe how the system works. When the driver turns the steering wheel, a rack and pinion mechanism in the front steering gear box moves the rack axially. This rack stroke steers the front wheels, and at the same time, turns the output pinion shaft by another rack and pinion mechanism in the front gear box to transmit the steering wheel angle to the rear steering gear box through the center

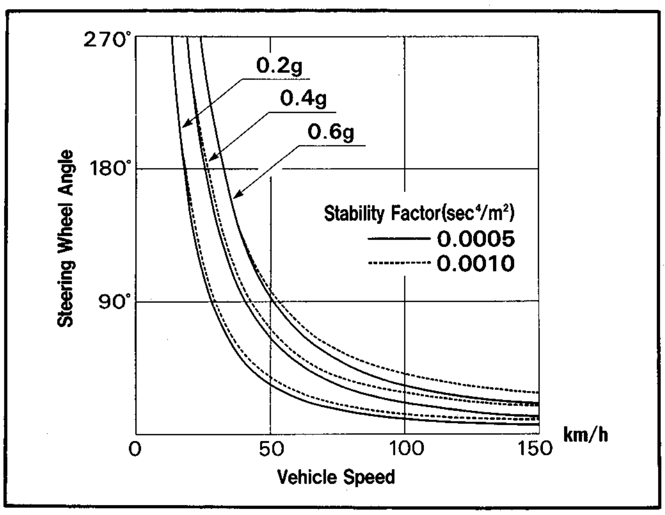

Figure 2. Relation between steering wheel angle and vehicle speed of Honda 4WS car at steady-state cornering

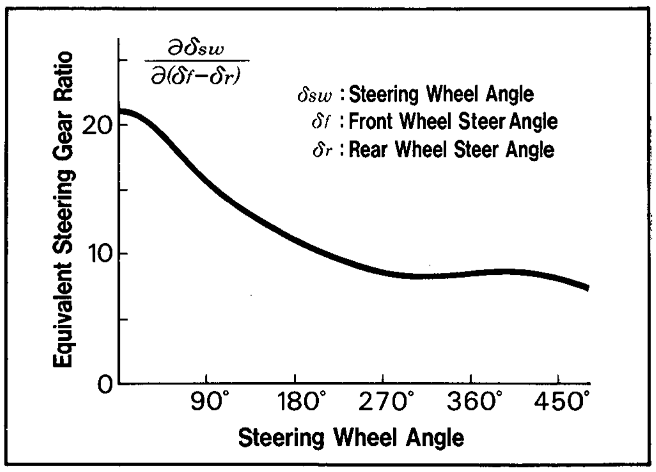

Figure 3. Equivalent steering gear ratio of Honda 4WS

steering shaft. In response to the steering angle thus transmitted, the stroke rod in the rear gear box moves axially to steer the rear wheels through the tie rod.

The rear gear box has a built-in variable gear ratio mechanism which changes the direction and ratio of the stroke rod's output stroke to the input, depending on the steering wheel angle. This gives the system steer angle dependent control characteristics. More specifically, when the steering wheel is turned from the straight-ahead position, the rear wheels are steered at first in the same direction as the front ones, but as the steering wheel angle becomes larger than a certain value, the rear wheels are steered in the opposite direction to the front ones.

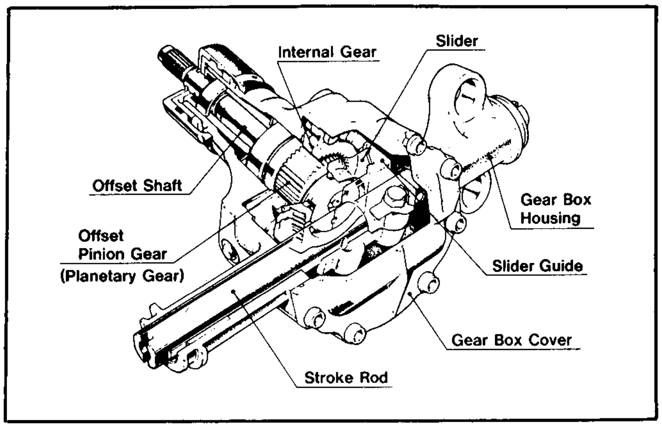

Figure 5. Perspective view of rear steering gearbox

Construction and Operation of the Rear Steering Gear Box

Fig. 5 shows the construction of the rear steering gear box. A schematic description of its basic mechanism is given in Fig. 6. As shown in these diagrams, the rear gear box uses a combination of two offset shafts, the revolution of which are synchronized by a planetary gear that meshes with a stationary internal gear. Through this mechanism, the orbital motion of axis PP′ around axis OO′ is joined with the QQ′′s movement by axis PP′′s rotation, so that changes in the lateral position of axis OO′ are transmitted to the stroke rod as its stroke output. Changes in the vertical position of axis OO′ are absorbed by the slider/guide mechanism.

Fig. 7 shows the operation of the planetary gear when the turning angle of the shaft, input into the rear steering gear box, is varied as 90°, 180° and 270°.

Fig. 8 describes the input/output characteristics given to the rear steering gear box by its working mechanism discussed above. This is how the steer angle characteristics of the front and rear wheels in Fig. 1 are obtained.

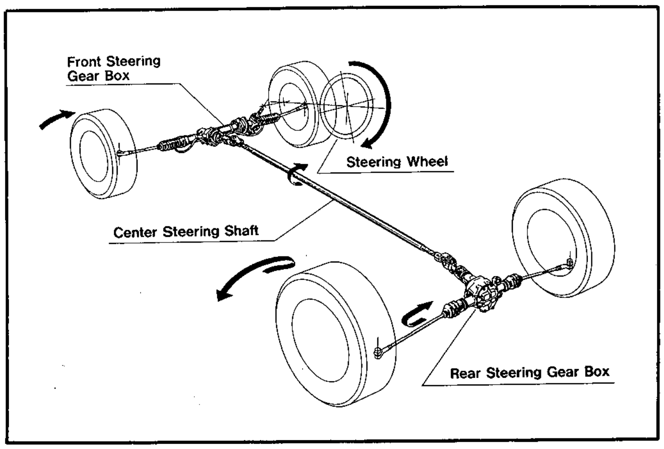

Figure 4. System construction of Honda 4WS

Test Results of Honda 4WS System

A series of tests were conducted on the Honda 4WS installed in a compact car powered by a 1.8-liter engine to compare its performance with that of a two-wheel steering car of the same basic specifications. Some of the tests findings are discussed below:

Frequency Response Characteristics at High Speed

Fig. 9 shows the lateral acceleration and yaw velocity response of the vehicle to steering input for a steering wheel angle of around ±45°, which is within the smaller steering angle range where the rear wheels are steered in the same direction as the front ones.

The Honda 4WS and the comparable two wheel steering car differed most in phase delay in lateral acceleration: the Honda 4WS vehicle had a shorter delay at high steering frequency. Another notable finding is that the change in yaw rate gain with Honda 4WS is less than with two wheel steering up to a high steering frequency

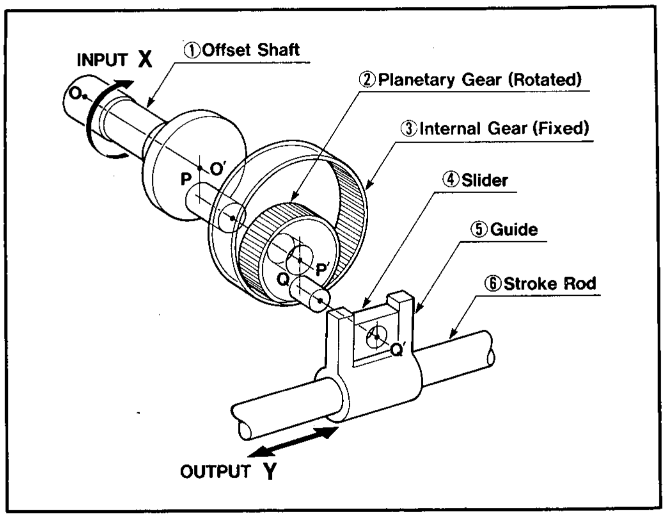

Figure 6. Basic mechanism of rear steering gearbox

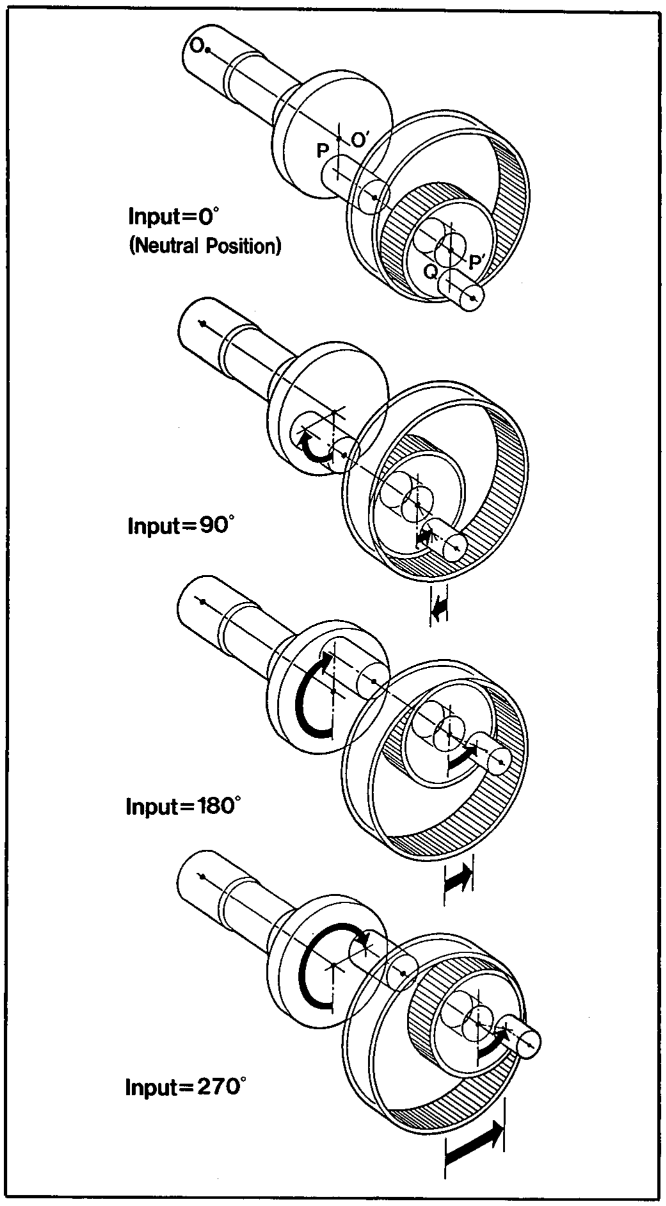

Figure 7. Operation of rear steering gearbox

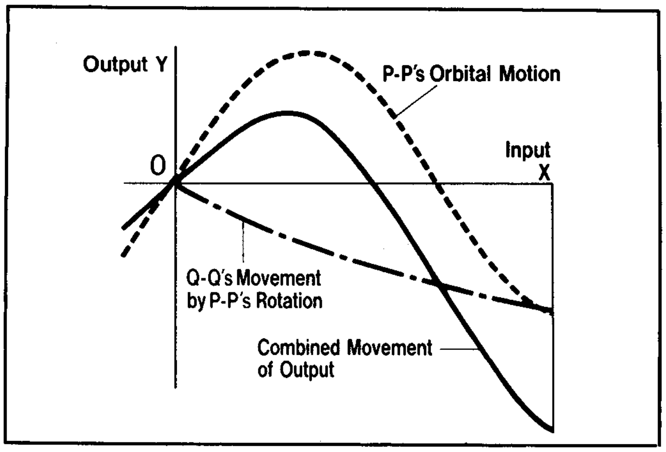

Figure 8. Input/output characteristics of rear steering gearbox

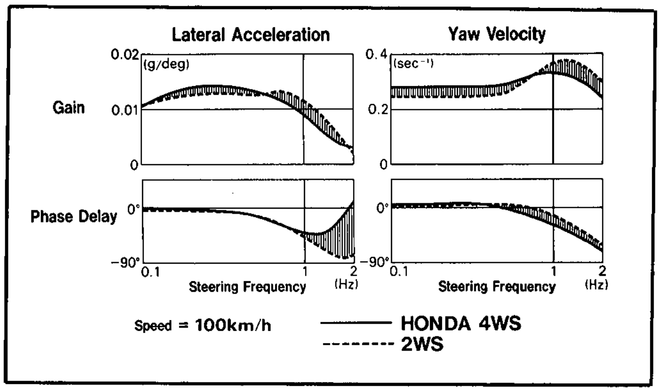

Figure 9. Vehicle steering response characteristics

Lane-Change Test at High-Speed

In this test, the driver was requested to steer the vehicle in such a manner that the position of its center of gravity would follow as closely as possible the desire path marked with an objective line on the test course surface as described in Fig. 10.

The difference between the two test vehicles in steering response during an example lane change is shown in Fig. 11. In this particular example, the Honda 4WS vehicle had smaller amplitudes of lateral acceleration, yaw velocity and roll angle.

Turns at Very Low-Speed

A U-turn test was conducted at very low speed with the steering wheel set at the maximum steering wheel angle. Because of its minimum turning radius, some 0.5 meters shorter than the two-wheel steering car, as shown in Fig. 12, the Honda 4WS vehicle could make U-Turn in a test course 1 meter narrower than the minimum course width required by the other test car. Of course, these results will vary slightly with the wheelbase, track and other specifications of the vehicle in which the steering system is installed.

Conclusion

In the foregoing sections, we have described the operational and design features of the steer angle dependent Honda 4WS.

The Honda 4WS offers both steering ease at high speed and good maneuverability at low speed by

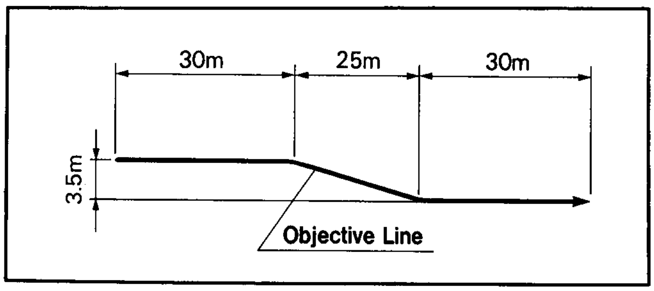

Figure 10. Course layout of lane-change test

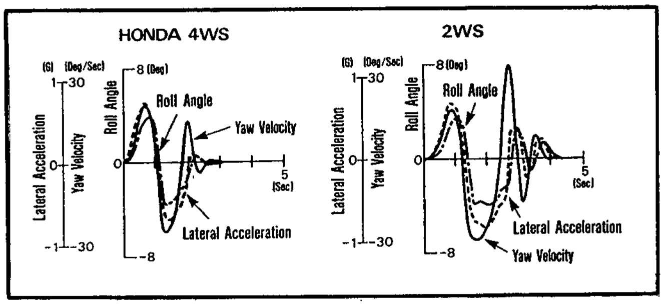

Figure 11. Results of lane change test (vehicle movement)

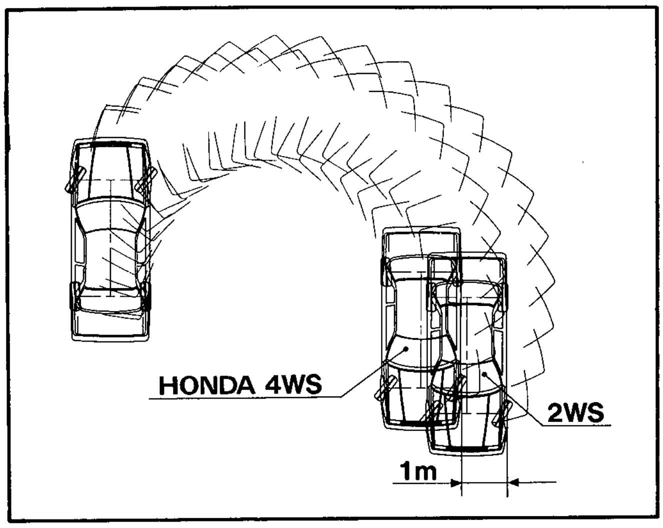

Figure 12. Difference of vehicle path during U-turn

mechanical devices alone. We hope the present report will be helpful for further research and development in this field.

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4WD Vehicle Behavior During Braking in a Turn

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Abstract

Vehicle behavior during braking in a turn is important for vehicle stability and controllability. Behavior of a four wheel drive (4WD) vehicle is different from that of a two wheel drive (2WD) vehicle due to larger moment of inertia of driveline and restraint condition between front and rear anxle.

Theoretical analysis and experimental study were conducted on the behavior of three typical 4WD systems; (1) 4WD with center differential; (2) 4WD with directly connected front and rear axles; (3) 4WD with viscous coupling (viscous transfer type).

In order to confirm the validity of the simulation, vertical, longitudinal and lateral forces working on four wheels during braking in a turn were measured by a six-component wheel load cell installed on each wheel.

This study clarifies that the influence of suspension characteristics and brake force distribution is important in 4WD vehicles with center differential and the influence of suspension characteristics is important in directly connected 4WD vehicles and 4WD vehicles with viscous coupling.

Introduction

The superiority of 4WD vehicles in terms of their performance under various road conditions has recently come to attract attention(1)(2), and several papers on the stability and controllability of 4WD vehicles have been published(3)(4)(5)(6). However, few papers have dealt with their behavior during braking in a turn.

A wheel rotational moment of inertia in 4WD vehicles is different from the one of 2WD and other 4WD systems due to the various relationships between front and rear axles. Thus, 4WD vehicles behave differently from 2WD vehicles during braking in a turn and one of 4WD systems behaves differently from others.

This paper discusses the differences in the behavior of various 4WD systems during braking in a turn, based mainly on a theoretical analysis, and presents the results of a study made on the influence of vehicle configuration and driving conditions on vehicle behavior.

Experiment and Simulation Conditions

Conditions

The conditions are basically, as prescribed in ISO 7975 (Road Vehicles—Braking in a turn—Open loop test procedures).

The conditions are as follows:

1. Road surface: Coefficient of friction $\mu = 0.8$ (except for 4.2.1. in which the influence of the coeffient of friction of road surface was analyzed)
2. Initial path radius: 50 m
3. Initial lateral acceleration: 6.6 m/s^2 (higher value than the ISO standard of 5 m/s^2 $\pm$ 10% was adopted for clearer differentiation of behavior)
4. Longitudinal deceleration: 2 m/s^2 to 7 m/s^2 in about 1 m/s^2
5. Evaluation items: Ratio of yaw velocity at 1s after brake application and initial yaw velocity $\left(\frac{\dot{\psi}1s}{\dot{\psi}0}\right)$ (hereafter referred to as the normalized yaw velocity)

In the experiments, a brake fluid pressure limiting controller was used to ensure more rapid attainment of a target braking deceleration and repeatability of deceleration. The temperatures of the front wheel brake pads were maintained below 90°C and the difference between the right and left front wheel pads was kept within a 30°C range, in order to minimize brake force variation due to temperature.

Outline of a Test Vehicle

1. Vehicle specifications
   - Vehicle mass 1398 kg
   - Wheelbase 2.465 m
   - Track (Front) 1.420 m
   - Track (Rear) 1.425 m
   - Axle load distribution 56/44
   - Final drive gear ratio 3.7
2. 4WD system
   (a) 4WD with central differential (hereafter abbreviated as CD 4WD)
   (b) 4WD with directly connected front and rear axles (hereafter abbreviated as Rigid 4WD)
   (c) 4WD with viscous coupling (hereafter abbreviated as VC 4WD)

A viscous coupling with the characteristics as shown in Fig. 1 was installed on the propeller shaft.

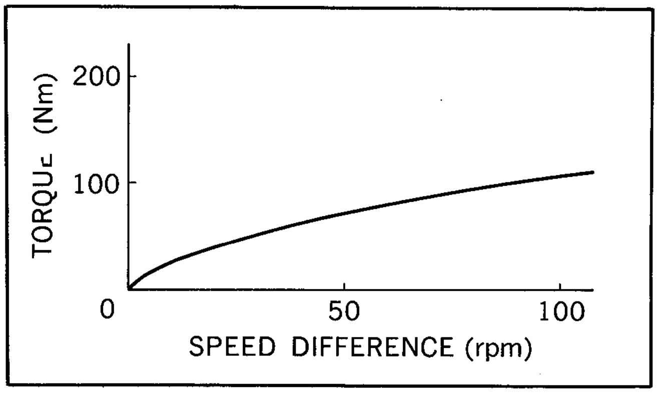

Figure 1. Performance characteristics of viscous coupling

3. Brake force distribution

Fig. 2 shows the angled line brake force distribution and ideal brake force distribution during braking.

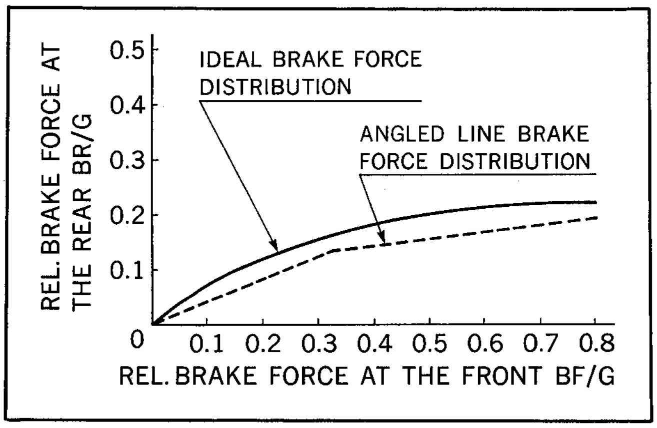

Figure 2. Comparison of brake force distribution

Description of a Computer Simulation Model

The simulation model consists of a body and driveline including tires. Degrees of freedom are six for the body and four for the driveline, respectively. The body model (Fig. 3) also takes into account the roll steer and compliance steer. The driveline model (Fig. 4) includes the inertia of the transmission and four wheels. The equations of motion of the body and of the wheel rotation axis are shown below.

(Equation of motion of vehicle body)

$$m \frac{dV}{dt} \cos\beta - m V \frac{d\nu}{dt} \sin\beta = Fx$$

$$m \frac{dV}{dt} \sin\beta + m V \frac{d\nu}{dt} \cos\beta = Fy$$

$$m \frac{d^2z}{dt^2} = Fz$$

$$Ix \frac{d^2\phi}{dt^2} = Mx$$

$$Iy \frac{d^2\theta}{dt^2} = My$$

$$Iz \frac{d^2\psi}{dt^2} = Mz$$

where m: Vehicle mass

- V: Velocity of the center of gravity of the body
- β: Body sideslip angle
- ν: Heading angle
- ϕ: Roll angle
- θ: Pitch angle
- ψ: Yaw angle
- Fx, Fy, Fz: Force acting on the body
- Mx, My, Mz: Moment acting on the body
- Ix, Iy, Iz: Moment of inertia of the body about each axis

(Equation of motion of driveline model)

$$Ii \frac{d\omega i}{dt} = -Bi + Ti + FiRsi - Wi\epsilon$$

where subscript i denotes wheel location

- Ii: Rotary moment of inertia of wheel
- Rsi: Axle height
- Bi: Brake torque received from caliper
- Fi: Brake force at tire ground contact surface
- ωi: Wheel rotational speed
- Wi: Vertical load
- ϵ: Distance from contact point of vertical load to wheel axle
- Ti: Torque received from driveline

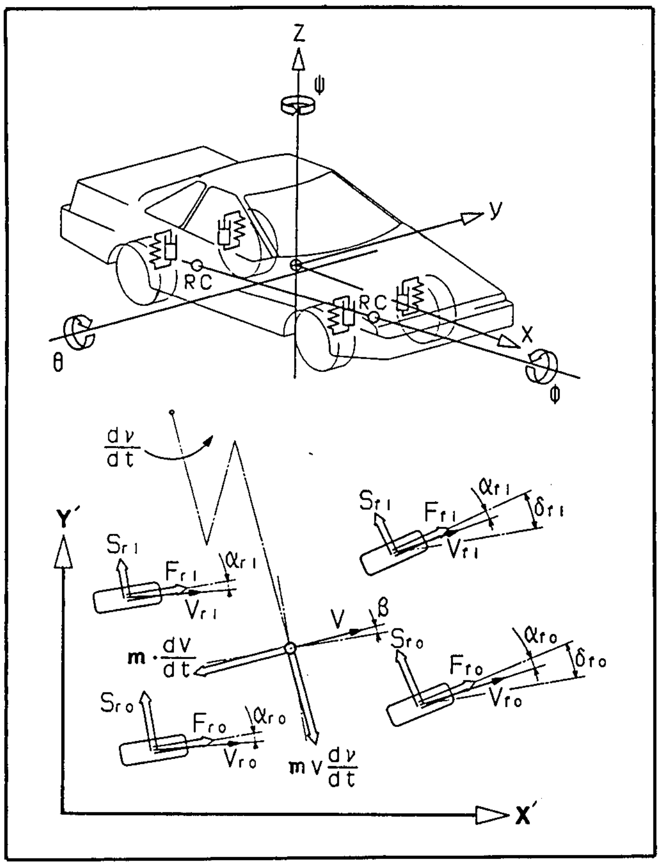

Figure 3. Vehicle model

In this equation

$$Ti = \frac{1}{4} \times 3.7 \times T_M \text{ for CD 4WD}$$

(where T_M = Torque due to inertia of transmission).

$$\omega 1 + \omega 2 = \omega 3 + \omega 4 \text{ for Rigid 4WD}$$

$$Ti = \frac{1}{2} \times 3.7 \times Tv \text{ for rear wheels of VC 4WD}$$

(where Tv = torque generated by viscous coupling

$$Ti = \frac{1}{2} \times 3.7 \times (T_M - T_V) \text{ for front wheels of VC 4WD}$$

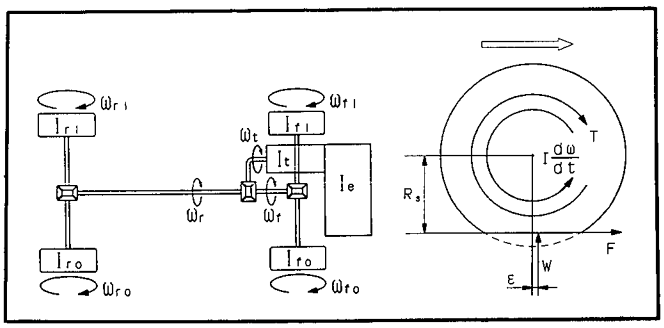

Figure 4. Drive model

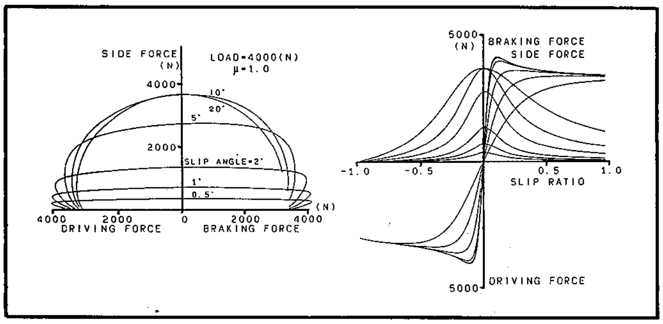

Figure 5. Tire properties

Assuming the tire model with the characteristics shown in Fig. 5, the side force is expressed as a function of tire sideslip angle α, slip ratio SR, tire traveling speed Vi, coefficient of friction between tire and road surface μ and vertical load W. Here, the slip ratio SR is defined as follows.

$$SR = 1 - Re\omega i / Vi \cos \alpha$$

Re: Tire rolling radius = $Vi/\omega i$ (when SR = 0)

Fig. 6 shows that the side slip angles for the Rigid 4WD and VC 4WD vehicle are larger than the CD 4WD's slip angle in higher deceleration region, and that they tend to verge to the inside. This is also reflected in the normalized yaw velocity.

Test Results

Test results shown in Fig. 7 show similar trends to the simulation results in Fig. 6.

The above-mentioned model will be used in the following discussion.

Forces Acting on Each Wheel During Braking in a Turn and Vehicle Behavior

Fig. 8 shows the simulated time histories of the loads, forces and rotational speed acting on each wheel in the three 4WD systems.

Figure 10 and 11 illustrate the wheel load cell and an amplifier.

The studies mentioned above clarify the characteristics of each 4WD system in the following manner:

1. Front brake force of the CD 4WD vehicle is larger than the one of other 4WD systems.

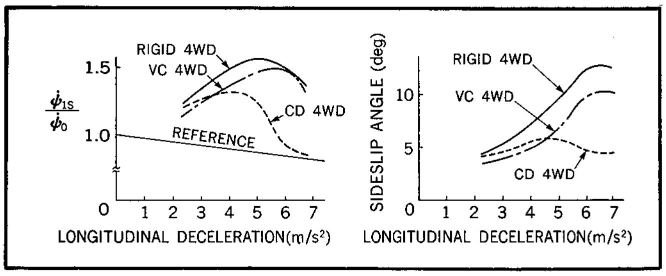

Figure 6. Normalized yaw velocity vs. sideslip angle of three types of 4WD systems

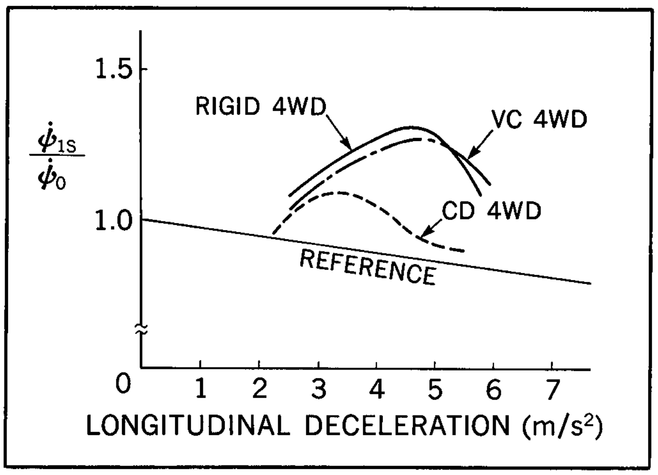

Figure 7. Normalized yaw velocity of three types of 4WD systems (measurement)

The earliest decrease of wheel rotational speed occurs at the front inner wheel due to absence of a constraining force between the front and rear axles.

2. The Rigid 4WD has lower front brake force and higher rear brake force than the CD

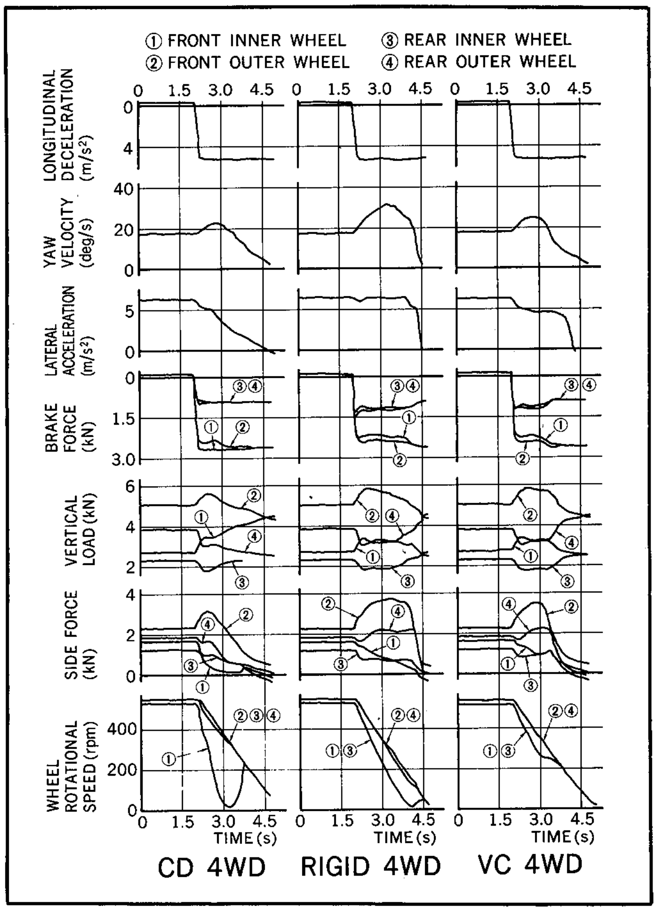

Figure 8. Simulated time histories of the loads, forces and rotational speed acting on each wheel in the three 4WD systems

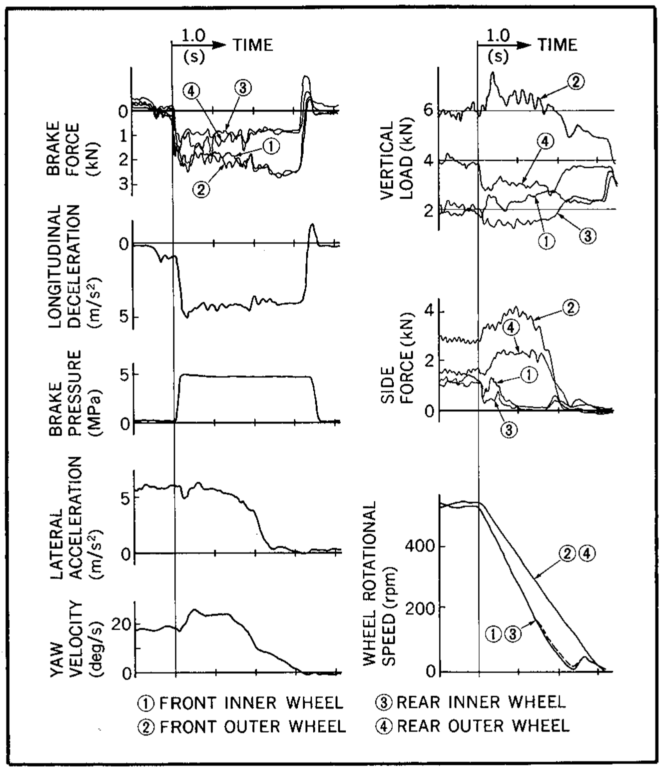

Figure 9. Results of measurement by six-component wheel load cells, etc. (rigid 4WD).

4WD, and the front and rear inner wheels rotational speed shows a drop simultaneously. This is due to the direct coupling of the front and rear axles, which causes identical rotation of the two axles. Therefore, the brake force distibution traces an ideal curve in Fig. 2, in which the front brake force is reduced by the rear brake force, and the rear brake force is increased by the front brake force vice versa.

3. The VC 4WD system shows a similar brake force distribution to that of the Rigid 4WD

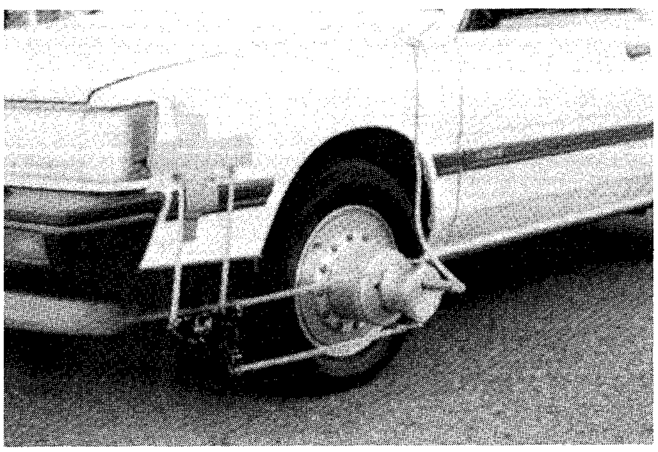

Figure 10. A six-component wheel load cell

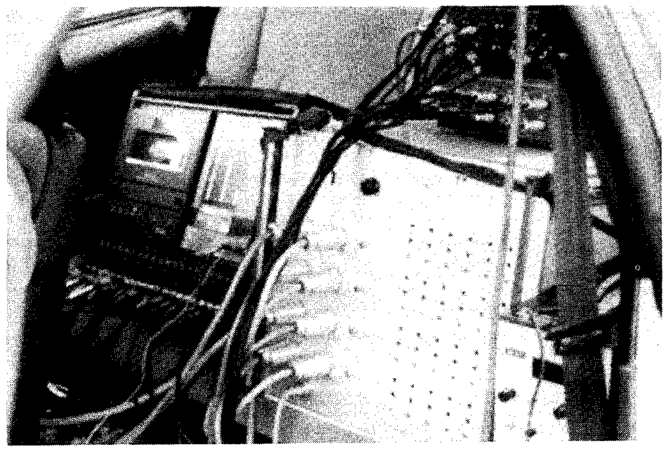

Figure 11. Amplifier, etc.

system in the initial stage of braking, and in the later stage shows a similar brake force distribution to that of the CD 4WD. As for the wheel rotational speed, the front inner wheel decreases first. This is due to the fact that the front and rear axles are coupled by a viscous coupling and the constraining force changes with the difference of speed of two axles as shown in Fig. 1. Specifically, immediately after starting to brake, the decrease of the front inner wheel rotational speed causes a difference of speed between the front and rear axles, and an increase of the constraining force. This leads to a brake force distribution similar to that of the Rigid 4WD. In the later stage of braking, the rotational speed of the rear inner wheel drops to the same level as the front inner wheel, so that the difference in speed of revolution of the two axles becomes smaller and hence the brake force distribution comes close to the CD 4WD.

Thus, the three 4WD systems have different brake force distributions and different decreasing way of wheel rotational speed, depending on the difference in the constraining force between the front and rear axles. As a

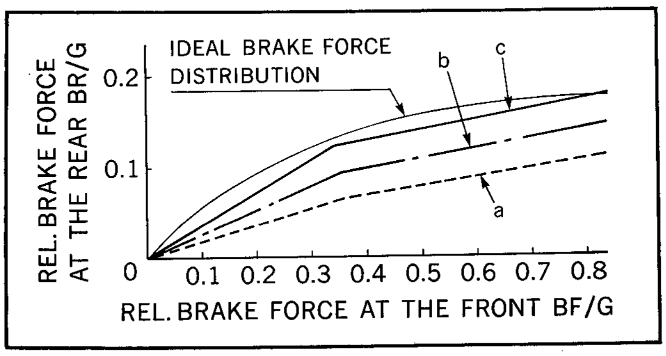

Figure 12. Changing of brake force distribution

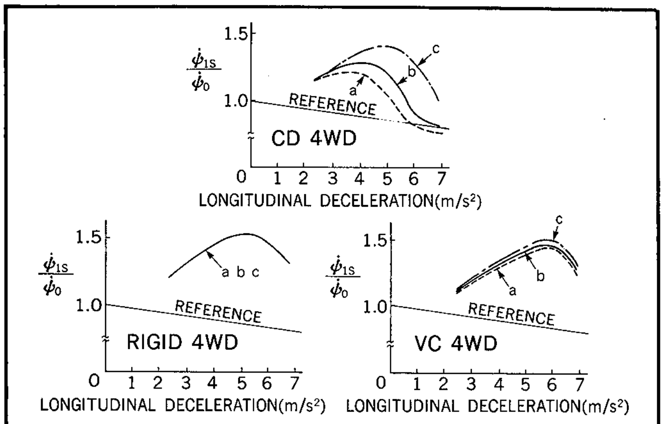

Figure 13. Influence of brake force distribution

result, the lateral force and yaw velocity also differ from one system to another, resulting in different behavior.

Simulation Study

Performance as a Function of Vehicle Factors (Specifications)

Influence of the brake force distribution

Fig. 13 shows the simulation results obtained by changing the brake force distribution of the three types of 4WD systems as shown in Fig. 12 (a, b, c).

From Fig. 13, the following observations can be made.

1. The behavior of the CD 4WD changes remarkably with the brake force distribution.
2. The Rigid 4WD is not influenced by the angled line brake force distribution; its behavior remains unchanged.

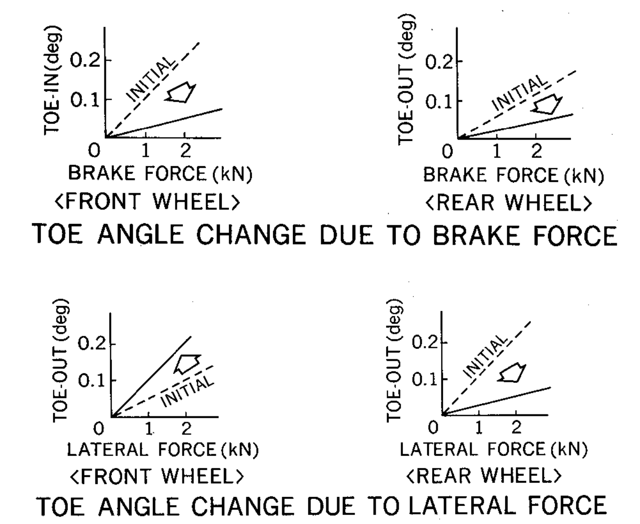

Figure 14. Compliance steering characteristics

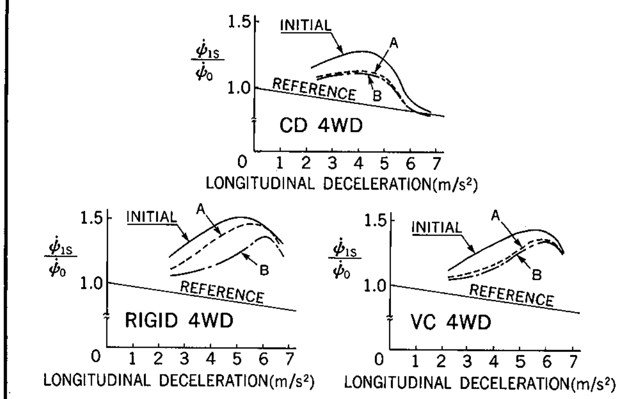

Figure 15. Influence of suspension characteristics (compliance steer)

3. The VC 4WD is less influenced by the angled line brake force distribution as its brake force distribution is more similar to that of the Rigid 4WD (ideal distribution) than that of the CD 4WD.

Influence of the suspension characteristics

Fig. 15 shows the simulated results obtained by changing the compliance due to longitudinal and side forces as shown in Fig. 14.

The normalized yaw velocity for all three systems decreases due to the change of the compliance steer caused by longitudinal force and side force shown above. Furthermore, when the compliance steer due to side force was also changed as shown above, the normalized yaw velocity can be further reduced. This is because cornering force increase at the front wheel and decrease at the rear wheel can be controlled by the compliance steer.

Influence of height of the center of gravity

Fig. 16 shows the simulation results obtained by changing the height of the center of gravity of the simulation model from the reference level (0.43 m).

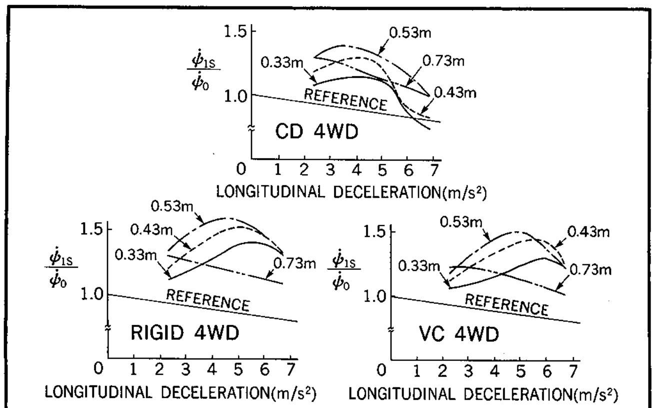

Figure 16. Influence of height of the center of gravity(1)

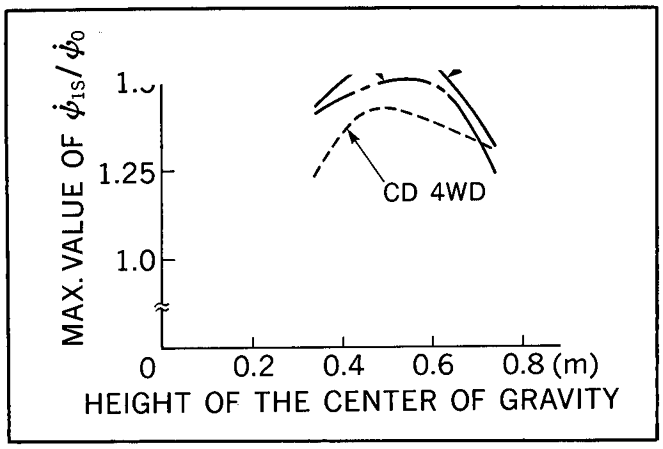

Figure 17. Influence of height of the center of gravity(2)

1. All three systems have the peak of normalized yaw velocity at each height of the center of gravity shown in Fig. 17.
2. A lower height of the center of gravity reduces load shift of the inner and outer wheels during a turn and of the front and rear wheels during braking, reducing change in the vertical load on the inner and rear wheels, making wheel lock less likely to occur.
3. A higher height of the center of gravity, on the other hand, caused very large load shift both laterally and longitudinally. This causes the inner wheels to lock even at low deceleration and hence the move is moderated.

Performance as a Function of Driving Conditions

Influence of the coefficient of friction of the road surface

Fig. 18 shows the simulation results for braking in a turn on three types of road surface conditions: dry pavement (μ = 0.8), wet pavement (μ = 0.5) and snow covered surface (μ = 0.3).

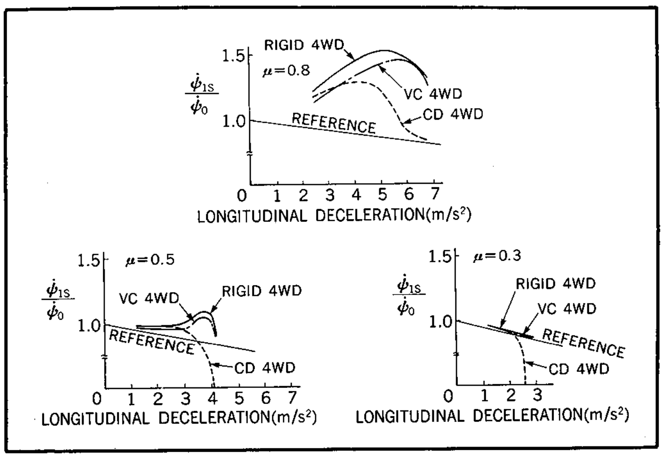

Figure 18. Influence of the coefficient of friction of the road surface

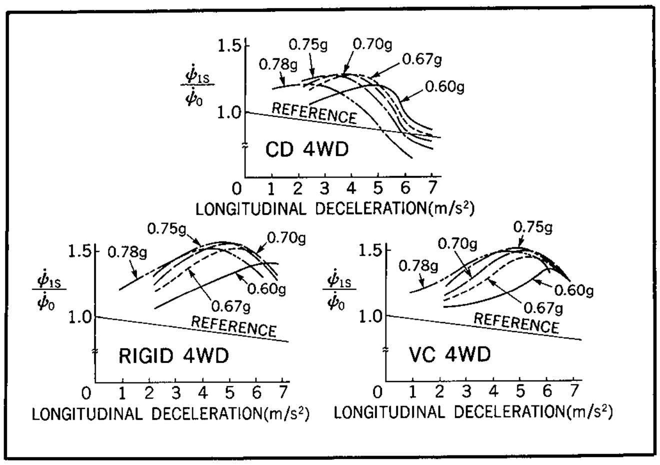

Figure 19. Influence of lateral acceleration(1)

The CD 4WD behaves more stably with a smaller normalized yaw velocity than other systems on road surfaces with higher coefficients of friction but shows a drift-out tendency as the coefficient becomes smaller. This is due to the fact that front inner wheel locks first, regardless of the coefficient of friction.

The Rigid 4WD retains turning ability even in the case of a small coefficient of friction. This is due to the fact that its brake force is distributed toward the rear axle and consequently alleviates a drift-out tendency.

The VC 4WD exhibits behavior equivalent to the Rigid 4WD as the coefficient of friction decreases. This is due to the fact that the transferring force (rotational speed difference) becomes smaller as the coefficient of friction is decreased and hence the system resembles the Rigid 4WD, with less tendency to drift-out.

Fig. 19 shows the simulation results obtained by changing the lateral acceleration with the coefficient of friction of the road surface and the turning radius kept constant. The brake deceleration at the peak of $\psi 1s/\psi 0$ reduces according to the lateral acceleration as shown in Fig. 20.

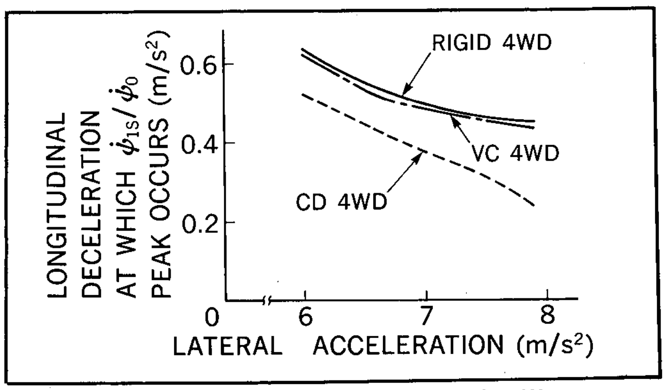

Figure 20. Influence of lateral acceleration(2)

From the tire characteristics shown in Fig. 5, it is apparent that the change in the cornering force due to brake force is small when the slip angle is small. Namely, the tire characteristics are less likely to change up to a high deceleration point. The CD 4WD had a smaller deceleration of a $\frac{\psi_{ls}}{\psi_0}$ peak than other systems because of earlier locking of the front inner wheel.

Conclusion

The braking performance during a turn of three types of 4WD system with different front and rear wheel constraining condition was examined. It was found that the differences of behavior of each system depends on the vehicle configuration and driving condition due to the differences in front and rear brake force and wheel speed.

These results indicate the importance of characteristics of brake force distribution and suspension compliance steer for the CD 4WD system, and indicate the importance of characteristics of suspension compliance for Rigid 4WD and VC 4WD systems.

In this study, vehicle behavior was examined with the steering angle is fixed during braking in a turn and does not deal with steerability. Further studies are expected on this subject and also on the 4WD system with ABS, in order to accomplish higher steerability and braking performance under wide variety of driving conditions.

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Improved Handling and Stability Using Four-Wheel Steering

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Abstract

The effect of four-wheel steering on vehicle handling and stability was investigated. It was found that vehicle stability is improved by steering the rear wheels in the same direction as the front wheels. The steer angle of the rear wheels should increase with increasing vehicle velocity. Transient characteristics of the rear wheel steer angle are important in achieving both good stability and steering response. One way of achieving this is to introduce a suitable time delay in the rear wheel steer angle. These results have been incorporated into the High Capacity Actively Controlled Suspension (HICAS) system. It was also confirmed by analysis and experiments that handling and stability are significantly improved by actively controlling the front wheel steer angle to achieve an optimum match with the rear wheels.

Introduction

There have been various reports on the use of active steer control for the rear wheels to improve vehicle stability and handling properties. (1)-(5) A variety of evaluation standards have been discussed recently for optimizing the control functions.

From these different research results the authors have gained the following insights. The sideslip angle of the vehicle can be reduced to zero by simultaneously steering the rear wheels in the same direction as the front wheels and in proportion to their steer angle. The result will be improved vehicle stability and better lateral acceleration response. On the other hand, this will also cause the yaw rate response to deteriorate. Transient sideslip will occur in the opposite direction from that of a two-wheel-steering system, which will be disconcerting to the driver.

The authors have attempted to overcome these problems by focusing on the transient characteristics of steer angle control. Several different types of typical steer angle control functions have been introduced to resolve the problems noted above. Their effectiveness in improving vehicle stability and handling properties has been confirmed through simulation analysis and experimentation.

Definition of Four-Wheel Steering

The steady-state sideslip angle of a vehicle, β, can be reduced to zero by controlling the rear wheel steer angle according to the following procedure.

$$Kr = \frac{\delta r}{\delta f} = - \frac{b\ell CfCr - aMCfV^2}{a\ell CfCr + bMCrV^2} \qquad (1)$$

In this equation, Kr is the ratio of the rear wheel steer angle relative to that of the front wheels and it varies according to the vehicle velocity, V. This idea is illustrated in Fig. 1.

Reducing the sideslip angle to zero has both ergonomic and dynamic aspects. In terms of ergonomics, the vehicle will be easier to steer because it will constantly proceed in the forward direction without any sideslip. As for the motion dynamics involved, less energy will be required to turn the vehicle when it enters a turn from a straight-line course. This will mean improved stability along with better yaw and lateral motion convergence.

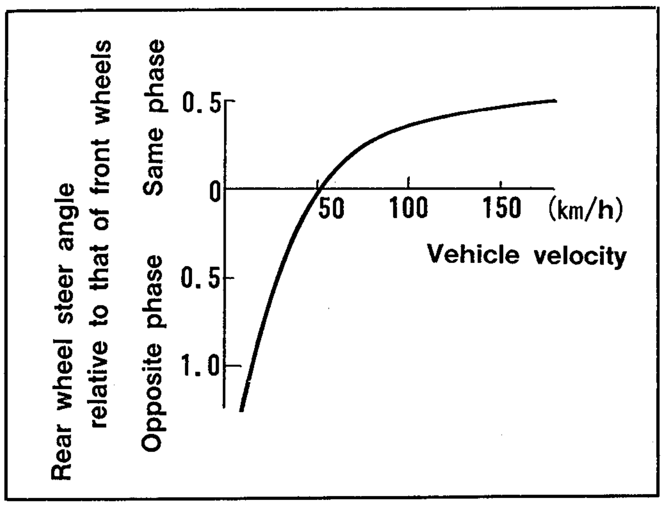

Figure 1. Rear wheel steer angle relative to that of front wheels (sideslip angle = 0)

The second aspect is explained in more detail in reference to Fig. 2. When a front-wheel-steering (2WS) vehicle turns, its nose generally points increasingly toward the center of the turn as the vehicle velocity increases. This is illustrated in (i) and (ii) in the figure. The large centripetal force needed for turning at high speed is obtained by means of the large sideslip angle and the increased side force of the tires. In other words, when a vehicle enters a turn from a straight-line course through a series of step steering inputs, it is necessary to turn the vehicle using β until the vehicle assumes a stable condition.

In contrast to this, the rear wheels of a four-wheel-steering (4WS) vehicle are steered in the same direction (phase) as the front wheels, which increases the slip angle of the rear tires. This reduces the sideslip angle of the vehicle and provides large centripetal force. To turn a vehicle with 4WS, it is only necessary to turn the tires, which have a small moment of inertia, instead of turning the vehicle body with its large moment of inertia. Since this requires much less energy, it allows easier convergence of yaw motion.

Steering the rear wheels according to Eq. (1) achieves good stability in intermediate and high speed ranges, however, it does not always provide a favorable steering sensation. This is because it tends to produce a strong understeer characteristic and it causes yaw response to deteriorate.

Steering the rear wheels in the opposite direction (phase) of the front wheels at low speed improves maneuverability. However, it is better to treat maneuverability separate from handling and stability since their respective aims are different.

Analysis Using a Mathematical Model

Analytical Model

A two-wheel model like that illustrated in Fig. 3 was employed in the analyses. The front and rear wheel control functions, Af and Ar, are given in the

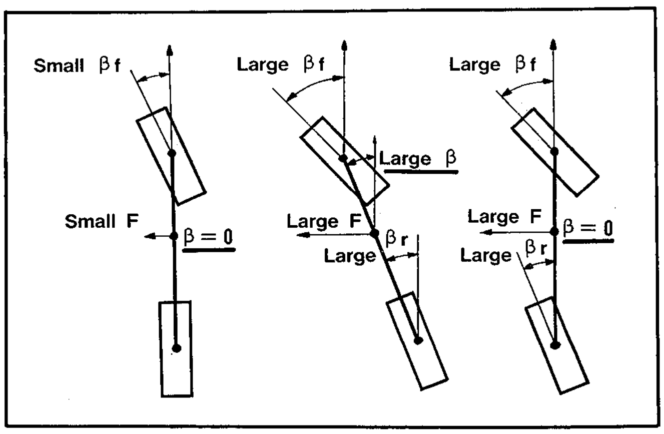

Figure 2. Relationship between centripetal force (F) and sideslip angle (β)

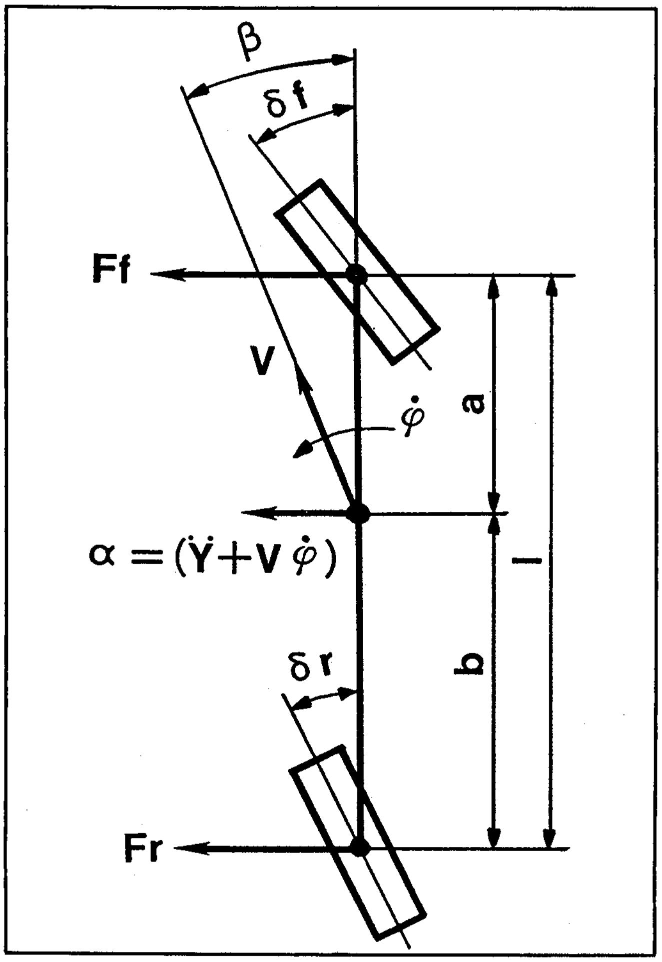

Figure 3. Two-wheel steering model

form of transfer functions relative to the steering wheel angle, as shown in Fig. 4.

The basic equations are given below.

$$M\alpha = Ff + Fr \tag{2}$$
$$I\ddot{\varphi} = aFf - bFr \tag{3}$$
$$Ff = Cf\{\delta f - (a\dot{\varphi} + \dot{y})/V\} \tag{4}$$
$$Fr = Cr\{\delta r - (-b\dot{\varphi} + \dot{y})/V\} \tag{5}$$
$$\delta f = (\theta/N)\cdot(1 + Af) \tag{6}$$
$$\delta r = (\theta/N)\cdot Ar \tag{7}$$

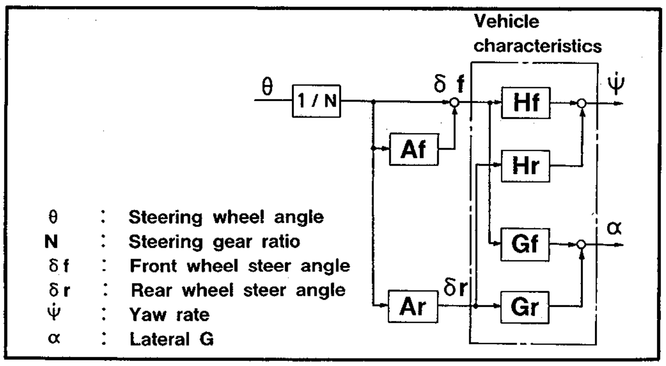

Figure 4. Block diagram of simulation model

From Eqs. (2) - (7), the vehicle's response relative to the steering wheel angle is given as follows:

$$\frac{\text{yaw rate}}{\text{steering wheel angle}} = B_1 \bullet \frac{\omega n^2 (1 + \tau_1 S)}{S^2 + 2\zeta n\omega n + \omega n} \tag{8}$$

Here,

$$\frac{\text{lateral acceleration}}{\text{steering wheel angle}} = B_2 \bullet \frac{(\omega n/\omega_2)^2 \ (S^2 + 2\zeta_2\omega_2 S + \omega_2{}^2)}{S^2 + 2\zeta n\omega n + \omega n} \tag{9}$$

$$B_1 = \frac{V\,[(1+\delta f) - \delta r]}{\ell(1 + KsV^2)} \tag{10}$$

$$B_2 = \frac{V^2\,[(1+\delta f) - \delta r]}{\ell(1 + KsV^2)} \tag{11}$$

$$\omega n = \frac{\ell}{V}\sqrt{\frac{CfCr(1 + KsV^2)}{IM}} \tag{12}$$

$$\omega_2 = \sqrt{\frac{\ell Cr}{I}}\sqrt{\frac{(1+\delta f) - \delta r}{(1+\delta f) + \frac{Cr}{Cf}\,\delta r}} \tag{13}$$

$$\zeta n = \frac{(Cf + Cr)I + (a^2Cf + b^2Cr)M}{2\ell\sqrt{IMCfCr(1 + KsV^2)}} \tag{14}$$

$$\zeta_2 = \frac{b}{2V}\sqrt{\frac{\ell Cr}{I}}\sqrt{\frac{[(1+\delta f) + \frac{a}{b}\,\delta r]^2}{[(1+\delta f) - \delta r]\ [(1+\delta f) + \frac{Cr}{Cf}\,\delta r]}} \tag{15}$$

$$Ks = \frac{M}{\ell^2}\left(\frac{b}{Cf} - \frac{a}{Cr}\right) \tag{16}$$

$$\frac{1}{\tau_1} = \frac{Cr\ell}{aMV} \bullet \frac{(1+\delta f) - \delta r}{(1+\delta f) - (\frac{bCr}{aCf})\delta r} \tag{17}$$

Using these equations, a comparison was made of the four typical control methods noted below in an effort to determine what phase characteristic should be given to the rear wheel steer angle control. An investigation was also made of the vehicle characteristics obtained with control method (E) in which first-order advance control was applied to the front as well as the rear wheels.

Control methods

(A) Front-wheel steering
(B) Proportional steering control for rear wheels
(C) First-order delay control for rear wheels
(D) First-order advance control for rear wheels
(E) First-order advance control for both front and rear wheels

The control functions for each method are given in Table 1. The 4WS methods are compared below using the front-wheel steering system characteristics (A) as the baseline.

Table 1. Control functions.

Control system \ Functions		A_f	A_r
(A) Front-wheel steering	(——)	0	0
(B) Proportional	(----)	0	K_r
(C) 1st-order delay	(---)	0	$K_r/(1+T_r \cdot S)$
(D) 1st-order advance	(—–)	0	$K_r - T_r \cdot S$
(E) Front and rear wheel	(—◦—)	$K_f + T_f \cdot S$	$K_r - T_r \cdot S$

Proportional Steering Control (B)

With this method the steer angle of the rear wheels is controlled in proportion to that of the front wheels. The rear wheel control function, Ar, is equal to Kr (rear wheel steer angle/front wheel steer angle) in Eq. (1). The step response characteristics obtained for various Kr values are shown in Fig. 5-(B). The Kr value of 0.35 in this figure is equivalent to zero steady-state sideslip in Eq. (1).

As the rear wheel steer angle is increased the overshoot of the yaw rate is reduced and improved stability is obtained. The results indicate that the rise time characteristic for lateral acceleration is also improved. On the other hand, the rise time for the yaw rate is delayed and a large phase delay occurs in the yaw rate, as can be seen in Fig. 8.

These results were also made clear by analysis. In Eqs. (9), (13) and (15), $\omega 2$ decreases as the rear wheel angle (δr) increases, and there is a corresponding increase in ζ_2. This means that the phase delay in lateral acceleration is reduced, resulting in improved response and stability (Fig. 6-a).

On the other hand, the yaw rate response is reduced because of the phase delay in the frequency characteristics (Fig. 6-b). This occurs on account of the fact that $1/\tau_1$ becomes larger as δr increases, which is evident from Eqs. (8) and (17). The reason for this is that vehicles are generally designed with an understeer characteristic such that $bCr/aCf > 1$.

While the sideslip angle decreases under a steady-state condition, its transient value increases in the opposite direction from that of front-wheel steering, which is disconcerting to the driver. This signifies that greater lateral acceleration is being generated than the yaw rate and that there is a strong tendency for a parallel shift in the lateral direction (Fig. 5-(B)).

To reduce this feeling of disconcertion, it is necessary to decrease δr which is what happens when steady-state sideslip is generated. This means that a

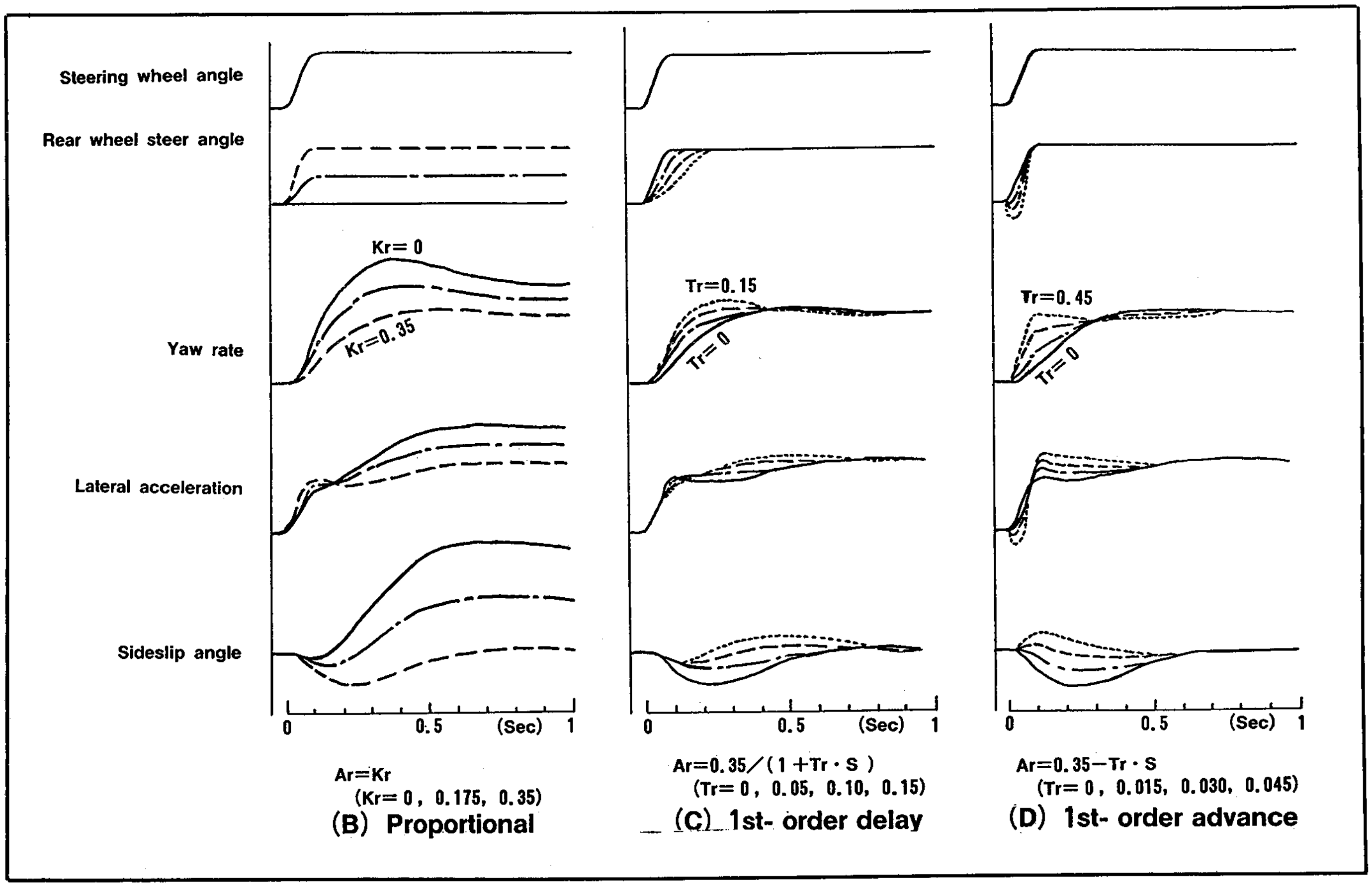

Figure 5. Step response with the steering characteristics

trade-off must be made between the disconcerting sensation and zero sideslip, as both of them cannot be resolved at the same time.

First-order Delay Control (C)

As the foregoing discussion has indicated, steering the rear wheels in the same phase as the front wheels causes the rear wheels to generate side force which hampers the generation of yaw. Consequently, this approach has the drawback that it delays the generation of yaw which is essential for turning (Figs. 5-(B) and 8).

The function of first-order delay control is to retard the generation of side force at the rear wheels until the necessary yaw rate is obtained.

The step response characteristics for first-order delay control are shown in Fig. 5-(C), where time constant Tr is taken as the parameter. The rear wheel control function Ar equals Kr/(1 + Tr•S). The value of Kr is determined by Eq. (1). Although lateral acceleration response is somewhat lower, a large improvement is seen in the yaw rate response and the sideslip angle is reduced even during transient conditions. Since an excessively large time constant τ causes yaw rate overshoot, it is concluded that an optimum value exists for Tr. The frequency characteristics shown in Fig. 8 also indicate improvements in both the yaw rate and lateral acceleration.

First-order Advance Control (D)

As described above, first-order delay control improves the yaw rate response by retarding the generation of side force at the rear wheels. By contrast, first-order advance control, Ar = Kr − Tr•S, is a more active approach to improving the yaw rate response. With this method the rear wheels are steered in the opposite phase from the front wheels under transient conditions.

The step response characteristics obtained with this method are shown in Fig. 5-(D). As Tr increases and the opposite phase components become larger, the rise time characteristic of the yaw rate is further improved. This effect is prominently seen in the frequency characteristics, especially in the high frequency region.

If Tr becomes too large, however, excessive lateral acceleration is generated in the opposite direction, which creates a disconcerting sensation for the driver. It is therefore concluded that a optimum value exists for Tr.

First-order Advance Control for Both Front and Rear Wheels (E)

The results presented so far have made it clear that both response and stability can be substantially improved by steering the rear wheels. However, reducing

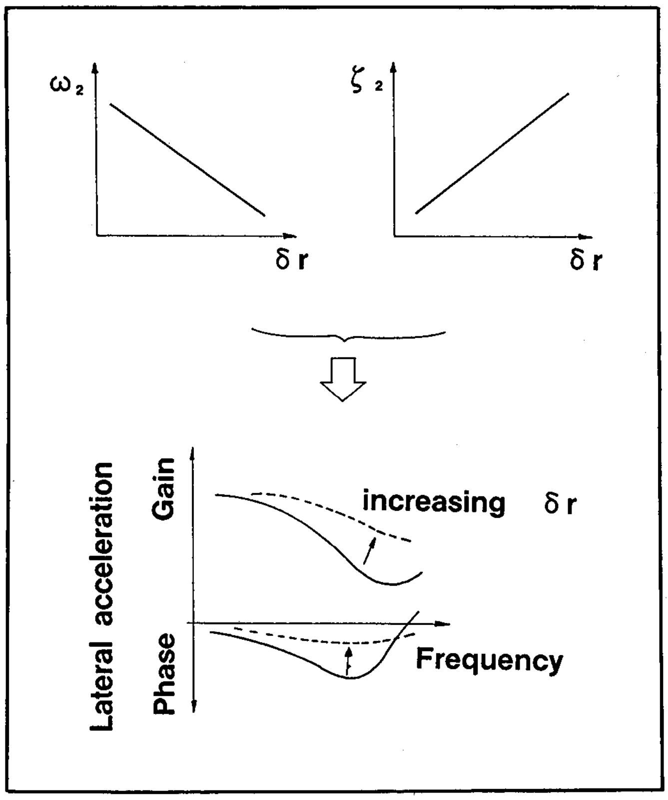

Figure 6-a. Changes in lateral acceleration characteristics

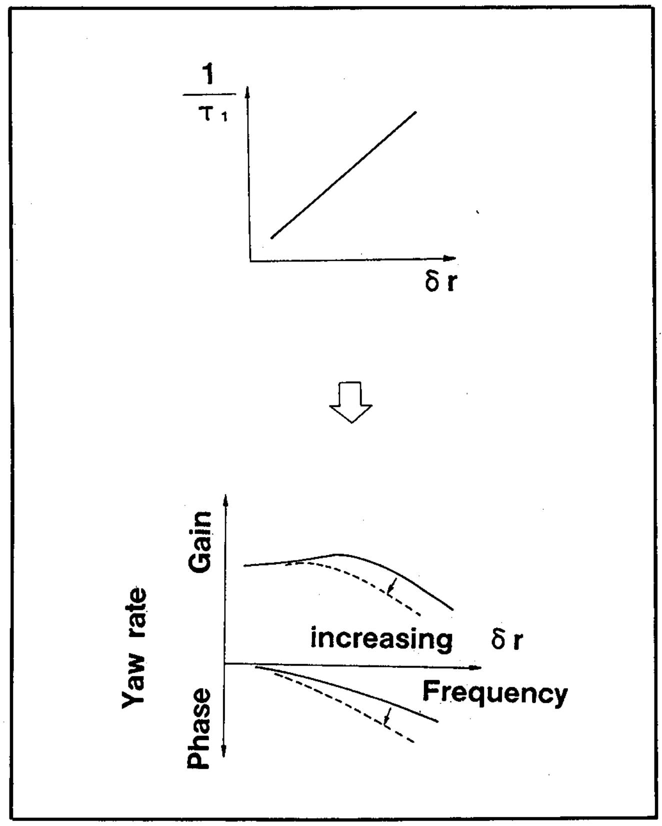

Figure 6-b. Changes in yaw rate characteristics

the steady-state sideslip to zero causes the yaw rate gain to drop and cornering performance deteriorates as a result. Thus setting the steady-state sideslip at zero would cause problems in the practical use of the vehicle. The side force generated by the rear wheels makes it impossible to improve the yaw rate and lateral acceleration simultaneously, as was made clear by Eqs. (2) and (3). This drawback can be overcome by applying first-order advance control to the front wheels as well.

Theoretically, it should be possible to build an ideal car that would have flat frequency characteristics for both the yaw rate and lateral acceleration. Such a vehicle would provide quick response and excellent stability. Further, its sideslip angle would be zero even under transient conditions, i.e., it would always point straight ahead. As a result, the vehicle is much easier to drive.

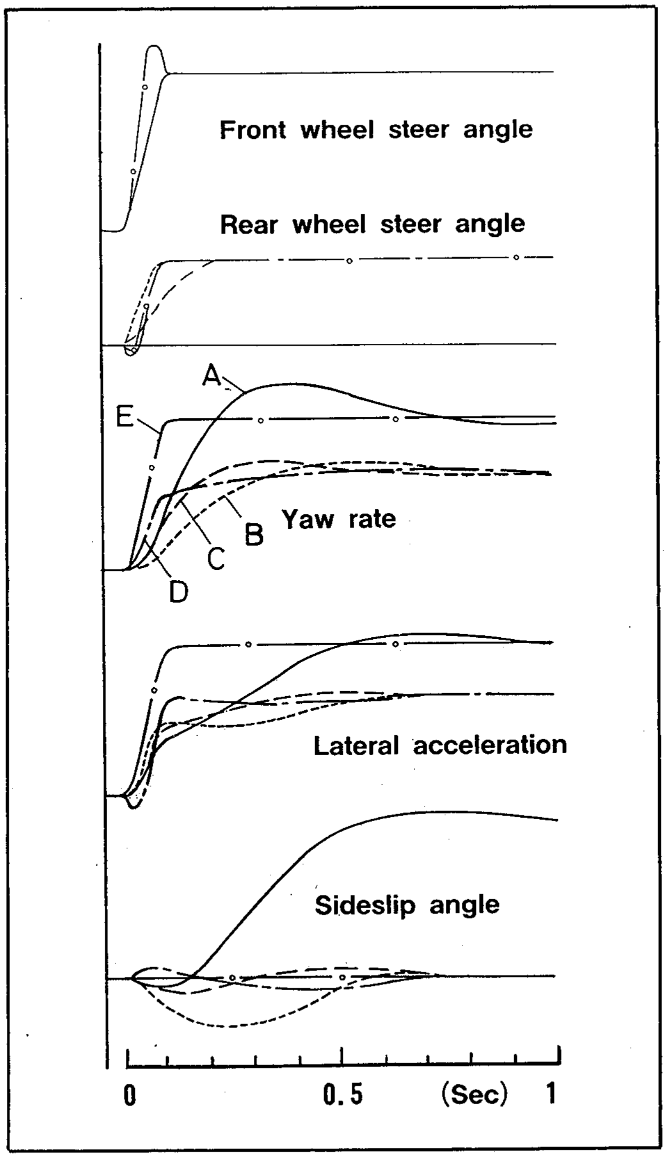

Figure 7. Step response (simulated)

The control functions that would provide such vehicle characteristics can be found by solving continuous equations (18)-(24) for Af and Ar.

$$(1+\mathrm{Af})\cdot\mathrm{Hf}+\mathrm{Ar}\cdot\mathrm{Hr}=\dot{\varphi}_0 \tag{18}$$

$$(1+\mathrm{Ar})\cdot\mathrm{Gf}+\mathrm{Ar}\cdot\mathrm{Gr}=\alpha_0 \tag{19}$$

$$\mathrm{Hf}=\{(\mathrm{aMCf})\cdot\mathrm{S}+(\ell\mathrm{CfCr/V})\}/\triangle \tag{20}$$

$$\mathrm{Hr}=\{-(\mathrm{bMCr})\cdot\mathrm{S}-(\ell\mathrm{CfCr/V})\}/\triangle \tag{21}$$

$$\mathrm{GF}=\{(\mathrm{CfI})\cdot\mathrm{S}^2+(\mathrm{a}\ell\mathrm{CfCr/V})\cdot\mathrm{S}+(\ell\mathrm{CfCr})\}/\triangle \tag{22}$$

$$\mathrm{Gr}=\{(\mathrm{CrI})\cdot\mathrm{S}^2+(\mathrm{b}\ell\mathrm{CfCr/V})\cdot\mathrm{S}-(\ell\mathrm{CfCr})\}/\triangle \tag{23}$$

$$\triangle=(\mathrm{MI})\mathrm{S}^2+[\{(\mathrm{Cf}+\mathrm{Cr})\mathrm{I}+(\mathrm{a}^2\mathrm{Cf}+\mathrm{b}^2\mathrm{Cr})\mathrm{M}\}/\mathrm{V}]\cdot\mathrm{S} + \ell^2\mathrm{CfCr}/\mathrm{V}^2+(\mathrm{bCr}-\mathrm{aCf})\mathrm{M} \tag{24}$$

where $\dot{\varphi}_0$ and αo are the steady state values of $\dot{\varphi}$ and α for front-wheel steering.

This solution is in the form of 4th-order/3rd-order, but it can be transformed and rearranged to yield Eqs. (25) and (26), which are 1st-order advance equations.

$$\mathrm{Af}=\mathrm{Kop}+(\mathrm{Cr/Cf})\cdot\mathrm{Top}\cdot\mathrm{S} \tag{25}$$

$$\mathrm{Ar}=\mathrm{Kop}-\mathrm{Top}\cdot\mathrm{S} \tag{26}$$

Here,

$$\mathrm{Kop}=\frac{\mathrm{aMCfV}^2+\mathrm{b}\ell\mathrm{CfCr}}{\ell^2\mathrm{CfCr}+(\mathrm{bCr}-\mathrm{aCf})\mathrm{MV}^2} \tag{27}$$

$$\mathrm{Top}=\frac{\mathrm{CfIV}}{\ell^2\mathrm{CfCr}+(\mathrm{bCr}-\mathrm{aCf})\mathrm{MV}^2} \tag{28}$$

As a result, flat vehicle characteristics are obtained, as shown in Fig. 8.

Comparison of Control Methods

The step response and frequency characteristics for each control method are shown in Figs. 7 and 8.

(1) Proportional steering control (B)

The yaw rate overshoot is reduced and the time required for yaw motion convergence is shortened. The steady-state sideslip angle is reduced, but the transient angle increases in the opposite direction. While the rise time characteristic for lateral acceleration is improved, a larger delay occurs in the yaw rate as compared with front-wheel steering. This delay appears as a phase delay in the frequency characteristics.

(2) First-order delay control (C)

This method delivers virtually the same steady-state characteristics as proportional steering control and it also improves the yaw rate response. In comparison with front-wheel steering, it improves the yaw rate and lateral acceleration characteristics.

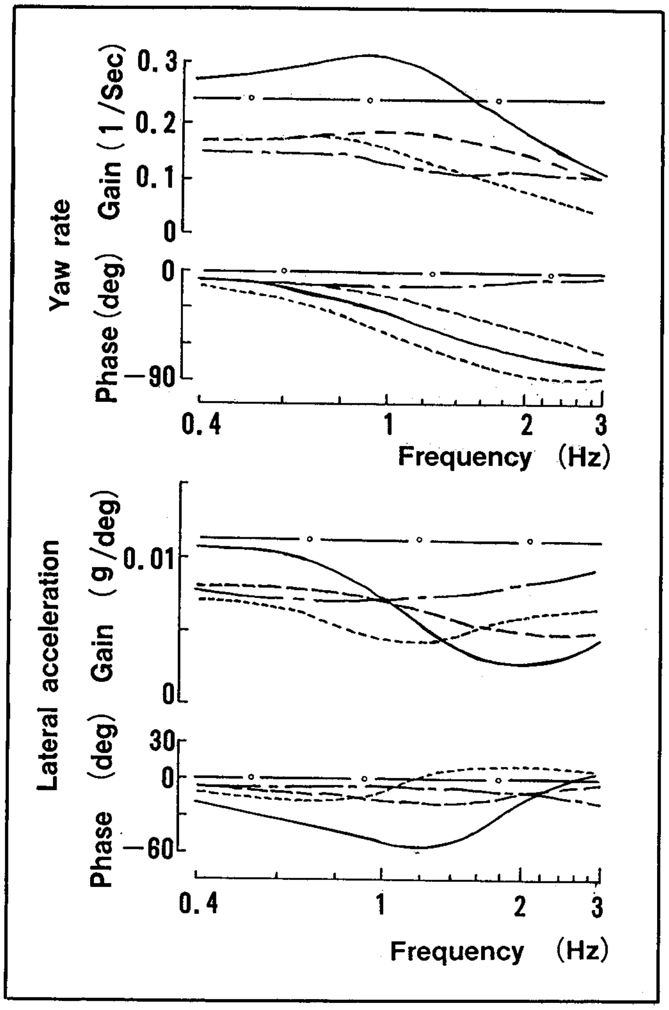

Figure 8. Frequency characteristics (simulated)

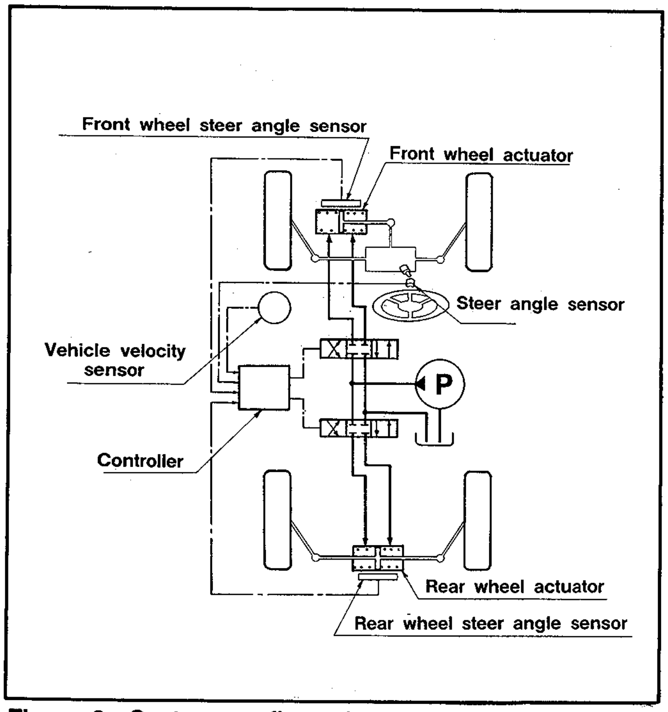

Figure 9. System configuration

(3) First-order advance control (D)

This method improves the yaw rate response even more than first-order delay control. An excessive increase in response, however, causes a momentary occurrence of lateral acceleration in the opposite direction. This suggests that an optimum value exists for yaw rate response.

(4) First-order advance control for both front and rear wheels (E)

This method provides an ideal control system that eliminates all phase delay and overshoot.

Experimental Four-Wheel-Steering Vehicle

Vehicle System

The system configuration of the vehicle used in conducting experiments is illustrated in Fig. 9. Hy-

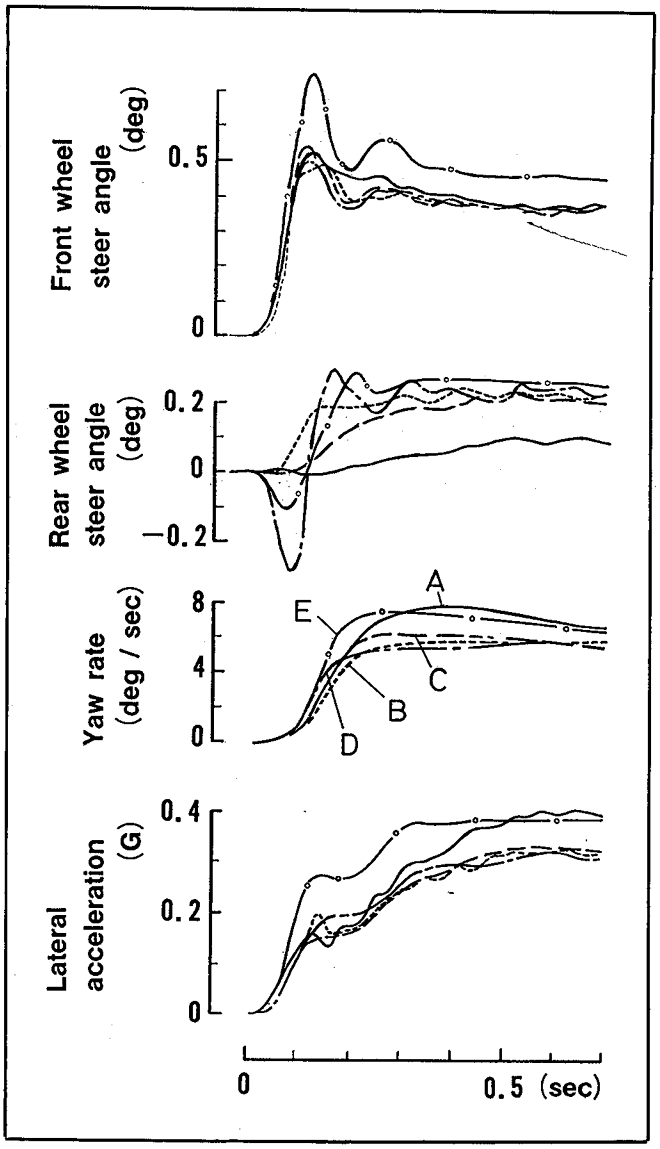

Figure 10. Step response (experimental)

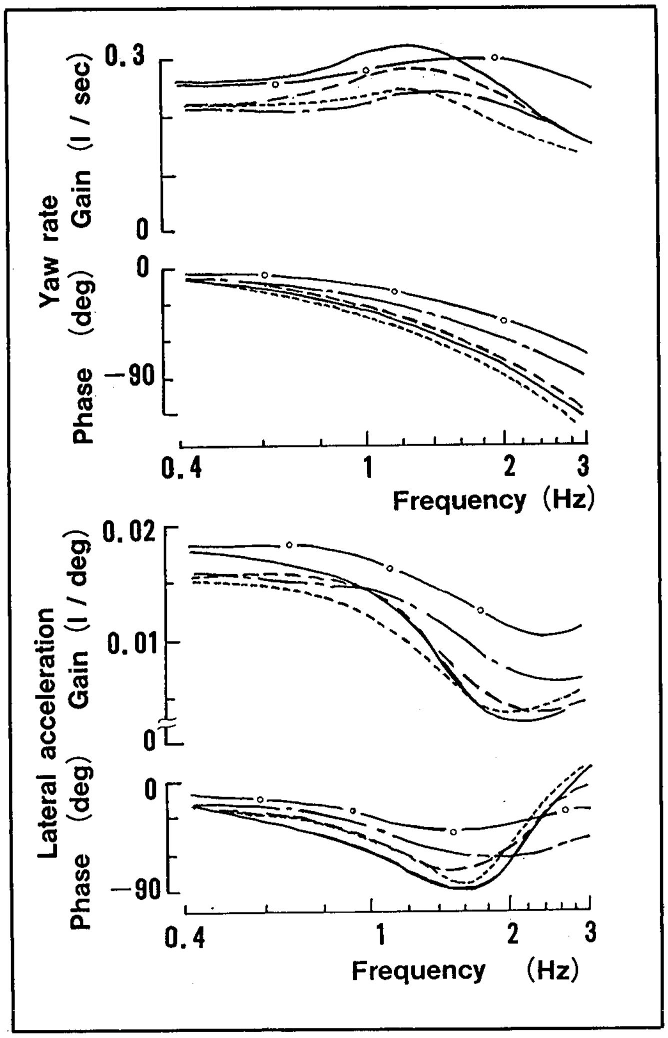

Figure 11. Frequency characteristics (experimental)

draulic actuators are provided at both the front and rear wheels. Based on the vehicle velocity and steer angle signals, the controller calculates the optimum control signals for controlling the front and rear wheel steer angles.

Experimental Results

The step response and frequency characteristics are shown in Figs. 10 and 11. Good agreement is seen with the simulated results given in Figs. 7 and 8.

The best yaw rate rise time characteristic is obtained with first-order advance control for both front and rear wheels, followed by first-order advance and first-order delay control in that order.

Lateral acceleration shows a two-stage rise characteristic, with front and rear wheel advance control providing the best performance in both stages. With first-order advance control there is a momentary delay in the initial rise stage, but large lateral acceleration is obtained in the second stage owing to the generation of the yaw rate.

Table 2. Subjective evaluation of four-wheel steering control.

Control system	Description
(B) Proportional	Although high stability is obtained, a disconcerting lateral shift occurs when the vehicle is steered.
(C) 1st-order delay	High levels of stability and response are obtained. Vehicle behavior is natural.
(D) 1st-order advance	Exceptionally good response is obtained which provides crisp, sharp steering. High gain is also felt.
(E) Front and rear wheel	Good balance of stability and response at high levels. Steering characteristics are natural and accurate tracking faithful to the driver's intention is obtained.

The results of a subjective evaluation of stability and steering response are given in Fig. 12 and brief summaries of the evaluators' comments are given in Table 2.

The results indicate that stability was greatly improved by steering the rear wheels in the same phase as the front wheels. When the same amount of control (proportional amounts) was applied, some difference was seen depending upon how the transient yaw rate characteristic was given, however, the difference between systems was very small.

The evaluation of steering response improved as more control was applied to cause the tire side force to be increasingly generated in the direction of the transient yaw rate (same phase for front wheels and opposite phase for rear wheels). Thus first-order advance control was evaluated the highest among the rear wheel control methods.

The front and rear wheel advance control method provided a good balance of stability and response at high levels. Because it kept the vehicle's sideslip angle to zero at all times (i.e. the vehicle was constantly pointed straight ahead), the evaluators commented that the vehicle was easy to drive and that it provided accurate tracking performance.

Development of HICAS System

System concept

The High-Capacity Actively Controlled Suspension (HICAS) system has been built around the first-order delay control method (C) described before.

The results of the authors' research to date have clarified the following points:(6)

- The frequency characteristics for the steer angle and steering force are virtually the same.

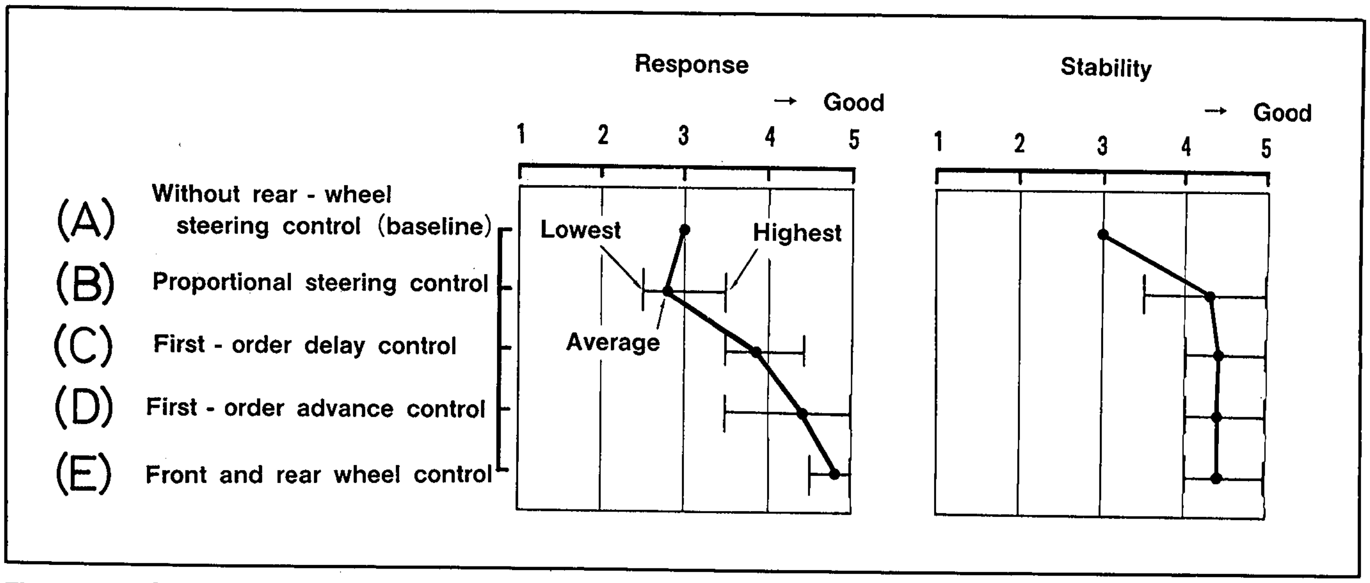

Figure 12. Subjective evaluation of four-wheel steering control

- A larger improvement is obtained in the stability factor Ks, damping ratio ζ, and natural frequency ω_n by controlling the rear wheel steer angle in proportion to the steering force rather than the steer angle (Table 3 and Fig. 3).

HICAS has therefore been designed to control the rear wheel steer angle in proportion to the steering force, applying approximately a first-order delay, as well as in proportion to the vehicle velocity. This approach achieves both high levels of response and stability. The first-order delay has been accomplished by optimizing the delay element of the hydraulic system.

System Outline

The system is illustrated schematically in Fig. 14. In general, the steer angle is small during high-speed driving. Moreover, as Fig. 1 illustrates, the steer angle at the rear wheels is small. This means that only a small steer angle is required for the rear wheels at high speeds. As a result, the rear wheel steer angle can be set within the range of the compliance steer angle. This eliminates the need for any complicated steering mechanism.

In addition, this design assures an exceptionally high level of safety even if the system should fail. In the event a failure should occur, the system will still function as an ordinary rear suspension.

Table 3. Effects of rear-wheel steering on yaw rate transfer function.

	(A) Front-wheel-steering model ($\delta r=0$)	(B) Model for rear-wheel steering proportional to front wheel steer angle (rear wheer angle $\delta r=kr\frac{\Theta}{N}$)	(B') Rear-wheel-steering model with steering force feedback (rear wheel steer angle $\delta r=kaFfT$) (T: Pneumatic trail)
Stability factor Ks	$\frac{M}{\ell^2}\left(\frac{b}{Cf}-\frac{a}{Cr}\right)$	$\frac{M}{\ell^2}\left(\frac{b}{Cf}-\frac{a}{Cr}\right)$	$\left[\frac{M}{\ell^2}\left(\frac{b}{Cf}-\frac{a}{Cr}\right)+\frac{MbkaT}{\ell^2}\right]$
Steady-state yaw rate gain A_0	$\frac{V}{N\ell}\cdot\frac{1}{(1+KsV^2)}$	$\frac{V}{N\ell}\cdot\frac{1-kr}{1+KsV^2}$	$\frac{1}{N\ell}\cdot\frac{1}{1+(Ks+\frac{MbkaT}{\ell^2})V^2}$
Time constant with first-order lead term ν	$\frac{aVM}{Cr\ell}$	$\frac{MV}{(1-krb)\ell}\left(\frac{a}{Cr}-kr\frac{b}{Cf}\right)$	$\frac{MV}{Cr\ell}(a-bkaCrT)$
Damping ratio ζ	$\frac{1}{2}\cdot\frac{(Cf+Cr)Iz+(Cfa^2+Crb^2)M}{\sqrt{MIzCfCr\ell}\sqrt{1+KsV^2}}$	$\frac{1}{2}\frac{(Cf+Cr)Iz+(Cfa^2+Crb^2)M}{\sqrt{MIzCfCr\ell}\sqrt{1+KsV^2}}$	$\frac{1}{2}\cdot\frac{(Cf+Cr)Iz+(Cfa^2+Crb^2)M+KaCfCrT(Iz-abM)}{\sqrt{MIzCfCr\ell}\cdot\sqrt{1+(Ks+\frac{Mb\,kaT}{\ell^2})V^2}}$
Natural frequency ωn	$\sqrt{\frac{CfCr\ell^2}{MIzV^2}(1+KsV^2)}$	$\sqrt{\frac{CfCr\ell^2}{MIzV^2\ell}(1+KsV^2)}$	$\sqrt{\frac{CfCr\ell^2}{MIzV^2}\left\{1+(Ks+\frac{Mb\,kaT}{\ell^2})V^2\right\}}$
$\zeta\times\omega n$	$\frac{1}{2}\cdot\frac{(Cf+Cr)Iz+(Cfa^2+Crb^2)M}{MI_zV}$	$\frac{1}{2}\cdot\frac{(Cf+Cr)I_z+(Cfa^2+Crb^2)M}{MI_zV}$	$\frac{1}{2}\cdot\frac{(Cf+Cr)Iz+(Cfa^2+Crb^2)M+kaCfCrT(Iz-abM)}{MIzV}$

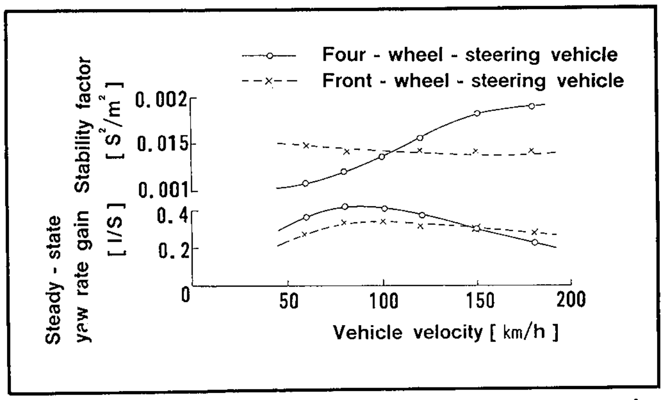

Figure 13. Stability factor and vehicle response gain

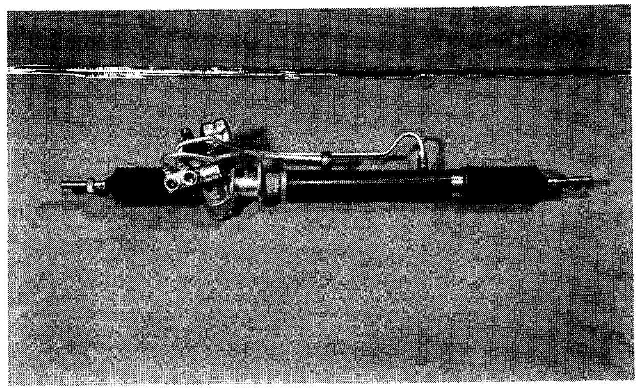

Figure 15. Double-valve steering gear

This system is called an active compliance steer control system and it plays a vital role in the HICAS system.

System Structure

Hydraulic Systems. Two hydraulic systems are employed, with one providing hydraulic pressure for power steering at the front wheels and the other used for steering the rear wheels. The hydraulic pump is built with a tandem structure.

Rear Wheel Steer Control Actuator. The hydraulic control valve for the rear wheels has a double-valve structure, as illustrated in Fig. 15. The valve construction is the same as that of the power steering valve at the front wheels, though the two valves differ in their performance characteristics.

The hydraulic pressure is controlled proportional to the steering force, and it is transferred to the hydraulic cylinder with a first-order time lag characteristic. The compact hydraulic cylinder presses against the rubber insulators that are installed where the suspension is attached to the vehicle body. Through the force generated by the hydraulic cylinder and the rigidity of the insulators, hydraulic pressure is fed back to control the rear wheel steer angle (δr) in accordance with Eq. (29) given in Fig. 14. As a result, the rear wheel steer angle obtained is virtually proportional to the lateral acceleration.

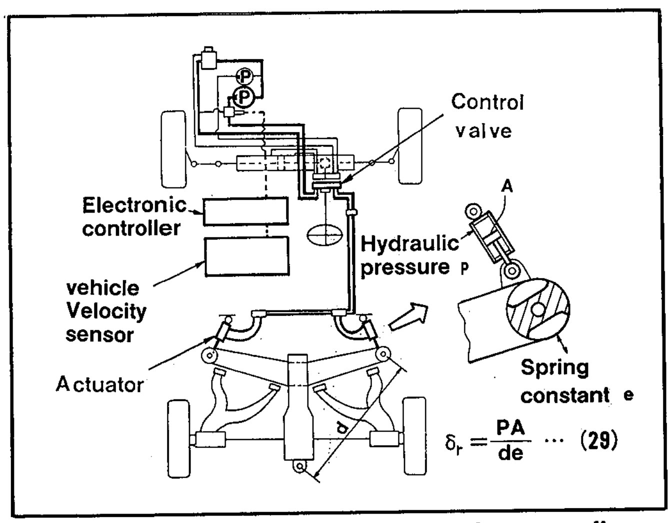

Figure 14. Outline of rear-wheel active compliance steer system with steering force feedback (HICAS)

Electronic Controller. The rear wheel steer angle is controlled relative to the vehicle velocity. As shown in Fig. 17, the steer angle increases as the vehicle velocity increases. This is accomplished by means of a bypass valve which is controlled by an electronic controller. The controller operates according to the signal received from a vehicle velocity sensor. This makes it possible to raise the vehicle's natural stability factor as the vehicle velocity increases, thereby improving vehicle stability.

The steering gear ratio is set at a small 13.3, which provides an improved yaw rate gain in low and intermediate speed ranges. The stability factor in the low-speed region where the rear wheels are not steered is also set on the small side at about 0.001 s^2/m^2. This contributes to better vehicle handling properties at low and intermediate speeds.

As a result, precise steering controllability is obtained at low speeds and also excellent vehicle stability is provided in the high speed range.

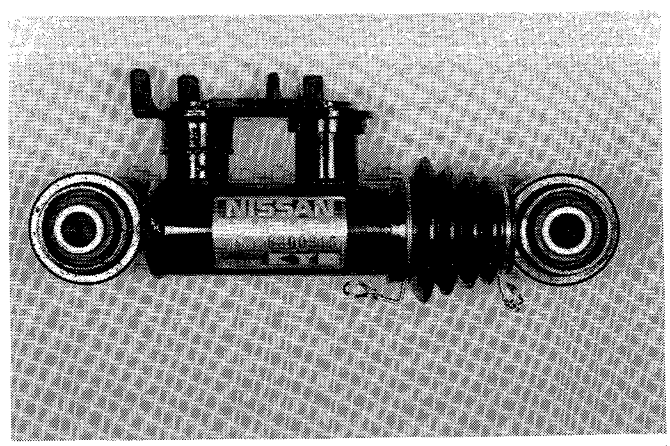

Figure 16. Hydraulic cylinder

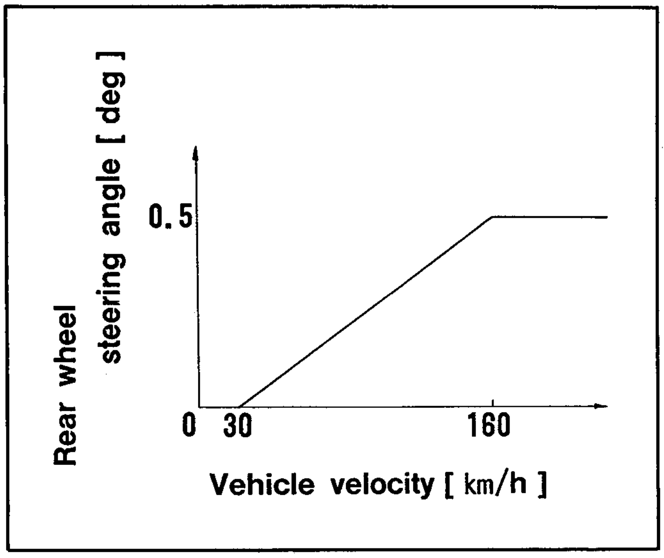

Figure 17. Rear wheel steering angle vs. vehicle velocity

Conclusion

1. Active control over the rear wheels has been found to be extremely effective in improving handling and stability. Two important factors in improving vehicle steering response are the steer angle ratio of the rear wheels and transient steering characteristics.
2. As far as rear wheel control alone is concerned, improved response can be obtained by applying a transient phase delay (first-order delay). Response is further improved with first-order advance control, whereby transient opposite-phase steering is applied to the rear wheels.
3. The degree of control latitude is further increased by applying first-order advance control to the front wheels, as well as to the rear wheels. Simulation results showed that this was the ideal control system configuration, in that it provided flat frequency characteristics for the yaw rate and lateral acceleration. Such a control system eliminates response delay and keeps the vehicle's sideslip angle at zero at all times, making the vehicle easy to drive.
4. A practical rear wheel control system, HICAS, has been developed that achieves substantial improvements in vehicle handling and stability, while assuring a high degree of safety and low cost. This is accomplished within the geometry of the rear suspension by applying feedback active compliance steer control. With this approach the rear wheels are steered with a first-order delay according to the front steering force and the vehicle velocity.

Nomenclature

ℓ	: wheelbase
a,b	: distance between center of gravity and front and rear wheels
Cf,Cr	: cornering power of front and rear wheels
M	: vehicle mass
I	: vehicle yaw moment of inertia
N	: steering gear ratio
θ	: steering wheel angle
$\delta f, \delta r$	: steer angle of front and rear wheels
y	: lateral displacement at center of gravity
Ff,Fr	: side force of front and rear wheels
V	: vehicle velocity
Hf,Hr	: yaw rate transfer functions of front and rear wheels
Gf,Gr	: lateral acceleration transfer functions of front and rear wheels
Af,Ar	: control functions for front and rear wheel steer angle
S	: Laplace operator
$\dot{\varphi}$	: yaw rate
$\dot{\varphi}_0$	: yaw rate constant of baseline vehicle
α	: lateral acceleration
α_0	: lateral acceleration constant of baseline vehicle

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Objective Testing for Vehicle Brake Balance Performance

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Abstract

This paper will describe GM technology for objective measurement of brake balance. Facilities will include the road transducer pad and torque wheel instrumentation. The need for multiple approaches to brake balance testing will be defined. Limitations of past methods, such as skid checks, will also be described. Statistical properties of data bases derived from these tests for relatively large numbers of production vehicles will be defined. This will assist other research activities in studying the primary safety performance for vehicles operating in North America.

Introduction

Significance of Vehicle Brake Balance

Conventional motor vehicles rely upon friction forces generated at the tire road interface for control behavior including starting, turning, and stopping. At each tire to road contact, a combination of longitudinal and lateral forces are necessary to generate a vehicle maneuver. For any particular tire to road interface, controllable lateral and longitudinal forces may be sustained as long as the vector resultant of these forces is less than or equal to the limit of adhesion for that particular interface. The limit of adhesion is usually expressed as the product of the normal force and the friction coefficient of the interface, i.e., $L = mu \times N$.

The ratio of the vector sum of the lateral and longitudinal forces acting on a tire divided by the normal force acting on that same tire is defined as the adhesion utilization. Adhesion utilization in straight line braking is directly related to vehicle brake balance and is regulated in the Common Market. Vehicle brake balance is defined as the distribution of braking forces between the front and rear axles that results from the application of force at the service brake pedal. Vehicle brake balance is one important factor in establishing the limits of control for both path and attitude of a vehicle and also in determining the theoretical limits of a vehicle deceleration under braking. However, vehicle brake balance is not the exclusive factor in determining the magnitude or the distribution of forces between the tires and the road on the front vs the rear axles. Such exogenous variables as road camber and grade, as well as power train characteristics, all may influence the distribution of forces both normal and parallel to the tire road interface. Nevertheless, brake balance is one factor that can influence the limit performance capability of motor vehicles.

The shortest achievable stopping distances are realized when a vehicle is ideally balanced. Any other balance configuration will inevitably result in longer stopping distances. While unanimity within the brake community as to what tradeoffs best meet safety needs and customer expectations may not be achievable, or desirable, developing any consensus depends upon having credible balance information from objective tests.

Vehicle brake balance is known to be a function of many operational variables including temperature of the braking elements, work history of the system, vehicle speed, loading condition, and deceleration. The challenge for the brake engineer is to select a combination of foundation brake and system components that best meets the customer's needs over the broad range of operating conditions to which the vehicle is exposed. Any measure of vehicle brake balance that comprehends only a discrete set of such operating conditions is limited in its utility.

Representations of Vehicle Brake Balance

Vehicle brake balance may be represented in many ways. The ECE R13 annex 10 required adhesion utilization format, and the normalized brake efficiency formats are most common. The relationships between the representation formats has been reviewed elsewhere (1) and will not be repeated here. Vehicle brake balance is normally presented up to the limit of adhesion on the first axle locking both wheels as a function of vehicle deceleration. Examples of vehicle brake balance formats are shown in figures 1 and 2.

Multiple vehicle tests of brake balance or multiple tests on individual vehicles are most easily presented in a cloud chart in the normalized brake efficiency format. This representation format permits compari-

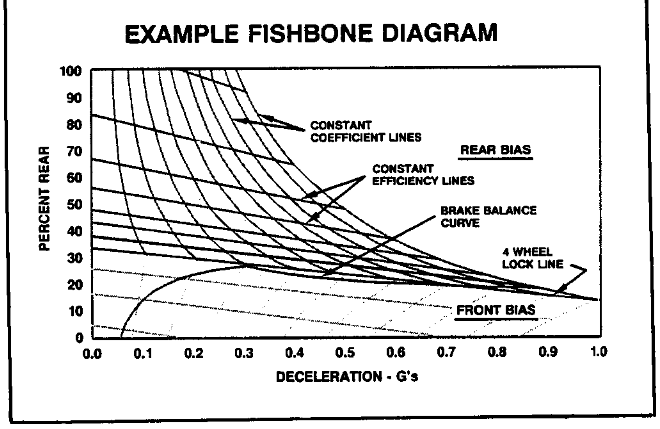

Figure 1

sons of multiple loading conditions of the same vehicle or multiple vehicle configurations most conveniently. An example cloud chart is shown in figure 3.

The objective measurement of vehicle brake balance has assumed an increasingly significant role in brake system design and development in recent years. The development of the Road Transducer Pad (RTP) and on-board digital data acquisition have greatly expanded objective test capability. The application of this objective test technology has provided new insights into the performance capability of vehicle brake systems, as well as identified important limitations for historical test practises.

Historical Assessment of Vehicle Brake Balance

"Skid Checks" or Wheel Lock Sequence

Historically, the most frequently used test technology was the wheel lock sequence or "skid check" procedure. The typical practise was to install a decelerometer and apply the service brakes at an increasing rate until the first axle locks. Usually, the first axle lock was detected by the driver via an audible squeal, a yaw cue, from a loss of steering control, or some combination of the above.

Depending upon the operator's knowledge of test weights, selection of a level roadway of generally uniform coefficient, ability to precisely determine the first axle to lock from either skid marks or wheel speed instrumentation, and skill in determining the vehicle deceleration at first axle lock, the test might yield a binary logic result of either front or rear axle locking first. Depending upon the apply rate, the vehicle pitch dynamics, and other factors, repeated tests might yield either result on a vehicle which is nearly ideally balanced. Minor variations in the speeds at first axle lock, the temperature of the braking elements, road camber, wind, etc., all could produce potentially conflicting results as to which axle locks first.

Because the tire to road peak friction coefficients are generally unknown at the time of test, the wheel lock or "skid check" procedure does not permit an estimation of vehicle brake efficiency or magnitude of deviation from ideal brake balance. Like many other test practises employed historically, the principal merit of the wheel lock sequence or "skid check" test is its expediency. It does not provide a useful characterization of vehicle brake balance or adhesion utilization.

Furthermore, this test method involves substantial cost. Since vehicle brake balance varies as a function of deceleration, any attempt to evaluate brake balance by "skid check" testing would have to be conducted on a range of known coefficient surfaces that are both level and uniform. Test surfaces have to be developed, constructed, maintained, and regularly monitored to provide appropriate vehicle decelerations. Surfaces for this testing may be wetted, which requires the construction of watering systems to insure uniform depth and coverage. Likewise, a traction trailer and two vehiclesmay be required to regularly measure the traction limit of such test surfaces. The costs associated with the construction and maintenance of a collection of test surfaces is known to exceed that associated with objective RTP or instrumented vehicle test methods.

As a result of the application of new objective test methods described later, we now know that a fundamental flaw exists in the application of "skid check" or wheel lock sequence tests as a means of determining vehicle brake balance. In order to qualify as an objective test, two criteria must be met. The first of these, statistical reliability, simply requires that the test practice employed must yield consistent results when repeatedly applied to the same subject, i.e., reproducibility. The second element, statistical validity, means that the test measure must be a valid indicator of the subject characteristic to be studied, i.e., accuracy. For differing reasons, the "skid check" or wheel lock sequence fails to meet either of these necessary requirements for objectivity.

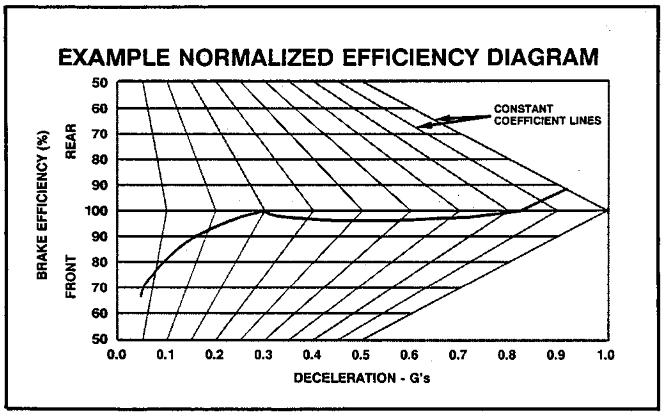

Figure 2

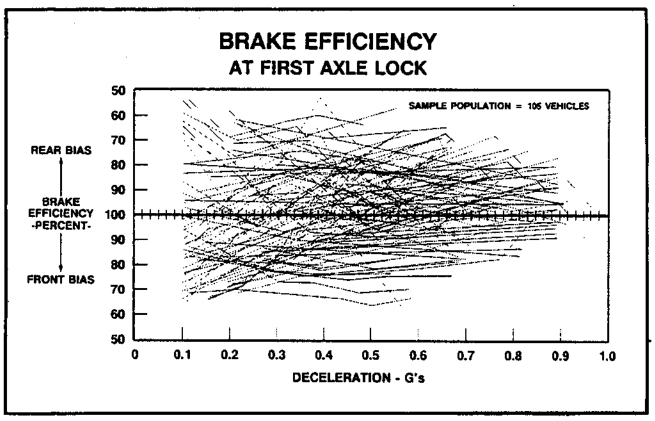

Figure 3

Detailed objective testing of numerous vehicle brake systems has shown that all practical vehicles have a brake balance distribution that spans a range of plus or minus 5 percentage points of rear braking at any given deceleration. A "skid check" test can only meet statistical reliability and validity requirements when the vehicle's brake balance distribution is sufficiently far from ideal that the influence of both intrinsic and exogenous variables precludes an erroneous result. For a vehicle whose brake balance is close to the ideal, the skid check procedure has a probability of producing either result that approaches 50%. The skid check procedure may sort extreme brake balance distributions, but unfortunately is quite likely to produce an erroneous result for vehicles with nearly ideal brake balance distributions.

Brake Balance by Nominal Design Calculation

Limitations on vehicle brake balance or adhesion utilization are specified in many countries for new vehicles. While the particular details of type approval and compliance enforcement vary, the basic requirements are generally similar to those of Annex 10, of ECE R13. At the risk of oversimplification, the essential elements of this regulatory requirement are that the nominal vehicle shall be designed to be front axle limited at both the lightly loaded (driver only) and laden (GVWR) conditions with exceptions permitted below 0.10g, between 0.30 and 0.45g, and above 0.80g. The vehicle shall also be capable of achieving a vehicle deceleration of 5.8 m/sec/sec for a tire to road coefficient of 0.80. The front axle adhesion utilization is limited by an upper bound given by $k = (z + 0.07)/0.85$ where z is the vehicle deceleration in g's. The format for presentation of the adhesion utilization curves in Annex 10 are shown in figure 4. The theoretical effect of these requirements is to limit the range of vehicle brake balance that is permitted over a broad range of loading conditions and vehicle decelerations.

This approach can be objective if brake factors (brake specific torques) are properly measured. Various techniques ranging from simple friction machines to fully instrumented vehicle tests can be utilized to determine brake specific torques objectively. However, no consensus exists at this time regarding the test method, test schedule to be employed, or how to deal with variability in test results. Objectively determined brake specific torques can be used to accurately project the nominal vehicle brake balance. For example, an instrumented vehicle test for determining brake specific torques has been proposed (2). Using this technique, a range of brake specific torques was determined as shown in figures 5 and 6. When the average values for both front and rear brakes are used in the calculations, the nominal vehicle brake balance is shown in figure 7. This may be compared to the average torque balance measured in the output test. The slight distinction between the two curves at low decelerations is due to the drivetrain effects on the measured values. The vehicle test is run with the drivetrain connected, i.e., with the transmission in gear. This technique does meet the statistical reliability and validity requirements for objectivity, and has been employed by General Motors for such determinations for several years.

Objective Testing for Vehicle Brake Balance

Road Transducer Pad (RTP)

The RTP was developed so that objective brake balance measurements could be obtained accurately and quickly on large numbers of vehicles without onboard instrumentation or modification. The RTP is an instrumented section of roadway which measures the braking forces developed at each wheel as a vehicle is driven across with the brakes applied. Detailed descriptions of the method, development and equipment are presented in (3).

The RTP allows brake balance to be determined on many vehicles in a practical manner for building statistical data bases and establishing vehicle variabil-

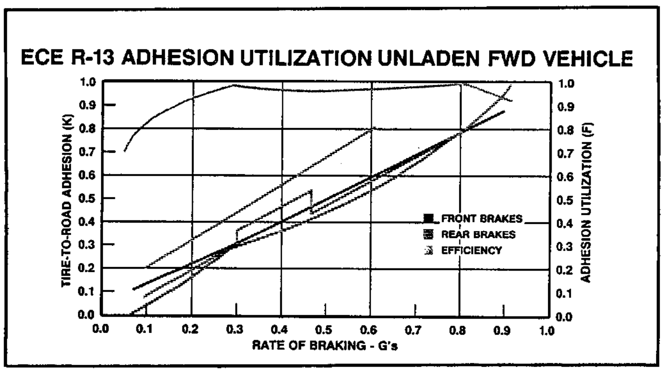

Figure 4

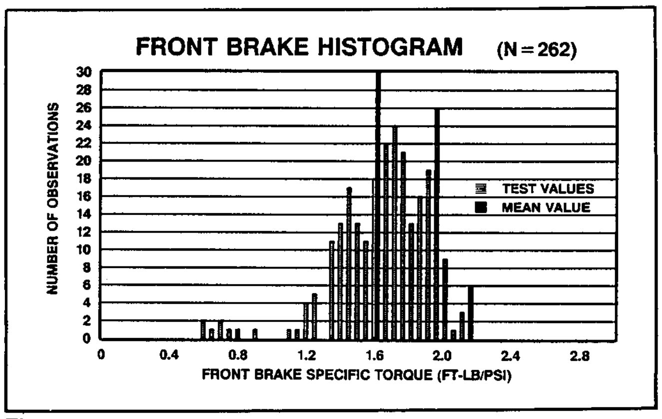

Figure 5

ity. Since no modification is required, customer vehicles and vehicles under durability assessment can be measured without disruption of vehicle hardware.

Objective brake balance assessment can be performed accurately and efficiently. Since actual road forces are measured, knowledge of dynamic tire radius is not required. Vehicle measurements are limited to a series of "snapshots" as the vehicles pass over the transducers.

Instrumented Vehicle Tests

Instrumented vehicle testing has been used successfully by General Motors to objectively measure brake balance. Torque wheels combined with other system transducers, and onboard digital data acquisition hardware, permit a complete vehicle brake system analysis to be performed relatively quickly. With this method, instrumented wheels replace the normal vehicle wheels and torque developed during braking is measured at each wheel. Since the torque transducers are rotating, slip ring devices are employed to couple the instrumented wheels to data acquisition equipment residing onboard the vehicle. Measurements are made at various decelerations and analyzed to objectively assess brake balance through the entire brake apply. Axle lock sequence for various tire-road friction levels and braking efficiencies are easily calculated from the data.

Dynamic wheel torque measurements and loaded tire radii are used to compute road forces at each wheel. Four sensors measure individual wheel velocities and a fifth wheel at the rear of the car measures vehicle speed. Other instrumentation is included to measure hydraulic pressures, brake temperatures and pedal apply force. Data is acquired by equipment housed in the passenger compartment.

Instrumented vehicles can provide objective assessments of total brake system performance. Efficiency and axle lock sequence for all road friction coefficients are easily obtained from proper analysis of torque wheel tests on a single high friction surface. Brake balance assessment requires knowledge of dynamic tire radius because wheel torques rather than actual road forces are measured. An array of custom transducers must be acquired, maintained and installed on test vehicles which makes the method less practical than the RTP for measuring large numbers of vehicles for statistical analyses. However, this method with its accurate and rather extensive results is often required for in-depth investigations of total brake system performance on development vehicles.

Correlation Between Torque Wheels and the RTP

To evaluate the statistical validity of the RTP and the use of torque wheel/digital data acquisition methods for brake balance assessment, a series of tests were conducted on a vehicle equipped with full instrumentation including torque wheels, pedal force and line pressure transducers, and event marker photocell detectors. A series of constant deceleration snubs were made over the RTP while the digital data acquisition system simultaneously recorded the wheel torques. A dynamic rolling radius correction was made to the torque wheel data for conversion to road force. The brake balance determined from both the RTP and the converted torque wheel data is shown in figure 8. Here one can see general agreement over a broad range of vehicle decelerations between the two test techniques.

Statistical analysis of the brake balance measurements revealed an average difference of 0.05 percentage points with a standard deviation of 0.744 percentage points. Perfect agreement between the two methods would have produced an average difference of 0 and a finite standard deviation due to random error only.

Discussion

The selection of a particular objective test method to employ for vehicle brake balance testing is more a

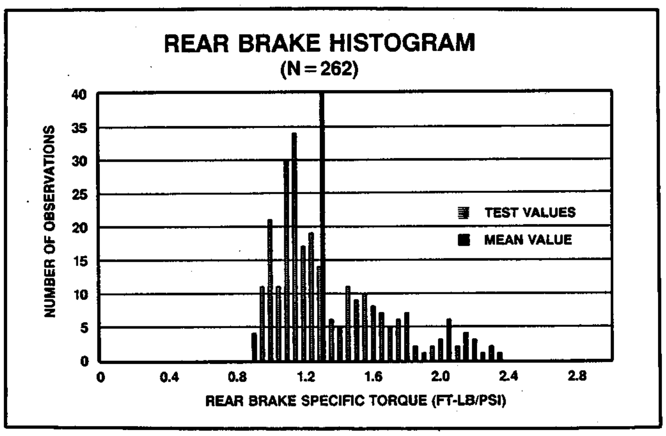

Figure 6

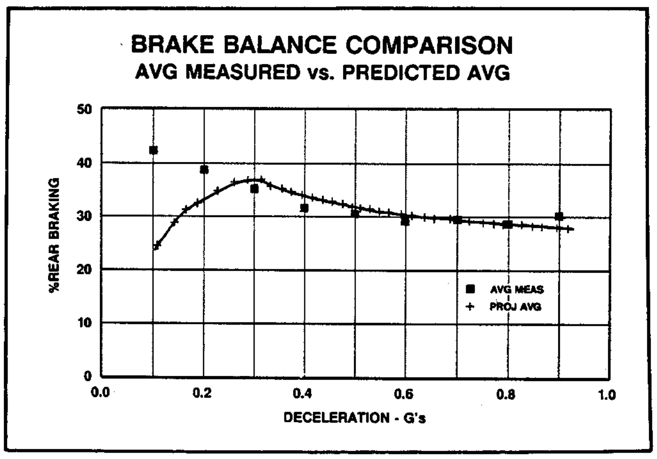

Figure 7

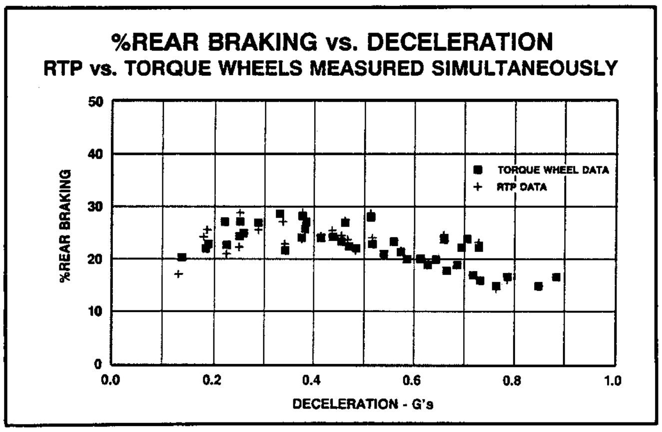

Figure 8

matter of purpose than technical accuracy. The ability of the RTP to evaluate a number of vehicles without modification makes it highly desirable for conducting surveys or monitoring test fleets for brake balance. The price of this convenience is a limited "snapshot" of the continuum of brake balance for the particular vehicle being evaluated and some sensitivity to driver skill in conducting the test. The objective data collected by the RTP is easily treated statistically and may be compiled into large database structures for archival reference.

The use of fully instrumented vehicle tests is best suited to comprehensive evaluations of single vehicle brake systems where a detailed understanding of many brake system operating characteristics is necessary. This test technique is typically applied to extended vehicle development or assessment studies. The wealth of brake system performance assessments possible with this test technique come at the price of an inventory of specialized torque wheels and digital data acquisition and processing facilities. This method can also be used to define brake factors for nominal vehicle adhesion utilization calculations, such as those of Annex 10.

The benefits of brake balance testing by either the RTP or instrumented vehicle techniques include the ability to construct objective test results in large databases for statistical evaluation. The testing conducted on test surfaces offering good adhesion permit measurement of vehicle brake balance over a wide range of decelerations and speeds. The ability to predict wheel lock sequence on a broad range of tire-to-road coefficients by analysis of test results reduces significantly the severe challenge of maintaining a collection of artificial test surfaces and watering systems.

While the capital costs associated with objective testing of vehicle brake systems can be substantial, it pales in comparison with the costs of construction and maintenance of large artificial test surface facilities complete with controlled watering systems. Simplifed RTP and instrumented vehicle packages have been proposed with capital costs on the order of $20,000. Although these simplified test technologies do not provide the complete brake system performance analysis generated by the more extensive systems described in this paper, they do offer the capability of providing objective measurements of vehicle brake balance over a broad range of vehicle decelerations.

Summary

Objective testing of vehicle brake balance can be accomplished through the use of instrumented vehicle technology, the RTP, or through objective tests to determine brake specific torques. The use of "skid checks" or wheel lock sequence tests, particularly on wetted surfaces, is neither statistically reliable nor valid. Exogenous variables may dominate such tests to such a degree that results are not true indicants of vehicle brake balance. The costs of testing by any of the objective technologies described can be small compared to those associated with the construction and maintenance of wetted test surfaces of known and consistent characteristics.

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Antilock System Performance Under Winter Conditions—What Should Be Required?

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Abstract

In Sweden accidents due to loss of stability and steerability caused by braking with locked wheels to a large extent occur on icy roads in the winter. It is therefore essential that antilock systems that are expected to eliminate this type of accidents perform well under these circumstances. In order to establish suitable test methods and realistic performance levels investigations have been made both with passenger cars and commercial vehicles including heavy trailers.

Stability and steerability in terms of staying within lane boundaries with restricted steering wheel corrections and braking performance in terms of braking efficiency related to a reference performance in terms of lateral friction, locked wheel friction, optimum friction (ECE/EEC) or friction according to the ISO TR 8349 method have been considered as primary safety criteria.

These investigations are described and the results summarized. As winter performance test methods the following are discussed and recommended together with suggested minimum performance

- J-turn braking performance test on ice
- split friction test with very low friction on one side
- straight line braking on ice
- Transition from low friction to high friction surface

Introduction

In Sweden the necessity of driving under winter conditions with very low road adhesion can be expected from the middle of October to the end of April or approximately six months of the year. Not only homogeneous low friction but also asymmetric so called split friction and transitions from low to high friction or the opposite are frequently met during the winter period.

Safe braking under these conditions is for obvious reasons a serious problem. Studded tyres have been found to be one effective way of raising the safety level and is widely used on passenger cars in winter time. They are also to some extent used on trucks but hardly on heavy trailers. Even with studs the friction level can be below 0.2 and the risk of wheel locking with loss of steering control and stability during braking is considerable during emergency braking.

Antilock braking systems with good performance under these conditions can therefore be expected to give a significant reduction in traffic accidents where braking is involved.

The Swedish Road Safety Office, The Swedish Board for Technical Development and other Swedish organizations interested in traffic safety research have therefore sponsored several projects with the aim to investigate what can be expected from vehicles equipped with antilock systems in terms of stability, steerability and braking performance. The Swedish Road and Traffic Research Institute (VTI) has been active in most of these projects.

This paper summarizes investigations during five winterperiods from 1980-1986 and presents proposals for antilock system test procedures and requirements, the aim of which is to ensure good braking performance under winter conditions.

These tests are to be regarded as proposed complementary winter service requirements to be added to the more general requirements in the ECE/EEC regulations.

Antilock System Tests Under Winter Conditions Carried Out by VTI

Test procedures

The following antilock system test procedures have been studied

- steerability tests on ice comprising
  1) J-turn braking
  2) Braking in a steady state turn
  3) Single lane change braking
- split friction test defined as straight line braking with one side of the vehicle on a high friction surface and the other side on a low friction surface (ice)
- straight line braking test on ice
- transition from low to high friction and the opposite with ice as low friction surface

The steerability and split friction tests evaluate different aspects of steerability and stability and braking efficiency while the two remaining test types primarily evaluates braking efficiency.

The conclusion from these studies is a recommendation to include the following winter service approval tests for antilock systems.

1. Driver controlled J-turn test on ice
2. Split friction test with one very low friction surface
3. Straight line braking test on ice

4. Transition test from very low friction to high friction.

In the following the different test procedures and test results will be presented and discussed.

Test vehicles

Tests were carried out with the following vehicles which are shown in figure 2.1. All heavy duty vehicles were air braked, with S-cam drum brakes.

- 1980 with a two axle heavy duty truck and a two axle drawbar trailer both equipped with two antilock systems, Girling and WABCO. (Vehicle 1 and 5)
- 1981 with the same vehicles (1 and 5) now equipped with three antilock systems, Girling GX, WABCO and Bosch. In addition a three axle heavy duty truck with Bosch antilock system was used. (Vehicle 3)
- 1984 with a three axle heavy duty truck and a two axle drawbar trailer with a prototype system. (Vehicle 4 and 6) and with the six heavy vehicle combinations 7-12 all equipped with WABCO antilock systems.
- 1985 with a three axle heavy duty truck (Vehicle 2) equipped with a new version of the WABCO antilock system and with two passenger cars. (Vehicle 14 and 15)
- 1986 with three passenger cars (Vehicle 16, 17 and 18) and the heavy duty vehicles 2 and 13. Vehicle 13 was equipped with a Bosch system.

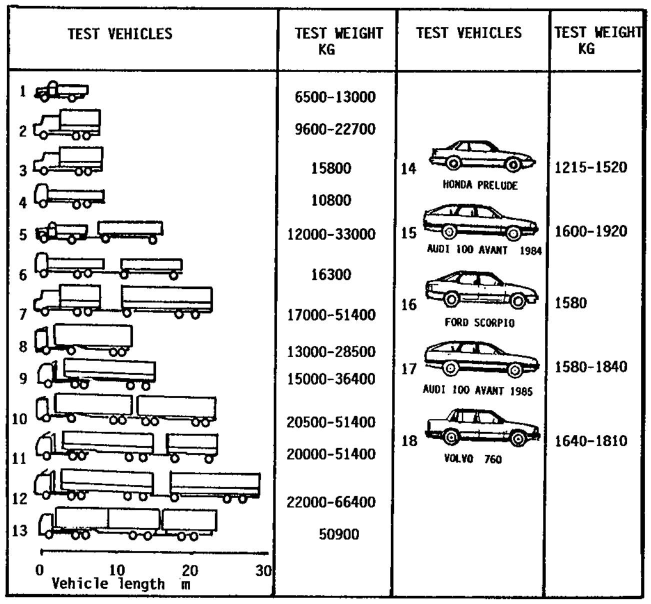

Figure 2.1 Test vehicles

Reasons for Choosing Ice Instead of Other Low Friction Surfaces

The reason for choosing ice as test surface is that it is a representative winter road condition that is durable and relatively easy to produce and maintain. A disadvantage is that it requires temperatures below 0°C. It has unfortunately till now not been possible to find an alternative surface that has all the important characteristics of ice. One of these characteristics is that a wheel operating at high slip reduces the friction of the ice for the following wheels. This means that on ice the front wheels can get higher friction than the rear wheels which has a destabilizing effect. On wetted low friction surfaces for test purposes the effect is normally the opposite as friction increases when the water is wiped away.

Steerability Tests

Justification

Vehicles with antilock systems on steered wheels will if they meet the requirements of a straight braking efficiency test and a split friction test certainly possess some degree of steerability. The split friction test can indeed be regarded as a kind of high friction steerability test. On very low friction surfaces a bad antilock system may, however, give either very poor stability or very poor steerability due to high slip levels and poor slip distribution between front and rear axles. In both cases the expected safety benefits will not be obtained and in the unstable case it might even be more dangerous to use such an antilock system than a normal brake system or vehicles with antilock system only on the rear axle. It is therefore essential to test the steering qualities during emergency braking on low friction in a special test.

This is also true for trailers as the steerability and stability of a vehicle combination depends also on the performance of the trailer. Three basic steering tests have till now been used in connection with antilock system evaluation:

1. J-turn braking.
2. Braking from a steady state turning condition (Braking in a turn).
3. Braking during a single lane change

J-turn braking as an open loop constant step steer input test for passenger cars with antilock systems was proposed by Sweden at the 1974 ESV congress. This was the result of a research program carried out mainly by the Swedish car industries SAAB SCANIA and VOLVO but in which VTI also took active part.

Braking in a turn on a high friction surface has been standardized by ISO for passenger cars with normal braking systems as an open loop test with constant steering input. TÜV Rheinland has tested and proposed this method also for passenger cars with antilock systems. In order to assess the steering reserve an additional test with increased steering angle is used in this case.

Single lane change braking is part of the SAE recommended practice J46 for testing of antilock systems (slip control systems).

Tests carried out by VTI

In 1980 VTI carried out tests on ice according to the three basic methods. The braking in a turn test was, however, made with driver control. For this test type both 100 m and 300 m radius were used. The tests were made with an air braked heavy truck-drawbar trailer combination equipped with two well-known European antilock systems.

The conclusion from these tests was that all three procedures can be considered meaningful in order to assess steering characteristics during braking on ice. The open loop type of testing turned out to be more severe than the closed loop type with driver control. The truck had a tendency to oversteer especially with one of the antilock systems which with fixed steering angle resulted in rear axle spin out. This could be avoided by driver control without excessive efforts. The tests at 300 m radius gave similar results to those at 100 m radius in terms of stability problems and deceleration level. This indicates that it could be sufficient to use the less expensive 100 m radius.

In 1981 it was considered desirable to choose one of the methods for further studies. The J-turn test was chosen being a compromise between the braking in a turn and single lane change in terms of driver skill, lateral space requirement and possibility to choose between open loop and closed loop versions of the test. It was decided to use 100 m radius in the closed loop version. During these tests the ice was roughened by a specially designed multi wheel trailer with studded passenger car tyres running at 10° slip angle (figure 4.1) in order to increase the friction level and reduce friction variations due to polishing and other environmental effects. The tests were carried out with the same truck and full trailer now equipped with three different antilock systems two of which had not been tested in 1980. An additional truck equipped with only one of the systems was also tested.

Further tests have been carried out with the closed loop J-turn test, in 1984 with one truck-drawbar trailer combination, two tractor semitrailer combinations and three tractor-semitrailer plus drawbar trailer (double) combinations and in 1986 with a single truck and a tractor-semitrailer-centre axle trailer (double) combination. In 1985 and 1986 a total of five passenger cars have also been tested.

Proposal for a J-turn braking test on ice

Based on the practical experiences from the VTI investigations a driver controlled J-turn braking test on ice with the following specification is proposed for combined evaluations of stability, steerability and braking performance of vehicles with antilock system on a steering axle. Other test surfaces can be used for a more general test for less severe conditions.

Test track surface. The test track surface shall be ice with a friction level that permits a maximum cornering speed without braking between 40 and 60 km/h. Especially for tests with heavy vehicles it is recommended that the ice is roughened by ordinary passenger car tyre studs, preferrably by using the special multiwheel trailer with side slipping wheels according to figure 4.1. In order to get a suitable friction level, an air and ice surface temperature well below 0°C is normally also required. The track surface must not deviate more than 1° from the horizontal measured over the track width in radial direction and over 10 m along the track.

Test track configuration. The test track which is shown in figure 4.2 shall consist of a straight entrance lane, 30 m long and 0.5 m wider than the vehicle, followed by a 100 m radius curved lane, 100 m long and 1.5 m wider than the vehicle.

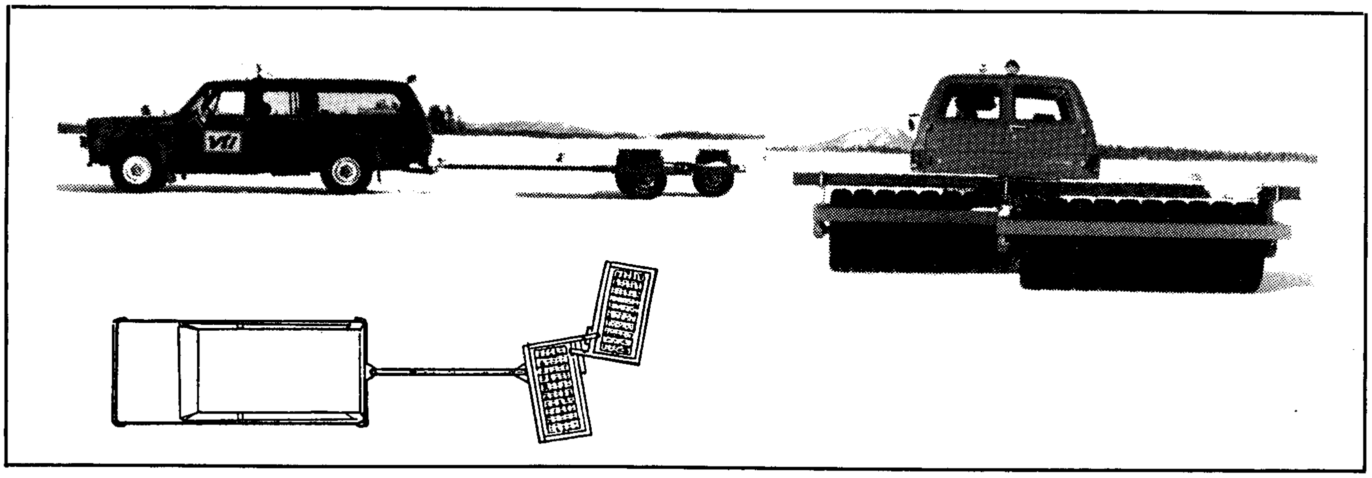

Figure 4.1. Multiwheel trailer with studded passenger car tyres for ice conditioning treatment

The lanes may be marked by cones every 10 m. It must be possible to drive at least 50 m in the circular lane at maximum lateral acceleration.

Instrumentation. The vehicle must be equipped with instruments for measuring and recording

- vehicle speed
- braking distance
- steering wheel angle

Test procedure for motor vehicles

1. Determine the maximum constant speed (V_M) at which it is possible to drive through the J-turn without excessive steering corrections.
2. Make J-turn braking tests with the antilock system in operation and full brake pedal application when the front axle is within ± 1.5 m from the end of the entrance lane. Determine the maximum initial speed Vo from which the test can be made without leaving the lane, and without excessive steering corrections. It is recommended to do this by changing the initial speed in steps of approximately 2.5 km/h starting from 75 percent of V_M.
3. Check that V_M has not changed. If this is the case use the new value of V_M as reference.
4. The vehicle is considered to be inside the lane as long as no part of the tyre tread has crossed the boundary lines.
5. Steering corrections are considered excessive if they exceed $\pm 180°$ from the straight line position.
6. V_M and V_o shall be the mean value from three tests.
7. Stability and steerability performance E_S is expressed by the ratio $(V_O/V_M)^2$.
8. Braking efficiency is expressed by $E_{BY} = a_{ALS}/a_{yM}$ Acceptable alternatives are:

$E_{BL} = a_{ALS}/a_L$

$E_{BE} = a_{ALS}/a_E$

where

a_{yM} = Maximum lateral acceleration

a_{ALS} = Mean deceleration with antilock system in operation

a_L = Mean deceleration with locked wheels from 40 km/h on the same type of surface

a_E = Maximum constant deceleration without wheel locking from 0.75 V_o km/h, calculated from front axle braking deceleration (ECE/EEC method)

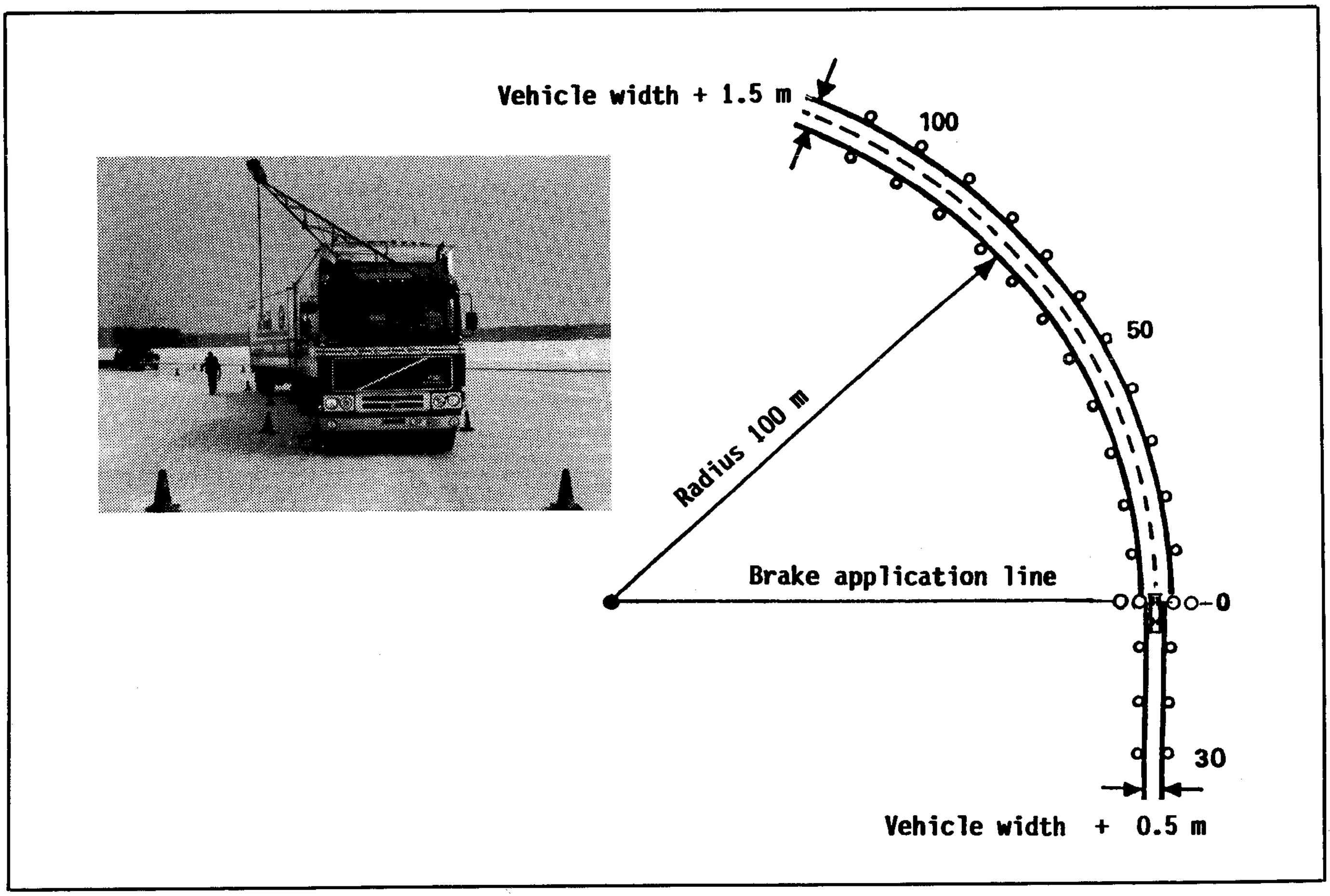

Figure 4.2. J-turn antilock braking test with driver control. Test track configuration

9. Proposed minimum performance requirements:
   Stability/steerability index $E_S \geq 0.64$
   Braking efficiency index $E_{BY} \geq 0.50$
   Braking efficiency index $E_{BL} \geq 0.90$
   Braking efficiency index $E_{BE} \geq 0.75$

Test procedure for trailers. The proposal concerning trailer testing is the following:

A trailer equipped with an antilock system shall be tested in combination with a towing vehicle also equipped with an antilock system. Both vehicles shall have the same type of tyre equipment. Both vehicles shall be tested unladen or laden. It is also recommended that jackknife preventing devices are used at all articulation points.

The test is performed in the same manner as for the single motor vehicle. In addition a test is made where only the trailer is braking. From this test the trailer deceleration is calculated. The rolling resistance of the truck should be taken into account with the values 0.015 for the driven and 0.01 for non driven wheels.

Test results

Test results from the closed loop J-turn test are presented in figure 4.3, 4.4 and 4.5. Figure 4.3 illustrates the relationship between the steerability/stability index E_S and the braking efficiency E_{BY} for passenger cars and for heavy duty vehicles.

Figure 4.4 presents the relationship between E_S and E_{BL} also with separate results for passenger cars and heavy duty vehicles.

Figure 4.5 shows the relationship between E_S and E_{BE}. The results are from passenger car tests only as front axle deceleration in the curve has not been made with heavy duty vehicles.

The diagrams show no correlation between the steerability/stability factor and the braking efficiency. Results from tests with increasing initial speed have depending on tyre equipment given both increasing and decreasing braking efficiency.

From the diagrams can be seen that if all results were to be accepted the following limits should be set:
$E_S \geq 0.5$, $E_{BY} \geq 0.50$, $E_{BL} \geq 0.9$, $E_{BE} \geq 0.75$.

The proposal to set the minimum requirement on E_S to 0.64 is based the opinion and that this value is more representative for state of the art. The vehicles that did not reach this value should be improved.

Split Friction Test With Very Low Friction On One Side

Justification

The split friction condition, where the wheels of the vehicle is on high friction on one side and very low friction on the other, is a typical winter time road surface condition in Sweden and drivers are trained to make use of one sided high friction when it is available. Vehicles with standard braking system are by design capable of utilizing the high friction. This is not necessarily the case when an antilock system is fitted. In fact there are systems on the market which adapt the braking force of both wheels on an axle to that demanded by the wheel with the lowest coefficient of friction. These systems have the advantage of not giving yaw stability problems and in principle being less costly. The disadvantage in terms of longer braking distance (S) becomes more and more pronounced when the low friction coefficient is lowered since approximately $S = A/K_2$ where A is a constant if the initial speed and braking efficiency is constant. K_2 = Low friction coefficient.

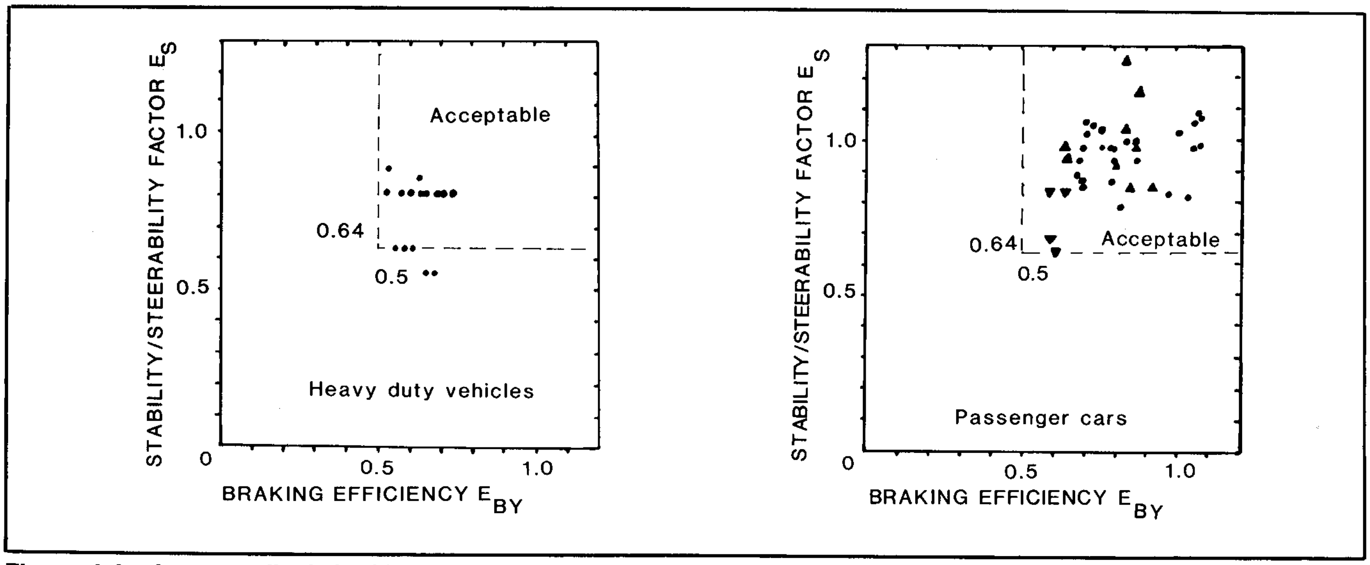

Figure 4.3. J-turn antilock braking test with driver control. Test results showing stability/steerability factor and braking efficiency E_{BY}

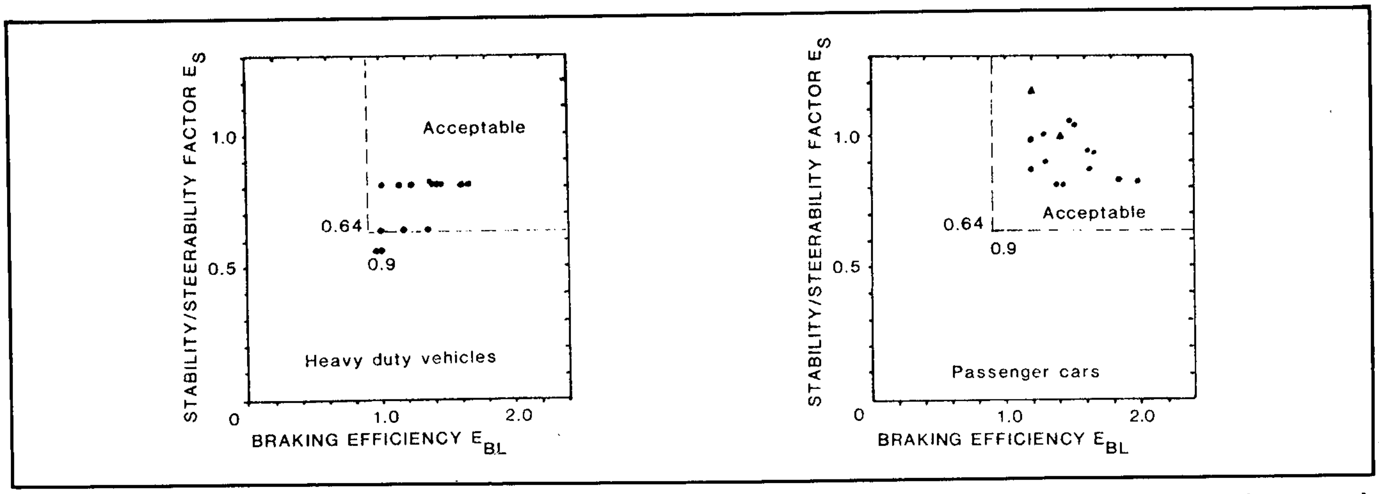

Figure 4.4. J-turn antilock braking test with driver control. Test results showing stability/steerability factor and braking efficiency E_{BL}

In figure 5.1 the braking distance from 50 km/h is shown as a function of the low friction coefficient when 5 m/s^2 can be achieved on the right friction surface and the braking efficiency is 100%. It can be seen that at a friction coefficient of 0.1 the braking distance is 95 m for a system adapting to the low coefficient (select low system) compared to 32 m for a system with individual wheel control. Antilock systems with these characteristics are not desirable on roads where split friction conditions with very low friction can be expected. In order to check split friction braking performance, the low friction coefficient should be as low as possible.

Tests carried out by VTI

VTI has performed tests on split friction surfaces with ice as low friction surface having peak and locked wheel friction coefficients 0.1 with standard tyres and sand bonded to the ice with water as high friction surface giving a friction coefficient of about 0.6 (figure 5.2). The test speed was 50 km/h. As shown in the figure, antilock braking tests were performed also on the low and high friction surfaces for reference purpose. Part of the test program also included measurement of the friction coefficients K_1 and K_2 according to ECE/ECE regulations. This was done with front axle braking without wheel locking.

Passenger cars, heavy trucks and truck trailer combinations with commercially available systems and also one prototype system (see figure 2.1) have been tested. The systems represent different control strategies from "select low" that adapts to the low friction to individual wheel control.

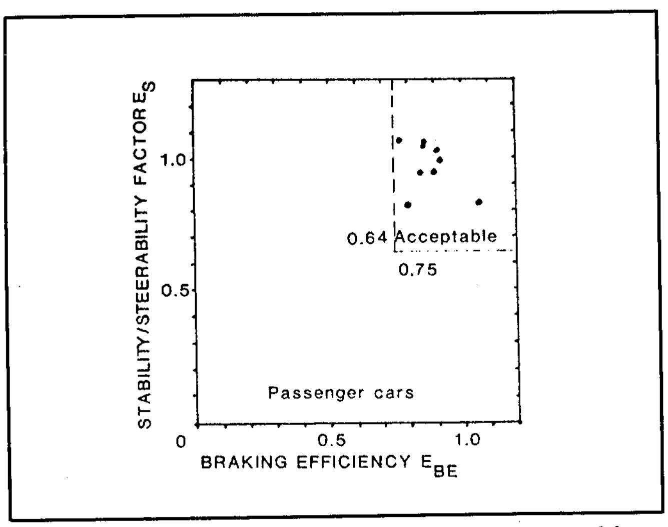

Figure 4.5. J-turn antilock braking test with driver control. Test results showing stability/steerability factor and braking efficiency E_{BE}

Results

In figure 5.3 the braking ratio Z_3 obtained in the split friction test is compared to the optimum braking ratio defined as $(Z_1 + Z_2)/2$. The braking ratios Z_1 and Z_2 are those obtained by antilock braking tests on the high and low friction surfaces with the vehicle in question. Braking ratio is defined as the mean deceleration divided by gravity acceleration (9.81 m/s^2).

It can be seen that the select low system has a braking efficiency of only 27 percent of the optimum braking ratio compared to values between 75 and 90 percent for systems with a higher degree of individual wheel control. In figure 5.4 the braking ratio Z_3 is compared to a minimum required braking ratio $(4Z_2 + Z_1)/5$ proposed by E. Petersen from WABCO. The tested vehicles with select low antilock systems did not meet this requirement. In figure 5.5 test results are compared with the ECE/EEC requirement $Z_3 \geq 0.75 (4K_2 + K_1)/5$. The ECE/EEC requirement was easier to meet than that from WABCO.

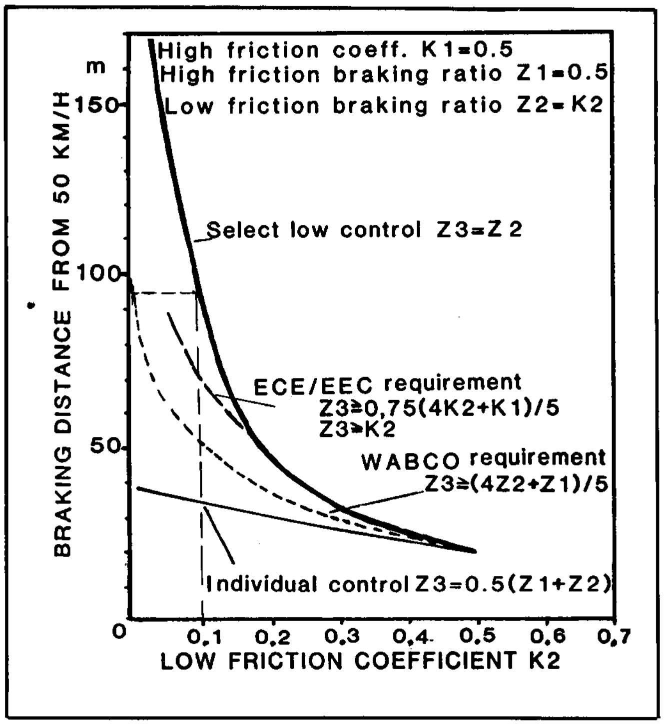

Figure 5.1. Split friction braking efficiency

Discussion

The formula used in figure 5.4 uses the actually achieved braking ratios on the two surfaces and in most cases allow a select low system to be used on the

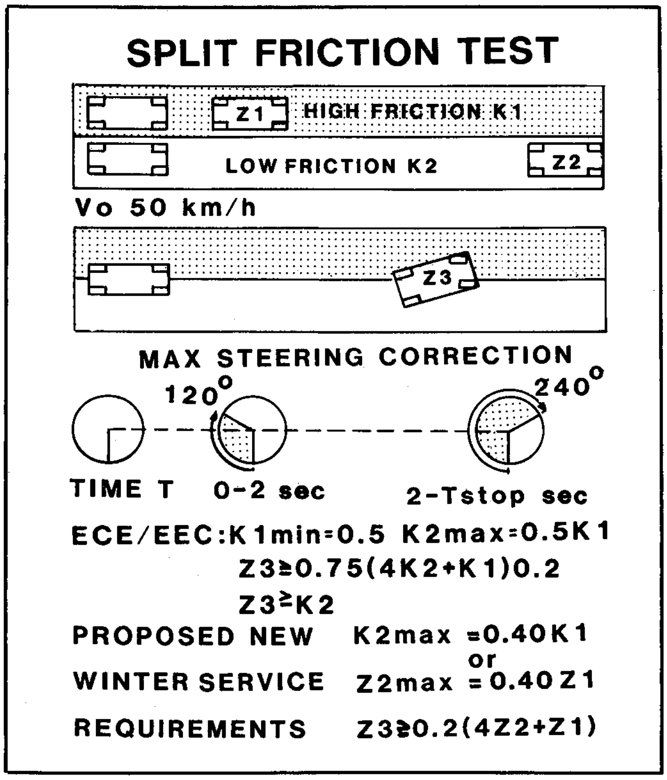

Figure 5.2. Split friction test procedure

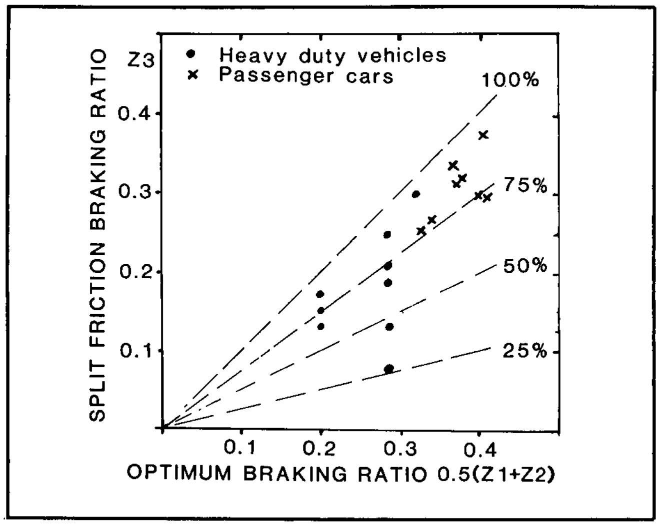

Figure 5.3. Split friction test results. Comparison with optimum performance

front axle of a two axle vehicle. It has been argued that systems with high efficiency on homogeneous surfaces are disfavoured as the requirement in absolute deceleration is higher than for a less efficient system. If 75 percent of the friction coefficients K_1 and K_2 are used instead of Z_1 and Z_2 this problem is eliminated but vehicles with an efficient select low system on all axles can pass the test if the coefficient of friction exceeds 0.2 for the low friction surface (see figure 5.1). If a sufficiently low K_2 value is required this risk is eliminated. It is however recommended to use Z_1 and Z_2 as the test procedure than is simpler.

The ECE/EEC antilock regulations do not require a split friction test for trailers. In Sweden a vehicle combination is allowed to weigh about 52000 kg of which 36000 kg may be the trailer weight. If a select low system is used on all trailer axles and on the front

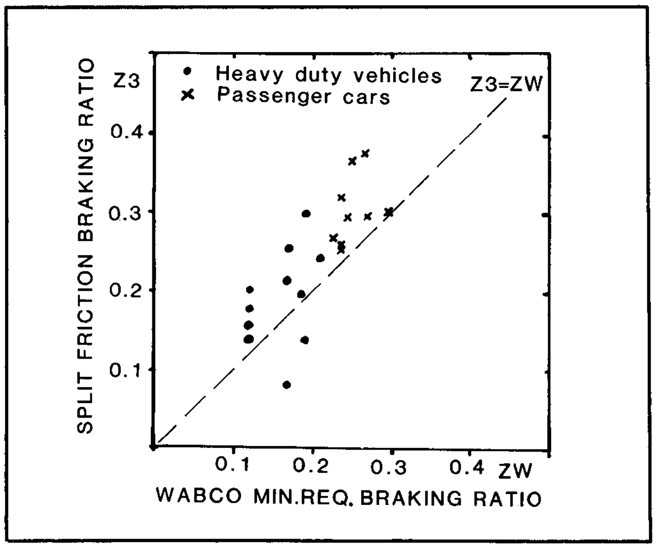

Figure 5.4. Split friction test results. Comparison with WABCO proposal

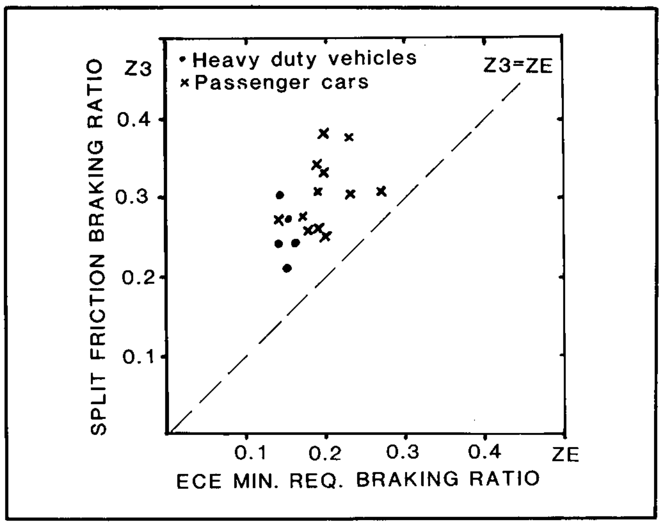

Figure 5.5. Split friction test results. Comparison with ECE/EEC requirements

axle of the truck, about 90 percent of the combination mass will be braked with low friction brake forces compared with only 66 percent if the trailer axles are individually controlled. If $Z_2 = 0.1$ and $Z_1 = 0.5$, Z_3 in the first case will be 0.140 and in the second 0.236. This difference is considered large enough to justify the same split friction requirements for trailers and motor vehicles.

Split friction braking performance of vehicle combinations with and without antilock system in operation

In 1984 split friction tests were also carried out in order to compare performance with and without antilock system. Four unladen heavy vehicle combinations (vehicles 6, 7, 10 and 11 in figure 2.1) were used. All were equipped with load sensing valves and WABCO antilock systems. Three drivers took part in the test but each combination was tested by only one driver.

The result of these tests was that the braking efficiency in most cases was 10-20 percent higher with the normal braking system than with the antilock system in operation. The efficiency of the antilock systems was about 80 percent of $(Z_1 + Z_2)/2$. Both with and without antilock system it was possible to keep the vehicle within a 3.5 m lane. The steering and braking task was, however, more difficult without antilock system. The maximum steering angles were about 90 degrees with and 135degree without antilock system.

Proposal for a winter condition split friction test

This test is specified in line with the ECE/EEC antilock system regulations except for

- Lower low friction
- Measurement of antilock braking ratios instead of friction coefficients
- Minimum performance requirement according to WABCO
- Trailer test requirement

Test track. The test track shall have two surfaces, one with high friction on one side of the center line and one with low friction on the other side. The width of each surface must be large enough to allow braking tests with the vehicle.

The low friction surface shall have a peak coefficient of friction of 0.1 ± 0.05 and a locked wheel friction of $0.1 - 0.05$ in the speed range 50 – 15 km/h. This requirement is met by new (less than 24 h) smooth ice even at -15° C according to VTI experience if standard tyres are used on the vehicle. This condition is considered to be met if the braking ratio with the antilock system in operation is between 0.15 and 0.04. The high friction surface shall have a peak coefficient of friction of at least 0.5. This condition is considered to be met if the braking ratio with the antilock system in operation is at least 0.375 (75% of 0.5). Sand bonded to ice with water can meet this requirement. A track length of 100 m has turned out to be sufficient.

Test procedure

1. Measure the mean braking ratio with antilock braking for each of the two surfaces in the speed range 40-20 km/h when the vehicle is braked from 50 km/h.
2. Measure the mean braking ratio Z_3 in the speed range 40-20 km/h when the vehicle is braked from 50 km/h with its left and right wheels on each side of the boundary between the high and the low friction surface.

Performance requirements

1. No part of the (outer) tyre treads must cross the boundary line during the stop.
2. Steering corrections must not exceed 120 degree during the first two seconds from brake application and not exceed 240 degree in all.
3. The braking ratio Z_3 must not be less than:

Z_1 = Braking ratio on high friction surface
Z_2 = Braking ratio on low friction surface
Z_3 = Braking ratio on split friction

Trailer test. The trailer should be tested together with a representative towing vehicle equipped with an approved antilock system. The combination must meet the requirements for a single vehicle. The braking efficiency of the trailer should also be tested by means of separate trailer braking according to the procedure in 5.6.2. When Z_1, Z_2 and Z_3 are calculated the rolling resistance of the truck should be taken into account with the values 0.015 for driven and 0.01 for non driven wheels.

Straight Line Braking on Ice

Justification

Straight line braking on a homogeneous surface is the classic and basic way of testing the braking performance of vehicles. The ECE/EEC antilock braking regulations prescribe straight line low friction tests. The friction level is however allowed to be as high as 0.4. A test on ice is therefore regarded as a useful winter service approval requirement.

Tests carried out by VTI

Straight line antilock braking tests on ice have been made in comparison with

- locked wheel braking
- best driver control braking
- peak friction measured with single axle braking according to ECE/EEC antilock regulation
- Standard reference tyre friction coefficient measured with the Swedish constant slip (13-15 percent) friction test vehicles BV 11 and BV 12 (see figure 6.1 and 6.2) measured at 40 and 20 km/h. BV 11 had a 4.00-8 tyre and BV 12 a 5.60 - 15 PIARC "Europe" tyre both with rib tread, in accordance with ISO TR 8349 on friction measurement.

Tests were carried out with the following vehicles which are shown in figure 2.1.

- 1980 with a two axle heavy truck and a two axle drawbar trailer both equipped with two antilock systems, Girling and WABCO. (Vehicle 1 and 5)
- 1981 with the same vehicles (1 and 5) now equipped with three antilock systems, Girling GX, WABCO and Bosch. In addition a three axle truck with Bosch antilock system was used. (Vehicle 3)
- 1984 with a three axle truck and a two axle drawbar trailer with a prototype system. (Vehicle 4 and 6)
- 1985 with a three axle truck (Vehicle 2) equipped with a new version of the WABCO antilock system and with two passenger cars, Audi and Honda.
- 1986 with three passenger cars, Audi, Ford Scorpio and Volvo.

Tests have been made from initial speeds ranging from 70 to 35 km/h for trucks and trailers and, for passenger cars from 110 to 50 km/h.

Most of the tests have been done in the temperature range -5°C to -20°C but tests have also been made near 0°C and down to -30°C. Except in 1980 the tests have been made on ice roughened by the special multiwheel trailer with studded passenger car tyres shown in figure 4.1. The treatment results in a somewhat higher and more uniform friction and reduces polishing effects which tend to lower the friction.

Results

The results are summarized in figure 6.3, 6.4, 6.5, 6.6 and 6.7.

From the figures can be seen that as a rule the braking efficiency with antilock systems is higher than with locked wheels and higher than best driver performance. In tests with laden and unladen vehicles the unladen vehicles tend to get higher deceleration but not necessarily higher braking efficiency based on peak value.

The 75 percent efficiency required by ECE/EEC regulations is not always met on ice by antilock systems for heavy vehicles. For the tested passenger cars with and without studs the efficiency is close to 100 percent.

Longitudinal friction coefficients obtained with the reference tyres on friction test vehicles BV 11 and BV 12 according to the constant slip method gave the same values as the peak friction coefficients obtained by single axle braking according to ECE Regulation 13 both for a truck and a passenger car with standard tyres (0.17).

Discussion

According to Annex 10 in ECE regulation 13 it is for normal brakes allowed to have a braking efficiency of 50 percent at a friction coefficient of 0.2. It could therefore be debated if the efficiency requirements on antilock systems on ice should be as high as 75 percent of the peak friction coefficient. The test results indicate that 90 percent of the locked wheel friction could be a more suitable requirement. A locked wheel test is also the simplest and least expensive alternative. In order to avoid stability problems the locked wheel braking tests on ice are recommended to be done from an initial speed of 40 km/h.

Proposal for a straight ahead braking test on ice

Based on the field test experience and theoretical considerations the following test is proposed.

Test surface: The test surface should be ice with a locked wheel friction of 0.1 $\pm$ 0,05 or a peak friction of 0,20 $\pm$ 0.05 measured with the test vehicle itself or the friction test vehicle BV11 or equivalent equipment. Roughness of the ice by means of the special multiwheel trailer according to figure 4.1 is recommended. The air and ice surface temperature should be below zero°C, preferrably between -5 and -15°C.

Test speed: The initial speed shall be 40 km/h for vehicles with tyres without studs and 50 km/h for tyres with studs (additional test for passenger cars).

Braking tests: Make locked wheel and antilock braking stops with a pedal force that on high friction would give at least 5 m/s^2. The mean value of the results from at least three tests of each type should be used for the efficiency calculation. For each test the mean deceleration is calculated by the formula $a = 5.56/T$ m/s^2 where T is the time from V = 35 km/h to V = 15 km/h.

Minimum requirement on braking efficiency:

$a_{ALS}/a_L \geq 0.9$ or $(Z_{ALS}/Z_{LOCK}) \geq 0.9$

a_{ALS} = Mean deceleration with the antilock system operating. $Z_{ALS} = a_{ALS}/9.81$

a_L = Mean deceleration with all wheels locked. $Z_{LOCK} = a_L/9.81$

Trailer tests: Trailer tests are made by braking only the trailer and correcting for the rolling resistance of the towing vehicle. The rolling resistance coefficient for nondriven wheels may be assumed to be 0.01 and for driven wheels 0.015. A better alternative is to make a coast down test with the towing vehicle alone.

Alternative test method: Tests may also be performed according to the procedure prescribed by ECE/EEC regulations but on the same ice surface. It is considered as more difficult to carry out and meet the requirements of this test.

Transition Test From Low to High Friction

Justification

When a vehicle with normal brakes is braked on very low friction and suddenly encounters a transition to a high friction surface the braking torque applied by the driver is immediately fully utilized up to the limit of adhesion for each axle as they pass on the new surface.

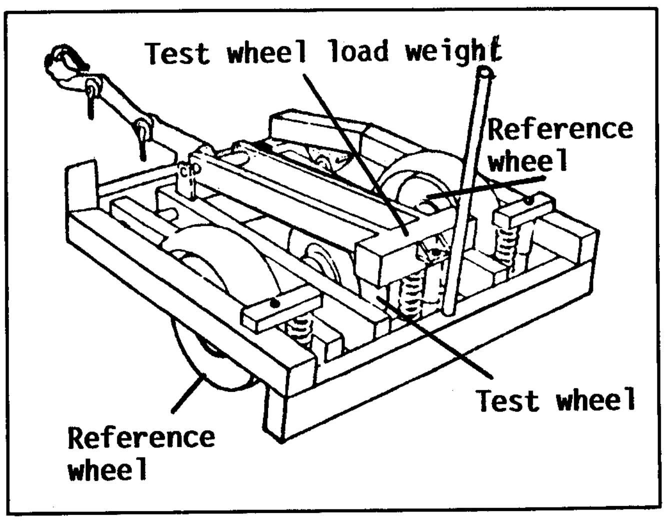

Figure 6. Friction test trailer BV11

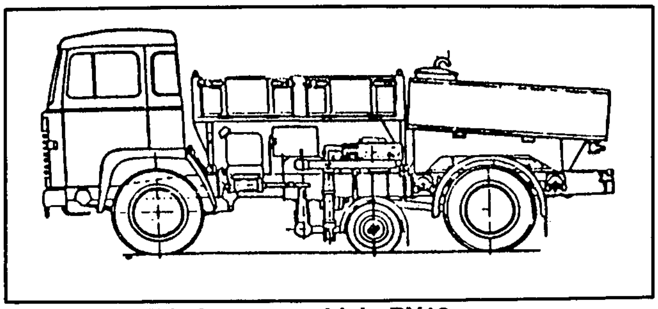

Figure 6.2. Friction test vehicle BV12

In the same situation but braking a vehicle with an antilock system fully adapted to the low friction there is a risk that the pressure recovery might be very slow and result in an unacceptably long braking distance compared with a normal braking system. The new ECE/EEC antilock braking regulations therefore demands a test in this respect. The requirement on the high/low friction ratio is however only 2:1 which is low for winter service conditions.

Tests carried out by VTI

Tests with one heavy duty truck antilock system on ice with very low friction resulted in pressure drops to near zero with recovery rates of not more than 3 bar/sec that could not be influenced by a sudden transition to high friction. This corresponds to about 1.5 seconds to reach 4.5 m/s^2 deceleration. Transition tests with other systems indicate that 0.7 seconds is a reasonable target from a technical point of view.

Discussion

If the vehicle deceleration is measured the wheelbase has to be taken into account. At 50 km/h an additional time delay of 0.7 seconds will cover all practical cases. A total deceleration transition time to 4.5 m/s^2 of 1.5 seconds for heavy duty vehicles and 1.0 second for passenger cars has been considered to be a reasonable requirement for winter service.

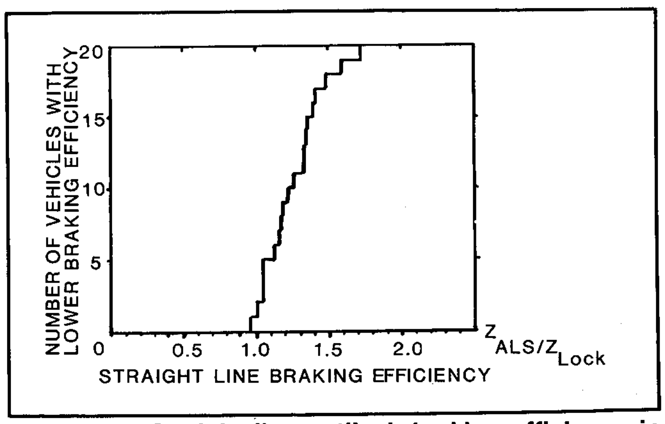

Figure 6.3. Straight line antilock braking efficiency in relation to locked wheel braking

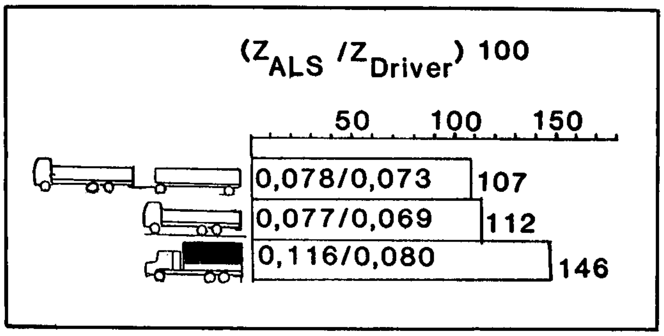

Figure 6.4. **Straight line antilock braking test results. Comparison with test driver performance**

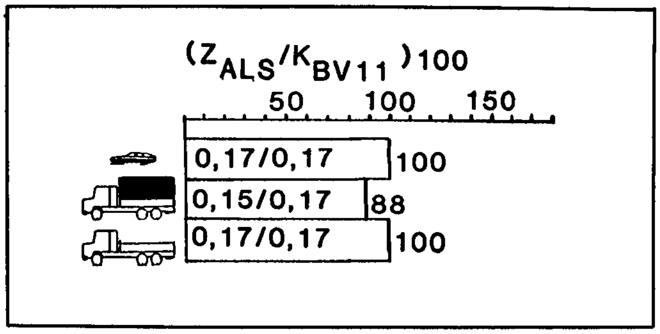

Figure 6.6. **Straight line antilock braking test results with standard tyres. Comparison with friction test trailer BV11**

Proposed winter service test

The test shall be made with full brake application at a pressure that corresponds to a deceleration of at least 5 m/s^2 starting on a low friction surface which must not give the vehicle a higher deceleration than 1.5 m/s^2 with the antilock system in operation. The vehicle speed at the transition to high friction must not be less than 50 km/h. The high friction surface must allow an antilock braking deceleration of at least 4.5 m/s^2. This deceleration must be reached within 1.5 seconds for heavy duty vehicles and within 1.0 seconds for passenger cars. This time is measured from the front axle transition time.

Hybrid Laboratory Testing—A Future Type Approval Procedure?

The practical difficulties are considerable from both technical and economical aspects in obtaining test tracks that give the desired friction characteristics and are large enough for safe high speed and cornering tests. In fact they are so large that the ECE/EEC regulations regard peak friction coefficients up to 0.4 at 40 to 50 km/h as low and do not specify speed or slip characteristics of the tyre/road adhesion. Furthermore the problems connected with brake lining characteristics must not be forgotten.

For antilock systems with electric wheel speed signals these problems can be eliminated by real time computer simulation of the tyre/road characteristics, brake torque characteristics and vehicle motion dynamics including wheel speed sensor signals. The real vehicle that is to be tested is connected to the computer through an interface so that the simulated wheel speed signals are received by its antilock system controller and the wheel brake cylinder pressures measured by sensors on each wheel are fed back to the computer. During the test the vehicle is stationary in the laboratory with the engine running. The test engineer has only to apply the brakes after starting the computer program.

This technique has been used by VTI with promising results.

At present it is possible to simulate:

- Straight braking on a homogeneous surface
- Straight braking on a split friction surface with steering corrections based on yaw motion
- Braking during steady state cornering with constant steer input
- J-turn braking with constant steer input applied at the same time as the brakes

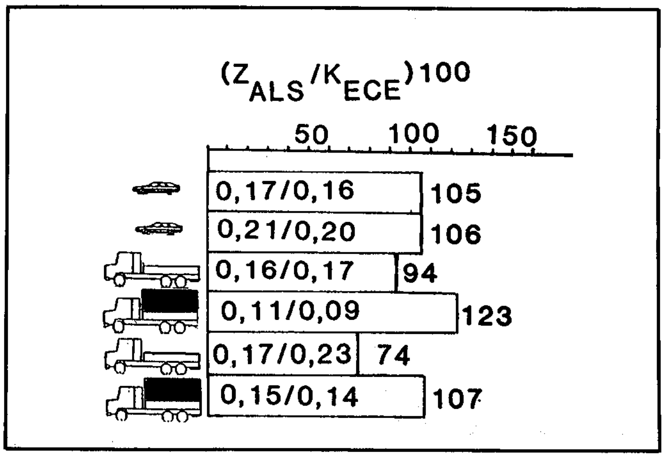

Figure 6.5. **Straight line antilock braking test results. Comparison with ECE/EEC friction coefficient**

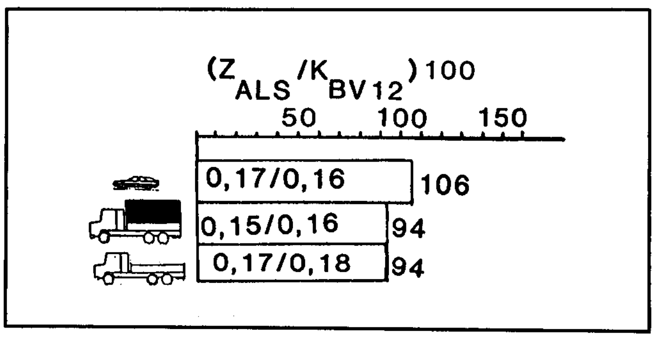

Figure 6.7. **Straight line antilock braking test results with standard tyres. Comparison with friction test vehicle BV12**

- Braking on a surface with changing friction can also be simulated as the computer programme contains two tyre models for each wheel

Validation simulations have been made with a two axle truck with an unladen weight of 6500 kg and a laden weight of 13000 kg. The vehicle was equipped with three different types of antilock systems. This vehicle was also used in the already mentioned tests on real ice tracks split friction tracks as well as on high friction tracks.

The following tests were used for the validation:

- Straight braking on homogeneous ice. Initial speed 10 and 20 m/s
- Straight braking on a split friction surface. Initial speed 10 and 20 m/s
- J-turn braking on ice with constant steering input corresponding to 100 m radius applied at the same time as the brakes. Initial speed 11 m/s
- Straight braking on a high friction surface with the peak friction coefficient 0.6. Initial speed 30 m/s

In all the tests the ranking order in performance was the same in simulation and real test. The general characteristics in terms of deceleration, lateral acceleration and yaw behaviour over time were also quite well reproduced. This also applies to wheel speeds and brake pressures. Tests were made both with identical tyre data on front and rear wheels and with somewhat reduced friction on the rear wheels. The best results were obtained in the latter case. This is in line with the fact that ice friction is reduced by the polishing effect of slipping tyres. In this case the front tyres polish the ice with the rear tyres. It is not believed that this method of testing can replace real world tests but it looks promising as a future complement for evaluating antilock system performance under conditions that are too difficult, expensive or dangerous to require in real type approval tests.

Advantages of an Anti-Wheel Lock System (ABS) for the Average Driver in Difficult Driving Situations*

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Abstract

The objective of this investigation was to establish the extent by which the average driver is able to exploit the increased steering and braking controllability given by an ABS braking system. To this end 77 average drivers, selected from various age groups, carried out 5 controlled driving tests simulating driving manoeuvres with a high accident risk element. The manoeuvres being carried out with two identical vehicles, one with standard conventional brakes, and the other with a Lucas Girling SCS system(1) installed. The following driving tests were programmed

- Avoidance manoeuvre (high adhesion)
- Straight line braking (high adhesion)
- Braking in a curve (high adhesion)
- Braking in a curve with diminishing radius (low adhesion)
- Split mue (μ) braking (high/low adhesion)

The SCS system proved to be an advantage to the average driver in all five manoeuvres. The advantage increased as the road adhesion reduced. In total, the average driver experienced problems of either leaving the marked lane or contacting obstacles 2.4 times more frequently with the standard than with the SCS equipped vehicle.

Introduction

It is estimated that in Germany the universal adoption of anti wheel lock systems would result in a 10-15% reduction in accidents involving heavy damage and/or injuries(1). This conclusion is based on an investigation of accidents and their causes listed below.

- Some 20% of road accident fatalities are pedestrians. Investigations show that contact is generally on the extreme front corners of the vehicle. A vehicle equipped with ABS has greater steering controllability under braking and therefore an avoidance manoeuvre would be possible which would prevent a large proportion of this type of accident.
- Locked wheel braking is evident in 30% of accidents. A vehicle equipped with an ABS system can achieve shorter stopping distances, particularly in reduced adhesion con-

*The investigation was carried out on the request of Lucas Girling ditions, and thereby reduce this type of accident.

- A third of all accidents involve one car only. About 50% of these accidents occur on curves, this type of accident would be reduced with ABS equipped vehicles by virtue of the ability to retain steering control under conditions of braking in a curve.
- Varying adhesion between the left hand and right hand wheels leads to the vehicle gyrating under braking. An ABS equipped vehicle is far easier to control under these conditions.

To evaluate the extent by which the average driver is able to exploit the increased steering and braking control given by an ABS braking system, five driving manoeuvres were devised which simulated high accident potential situations. These manoeuvres were carried out by the team of average drivers and consisted of the following(2):

- Avoidance manoeuvre
- Straight line braking
- Braking in a curve
- Braking in a curve with diminishing radius
- Split mue (μ) braking

Test Conditions and Results

77 average drivers were selected by an independent marketing company in the UK. The selected drivers were of varying age and driving experience, and approximately a third were female. All had experience with Ford Escorts, the vehicle selected for the test programme.

The test drivers were divided into 3 age groups (Figure 1), most being in the 26-40 group.

All test drivers undertook the five driving manoeuvres in sequence in two Ford Escort XR 3 i vehicles, the one equipped with standard braking system (std car), and the other with the Lucas Girling "Stop-Control-System"(3) (SCS car). In order to minimise the experience factor, half the test drivers (37) used the SCS car for the first test run, and then repeated the test sequence with the standard car. The remaining group of test drivers carried out the test programme in the reverse order. Both vehicles were instrumented to enable braking and steering inputs to be measured (Figure 2).

During a familiarisation drive in each vehicle, an observer informed the test driver on the working and advantages of the anti wheel lock system. Eventually the test driver was instructed by tape recorder to drive in 5th gear at a speed of 45 mph into the first test manoeuvre (avoidance test) (Figure 3).

On entry into a 3 m wide lane, the vehicle actuated a trip mechanism which released an obstacle, a dummy child, sited 35 m from the entry and hidden behind a wall on the left hand side. This obstacle was projected into the path of the vehicle. With a controlled time delay, the same mechanism triggered a second child dummy obstacle into the path of the vehicle 20 m beyond the first obstacle. The first obstacle could only be avoided by a steering and braking action, and the second obstacle by full braking only. The siting of the obstacles being arranged that, with an entry speed of 45 mph, it was impossible to stop the car before the first obstacle, so that steering action was essential to avoid contact. The road surface was wet with an adhesion of $\mu_G \approx 0.8$ (figure 4).

The result of these comparison tests are shown on Figure 5. The upper part shows the total driving tests with the standard car, and the lower part the same tests with SCS car. The two halves are divided to show the critical and safe driving situation results. The critical situation was defined by either vehicle contact with the obstacles, or with the lane marker cones.

No distinction was drawn between light contact with a marker cone and complete departure from the marked lane.

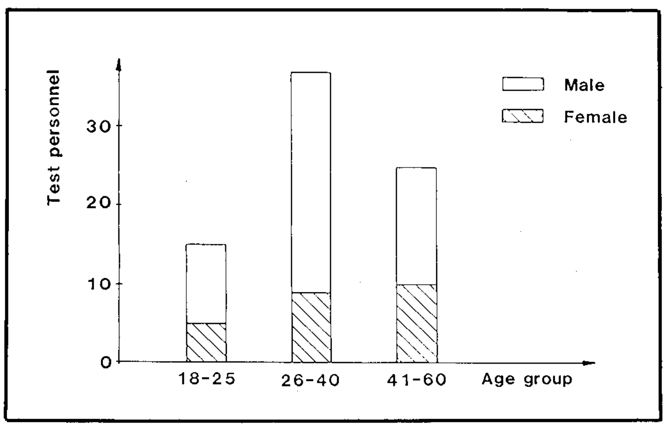

Figure 1. Age groups of the test personnel

Figure 2. Instrumented test vehicles

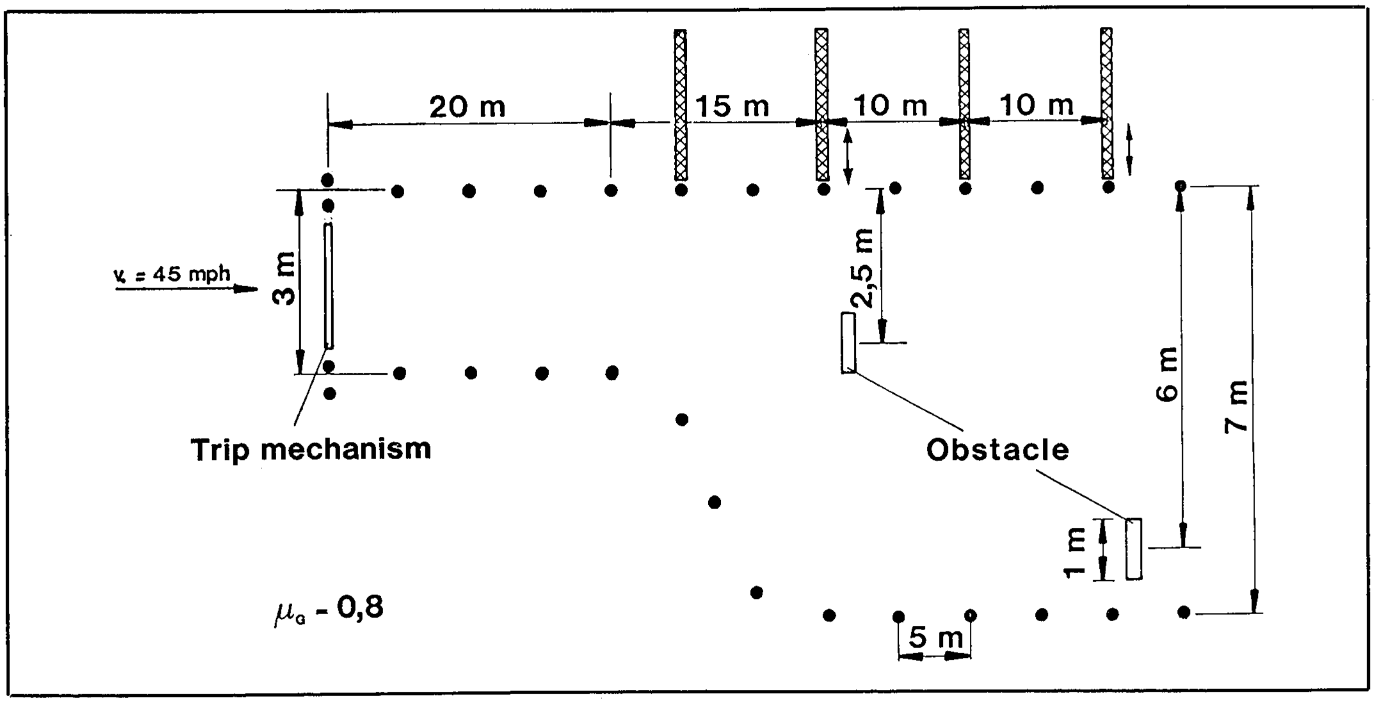

Figure 3. Avoidance manoeuvre (wet roads)

The test drivers recorded a total of 86 critical situation results, 47 with the standard car and 39 with the SCS car. From these results it can be deduced that the driver of the standard car is more prone to a critical situation by a factor 1.2. This low factor does not give a significant statistical advantage to the average driver with the SCS equipped vehicle. That the average driver in this driving manoeuvre had a relatively small advantage from the SCS system was, in part, due to incorrect decision, or too slow reaction to the driving situation, so that the full potential of the SCS system was not always exploited.

The initial reaction time to take appropriate action is critical for the successful completion of this steering/braking manoeuvre. Should the reaction time of the test driver exceed 1.75 s, then the remaining braking distance available is insufficient to avoid a collision with the obstacle if the entry speed of 45 mph is achieved.

The entry speed for the second test (straight line braking) (Figure 6) was fixed at 45 mph. On entry

Figure 4. Avoidance manoeuvre

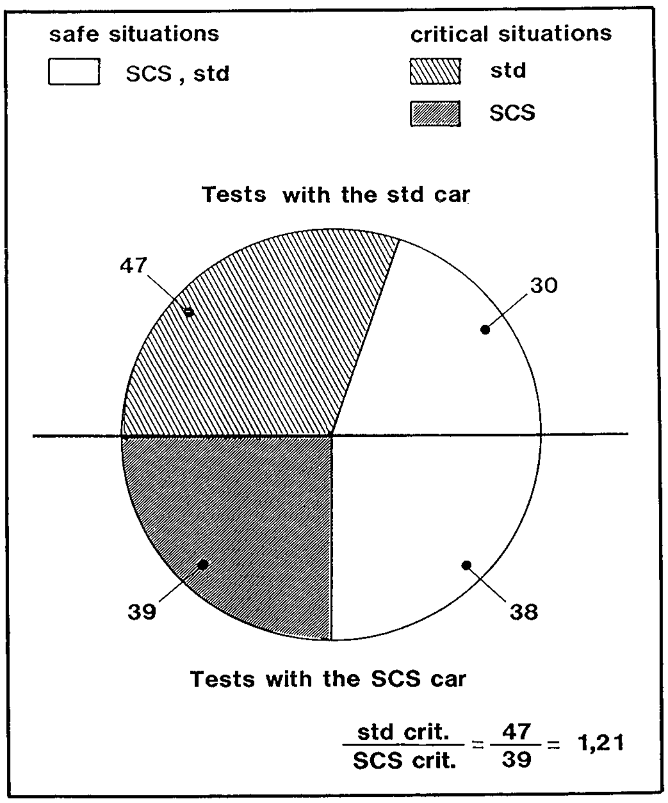

Figure 5. Critical situations in avoidance manoeuvre

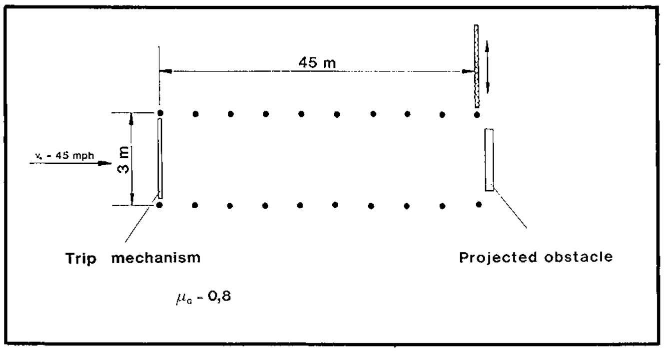

Figure 6. Straight line braking

into a 3 m wide lane, the vehicle triggered a trip mechanism releasing an obstacle situated 45 m beyond the entry point on the left hand side. The obstacle was concealed behind a wall and not visible to the driver until projected into the path of the vehicle.

The road surface was wet with an adhesion of $\mu_G \approx 0.8$ (Figure 7).

The test drivers recorded 15 critical situations in this driving manoeuvre either by contact with the obstacle, or contact with the lane marker cones. 9 critical situations were recorded with the standard car and 6 with the SCS car. The driver of the standard car is more prone to failure by a factor of 1.5 (Figure 8).

In this test the average driver's advantage with the SCS equipped vehicle was also statistically insignificant.

The reason for these results lies in the relatively high demand on the driver. The driving test requires both a quick reaction time and maximum deceleration to avoid a collision with the obstacle.

Only some of the test drivers were able to utilize the available braking capacity of the vehicles.

All the collisions with the obstacle which were recorded in the standard car, occurred in the first run of the test. Four of the test drivers failed due to excessive reaction time giving a reduced stopping distance impossible to achieve without a collision with the obstacle. In the tests with the SCS car, slow reaction was the cause of failure in only two instances.

Figure 7. Straight line braking

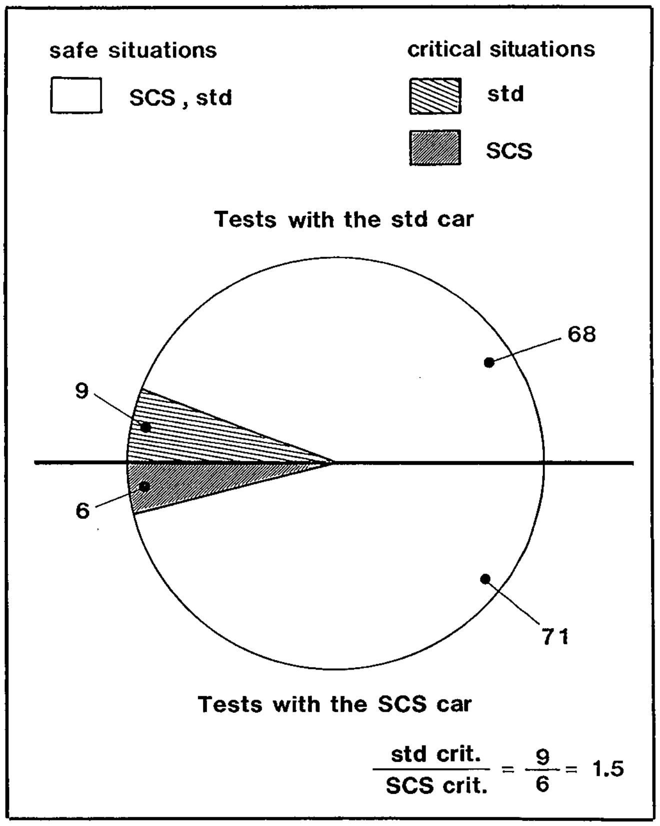

Figure 8. Critical situations in the straight line braking test

The entry into test 3 (braking in a constant radius curve), was controlled by an observer in the test car. The driver was instructed to drive at 45 mph and to stop the car as quickly as possible on entry, keeping the car within the cone marked lane (Figure 9).

The test track was again wet and had an adhesion of $\mu_G \approx 0.8$ (Figure 10).

Of the 30 tests in which the driver left the marked lane, 22 occurred with the standard car, so that the

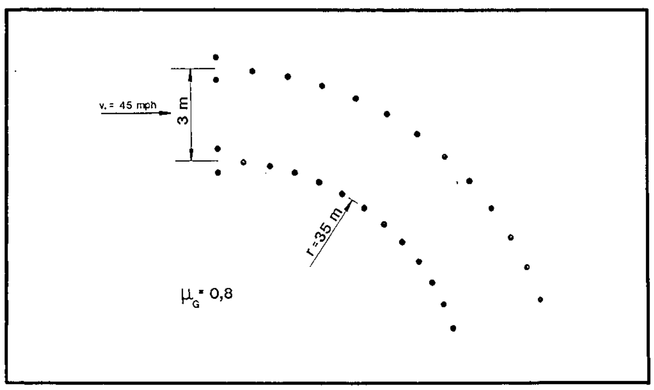

Figure 9. Braking in a curve

Figure 10. Braking in a curve

average driver was more prone to failure in the standard car by a factor of 2.75 (Figure 11).

All the test drivers in the standard car who left the marked out lane, lost control due to locked front wheels.

Whilst the tests 1 to 3 were on a wet surface but with a relatively high adhesion ($\mu_G = 0.8$), the tests 4 and 5 were carried out on a section of the test track with a special low adhesion surface ($\mu_G \approx 0.28 - 0.35$) apart from the difference in adhesion between tests 4 and 3, the curve of test 4 was of a progressively

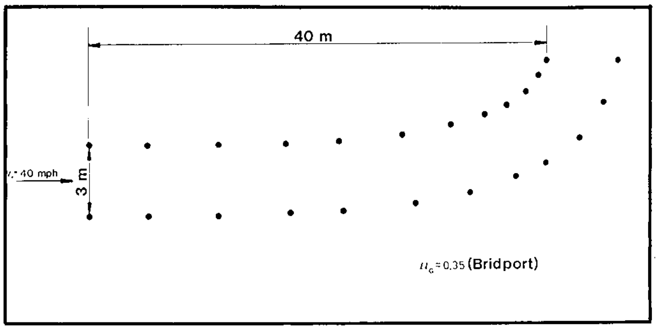

Figure 12. Braking in a curve with a tightening radius

tightening radius (Figure 12). The driver again received instructions from the observer to brake hard to a standstill on entry into the test area

In this driving manoeuvre only one driver of the standard car out of 77, was able to stop the car without leaving the lane. All the other drivers skidded out of the lane with locked front wheels. 66 drivers locked all four wheels.

In the SCS car, 28 test drivers also touched the marker cones or left the lane, however only 2 drivers completely left the marked lane, the other 26 drivers only disturbed two or three of the marker cones without leaving the marked lane. However, since the definition of a critical situation includes contact with a marker cone, the average driver is more prone to

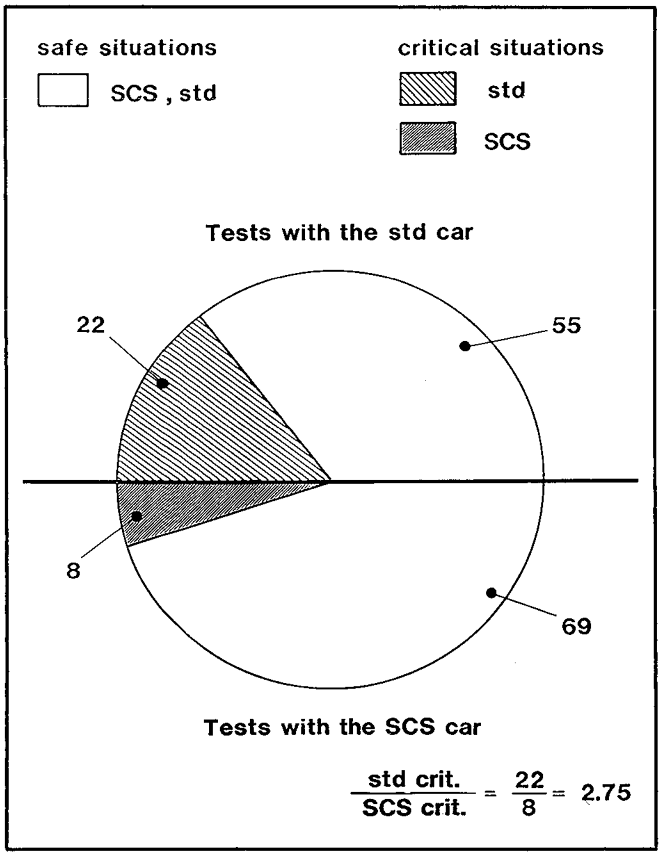

Figure 11. Critical situations in braking in a bend

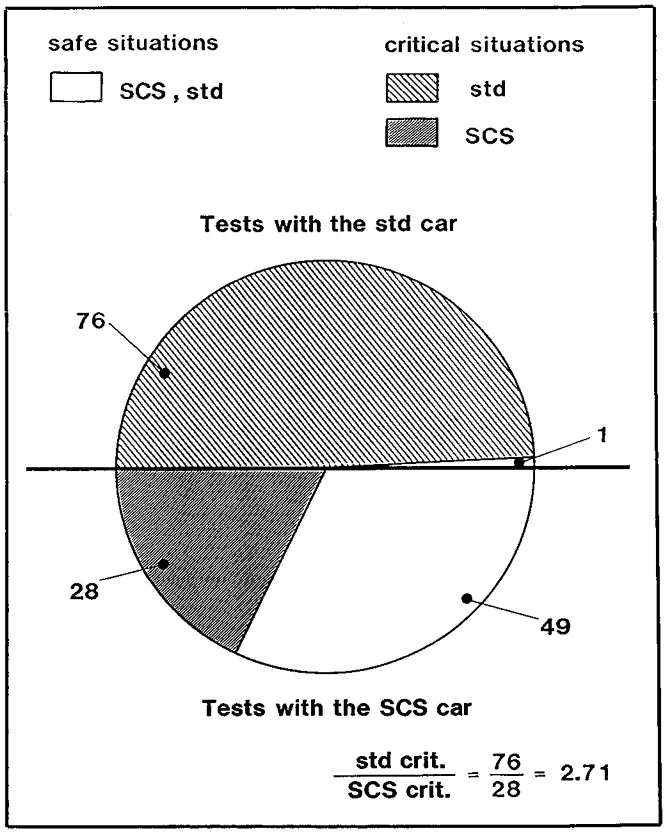

Figure 13. Critical situations by braking in a curve with a tightening radius

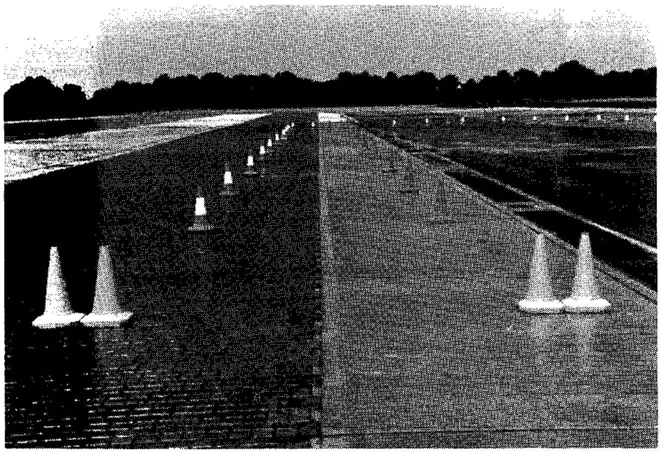

Figure 14. Split μ braking.

failure in the standard car by a factor of 2.7 (Figure 13).

This test is less demanding on driver capability—all drivers automatically attempted to keep to the course by strong steering action—thereby demonstrating the technical capability of the SCS car.

In test manoeuvre 5 (split μ braking) the driver was requested to enter the test zone at 35 mph. The left hand side has a surface of basalt with a low adhesion ($\mu_G \approx 0.28$) and the right hand side has a concrete surface profiled across the vehicle path giving an adhesion of $\mu_G \approx 0.8$ (Figure 14).

In order to ensure that the car straddles the two different surfaces, the test lane was reduced to 2.5 m (Figure 15). In this test also, the driver was instructed to brake hard on entry into the test zone.

Only 8 test drivers were able to bring the standard car to a standstill inside the marked lane. Of the 69 drivers of the standard car who recorded critical situations, 55 drivers locked all 4 wheels and spun the car through 90° to 180°. The other 14 drivers locked two or three wheels and disturbed three or four marker cones, but maintained directional stability. Of the 11 test drivers who recorded critical situations in the SCS car, 2 drivers spun the car as a result of applying the wrong steering lock. The other nine disturbed two or three marker cones but without losing directional stability to any great extent (Figure 16).

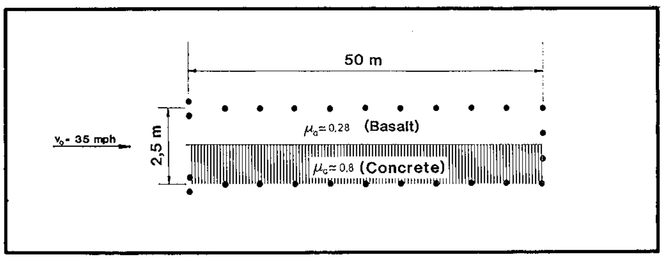

Figure 15. Split μ braking

The high advantage of the average drivers in the SCS car (std crit/SCS crit = 6.27) is directly related to the technical capabilities of the SCS car in such road conditions. The average driver does not meet such conditions very often in practice, and is, in consequence, not well prepared to take corrective driving action.

Conclusion

Only with the assumption that the various critical driving situations outlined above are equally relevant to accident incidence, and the necessary input from driver and vehicle is comparable, can a summarised total advantage of the SCS car over the standard car to the average driver, be deduced. With these assumptions, the advantage factor for the SCS car over the standard is 2.4 for the average drivers in critical driving situations. Figure 17 shows diagrammatically the summarised critical situations for all tests tabulated to show the standard car results in the left column and the SCS car in the right column.

The test programme clearly demonstrates that an automatic anti wheel lock braking system has a great advantage even for inexperienced average drivers by reducing the incidence of critical situations.

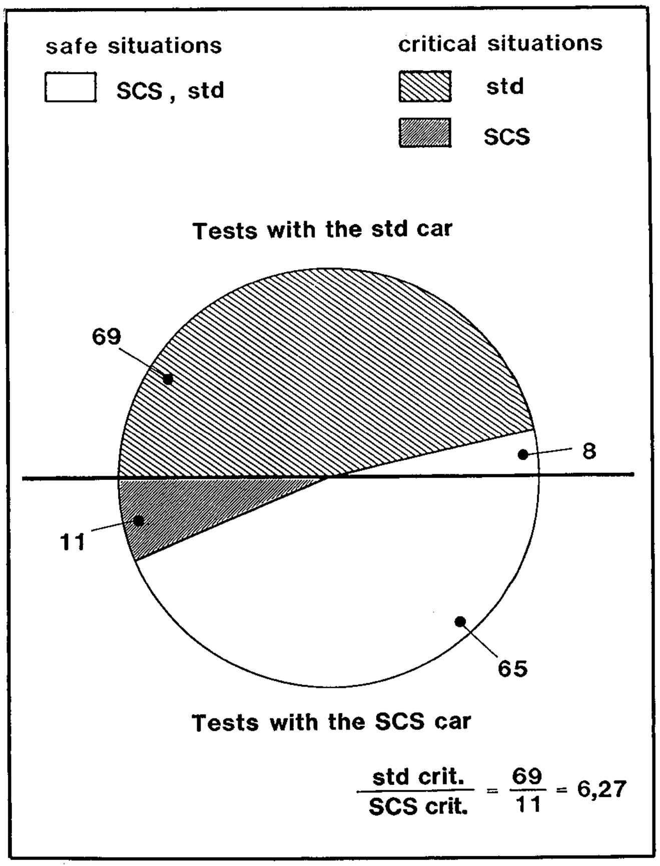

Figure 16. Critical situations in split mue (μ) braking

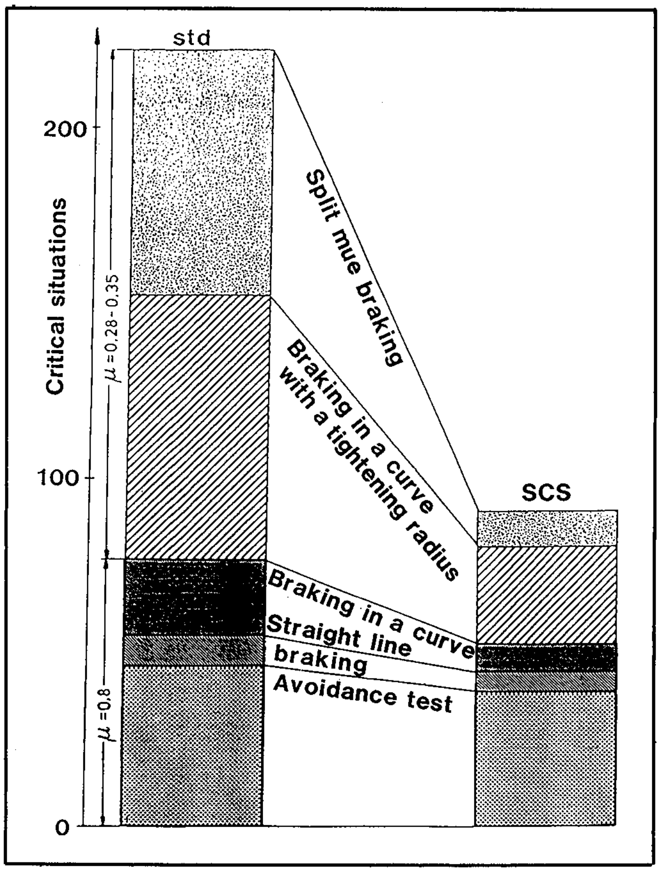

Figure 17. Summarised critical situations with both vehicles

The early introduction of such systems on a universal basis would effect a considerable reduction in both the severity and number of road accidents. Consideration now needs to be given to the question of how can the vehicle driver be given adequate training in order that he is able to fully utilize the potential of ABS systems to promote greater safety on the roads.

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Anti-Lock Brakes for Passenger Cars and Light Trucks

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Good morning. My name is Bob Munson and I am the Director of the Automotive Safety Office of the Ford Motor Company. My purpose in being here today is to acquaint you with the status of anti-lock brake programs at Ford. As perhaps some of you know, Ford was the first automobile company to make electronic four-wheel anti-lock brakes available in the North American market. This actually goes back to 1984 when four-wheel anti-lock systems were offered on some 1985 Lincoln models.

Today I want to discuss several topics relating to Ford's anti-lock brake system program. I will touch briefly on Ford's history of involvement with anti-lock brakes. I will also discuss the nature and performance of the systems currently available in our North American products. And finally, I'll give you an outline of our forward model plans for anti-lock brakes.

Let me turn first to our current truck programs. An electronic rear wheel anti-lock system supplied by Kelsey-Hayes is standard on all 1987 Ford F-series light trucks and Bronco vehicles, with anticipated combined sales of about 700,000 units.

The Kelsey-Hayes system in our 1987 light trucks prevents rear wheel lock-up at all but the lowest speeds. Controlling rear wheel lock-up is especially difficult in light trucks with conventional brakes because there are large differences in the load on the rear tires between loaded and unloaded conditions. As long as the rear tires continue to roll, they provide some lateral friction which keeps the vehicle tracking and helps the driver maintain control. According to recently released NHTSA research papers, keeping the vehicle from getting sideways in emergency situations may reduce the chances of overturning.

Four-wheel systems have been available on expensive European luxury and performance passenger car models for several years. Our aim has been to develop a more broadly affordable, state-of-the-art system.

Ford's first production anti-lock system was the "Sure-Track" rear wheel system. Ford offered the Sure-Track rear-wheel anti-lock system on some passenger car models as an option as far back as 1968. This option was subsequently dropped in the late 70's for want of customer interest.

Ford has been working since 1981 with ITT Teves to develop a practical four-wheel anti-lock system for our products. As I mentioned a moment ago, we first offered the Teves system on our 1985 Lincoln Continental and Mark VII cars. This system was provided as standard equipment on our 1986 Lincoln Continental and Mark VII. This year it is also standard equipment on the 1987 Thunderbird Turbo Coupe.

I'm sure that most of you are familiar with the concept of anti-lock brakes, but let me give a brief review of the essentials. Anti-lock brake systems are designed to prevent wheel lock-up during braking on most road surfaces. A four-wheel system provides the driver with greater steering control under braking in wet or icy conditions and can also reduce stopping distance, especially on slippery surfaces. The anti-lock function responds when the driver applies enough brake pressure to lock the wheels. The system monitors wheel speed and spin-down rates to sense impending wheel lock. When impending lock-up is sensed for a given wheel, line pressure to that wheel is reduced and modulated. In both the Ford two- and four-wheel systems, electronic sensing and processing makes it possible to cycle the system 12 to 15 times a second to keep the wheels near the peak of the mu/slip curve.

Reliability is obviously a very important consideration in the design of any brake system. The Ford-Teves four-wheel system uses redundant dual microprocessors and three separate braking circuits: one each for the two front wheels and one for the rear wheels. The software in both our two- and four-wheel systems incorporates fault checking which can detect problems with the anti-lock system, and if necessary, isolate it from the rest of the system, leaving a fully functional brake system.

The generally understood advantage of anti-lock brakes is that they largely eliminate the loss of lateral stability caused by locked wheels that can lead to loss-of-control accidents.

It can be shown that anti-lock improves lateral stability. The cornering force falls off rapidly as the wheels begin to lock and when the wheels are fully locked (100% slip), the cornering force is close to zero. By preventing lock-up, the anti-lock system maintains reasonably high levels of cornering force.

Four-wheel systems such as the Ford-Teves system can also provide a substantial improvement in straight line braking performance. The rapid cycling keeps the wheels close to the optimum point on the mu/slip curve.

In one test we made on a wet pavement, the peak mu was at about 20% slip. The Teves system controls slip to keep it close to the point that produces peak mu. The result is a useful improvement in the effective mu on most surfaces.

When we tested for the maximum average braking deceleration from various speeds on different surfaces, the deceleration was higher with anti-lock on all of the surfaces; but the advantage is greater on the more slippery surfaces. On hard, packed snow, for example, four-wheel anti-lock results in a 30% increase in deceleration.

The microprocessor algorithms that control the anti-lock function in both two- and four-wheel systems are very sophisticated. These routines analyze the raw wheel speed data to detect incipient wheel lock-up and send signals to the pressure control valves to prevent lock-up from occurring. The Teves system incorporates algorithms to estimate the surface mu and calculate optimum slip. It is only in recent years that microprocessors with the necessary speed and capacity have been available in economic production quantities.

But make no mistake about it, the four-wheel systems in production today are still expensive. Nevertheless, the installed cost to Ford of the four-wheel systems is considerably less expensive than the same product would have been just a few years ago. And we expect to bring the price down substantially in the future as the volume goes up. Several factors have come together in the last few years to make it possible to begin to offer high performance anti-lock systems on a broader basis.

The first, and perhaps most important, is the availability of reliable and relatively inexpensive microprocessors.

Second is the ongoing engineering development work carried out by ITT Teves, Kelsey-Hayes and others. In particular, the development of inexpensive high speed solenoid valves for pressure regulation was an important breakthrough. It is the combination of these valves with a microprocessor that makes the high cycle rates possible.

And the third factor is that people are more safety conscious now than they were a few years ago. This is true, not only of people in the industry, but our customers as well. I might also add that the very positive encouragement and support of Ford was a major factor in stimulating the suppliers' engineering development that led to current production systems.

The key to broader availability of state-of-the-art anti-lock systems is getting the cost down to the point where it can be offered on low- and medium-priced cars without turning them into high-priced cars. There is a three-way synergy between engineering development, price and demand for a product that operates to drive costs down. As interest in the product grows and the number of units sold increases, further development is stimulated and the cost decreases. This further increases sales, resulting in further decreases in cost, and so forth.

This process is already beginning to operate. We anticipate that an increasingly sophisticated and safety conscious buying public will demand state-of-the-art anti-lock systems.

Right now our plans call for increasing availability of anti-lock systems on passenger cars and light trucks. We plan to have rear wheel anti-lock systems as standard equipment on all of our light trucks by 1990. And we expect that four-wheel systems will be available on our 1990 medium-sized cars and perhaps on some smaller cars as well by the mid 1990's. We also plan to eventually introduce four-wheel systems on our light trucks. By the mid-1990's state-of-the-art, four-wheel electronic anti-lock systems should be available on most of Ford's U.S. and European passenger cars and domestic light trucks.

To conclude, I would like to emphasize the point that at Ford we regard anti-lock brakes in a very positive way, not as something we must reluctantly accept as inevitable. We believe that anti-lock brakes provide a meaningful safety benefit which our customers will come to demand and whose development we will continue to pursue.

Visual Performance Characteristics in Vehicles

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Abstract

Almost all of the information required to drive a vehicle is obtained through the eyes. Instruments convey information on vehicle conditions and their visibility is of primary importance. A study is conducted on two aspects of visibility in vehicles.

(i) Readability of instruments
(ii) Perception of objects in forward field of vision when reading instruments.

Experiments are conducted to analyze quantitatively the effects of display parameters, environment, and age on the above factors in order to produce instrumentation which is easy for older drivers to read. The parameters chosen are display brightness, character size, distance between the eye and display, and location of the display. The subjects are separated into two groups, one in their twenties and the other in their fifties. All have visual acuity of 0.7 or above. Each test run consists of a 20 min. adaptation time followed by the visual task. A Landolt's Ring is used as the display character and subjects are asked to indicate whether the Landolt Rings opens to the left or right by operating a hand-held two-way switch.

Introduction

Senior citizens (aged 65 years and older) in the more advanced nations of North America and Europe accounted for approximately 15% of the overall population in these countries in the 1980s. It appears that this figure will continue to increase annually. In Japan, too, a sudden aging will occur, achieving the present percentage of about 15% of Western nations in the year 2000. Predictions are that this figure will then reach approximately 20% by 2020.

Parallel to this trend, the number of senior citizens possessing a driver's license, as well as the number of accidents these people cause, appear to be on the rise. The physical reactions of old-aged drivers has suddenly become, as a consequence, a matter demanding urgent attention.

With advancing years, a driver's physical abilities change. This change involves a decrease in visual, auditory and motor functions. Among these, the loss of visual acuity is said to be exceedingly fast.

The driver of an automobile generally shifts his or her attention repeatedly while driving, from the road ahead, to objects on the road and the various readouts on the dashboard. In doing so, the driver must conduct a series of frequent eye focalization actions. It is known that among the visual capacities, deterioration of the focal regulation function is known to begin, and accelerate rapidly, from the middle 40s.

This paper examines the factors involved in visual recognition by older drivers when regulating the focus of vision in response to objects in the visual field ahead and objects within the vehicle. The optimum conditions, from the older driver's viewpoint, of visual displays are also clarified. For this research, a panel of drivers in their 50s was selected to represent senior drivers.

Analysis of Target Recognition Time at Night

It was hypothesized that, for an examination of target visibility, night time driving would present the most difficult conditions. Accordingly, a test designed for certain degrees of lighting representative of evening hours was conducted to examine target recognition time.

The experiment

The subject is dark adapted to 0.1 cd/m ^{2}, a representative road surface degree of brightness which would be visible from the driver's seat. The subject's focus is directed to a screen three meters in front. At a given directive signal, a target (Landolt's ring) is displayed under the various conditions listed below. The subject notes the direction of the gap in the ring (facing either left or right), indicating this by appropriate switch response. Response time is determined by the time which elapsed between the directive signal and subject response. A schematic diagram of this test is given in Fig. 1.

Test condition:

Target height	1.8; 2.5; 3.5; 5.0 and 10 mm.
Target luminance........	0.1; 1.5 and 7.0 cd/m ^{2}.
Target distance	40; 70; 80; 90 and 160 cm.
Target position..........	
from speedometer position:	left/right - 15° and 30°
from screen center:	above/below - 15° and 30°
Panel	20s age group - 5 male subjects (Visual acuity of 0.7 or above, having driver's license)
	50s age group - 5 male subjects (Visual acuity of 0.7 or above, having driver's license)

Results and discussion

(1) *Target height and luminance* A speedometer placed in a suitable position (70 cm, at an angle 30° below eye level) was used to display visual objects ("Targets") having varying height and luminance. Fig. 2 presents graphs relating measured response time and target height for different degrees of luminance.

1. With a target luminance representative of brighter displays, such as for 7 cd/m ^{2}, the difference in response time for small targets is small.
   With a target luminance of 1.5 cd/m^2, the response time for small targets is long for subjects in their 50s.

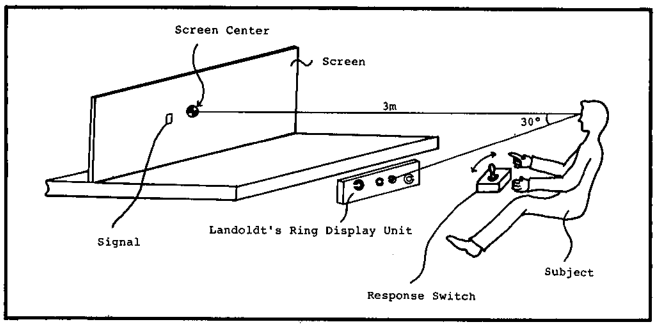

Figure 1. Schematic diagram of test

2. With a target luminance representative of darker displays, such as 0.1 cd/m ^{2}, the response time increases as the height of the target decreases. This tendency is more noticeable for subjects in their 50s.

The data suggest striking influence of advancing years in the visibility of targets when height and

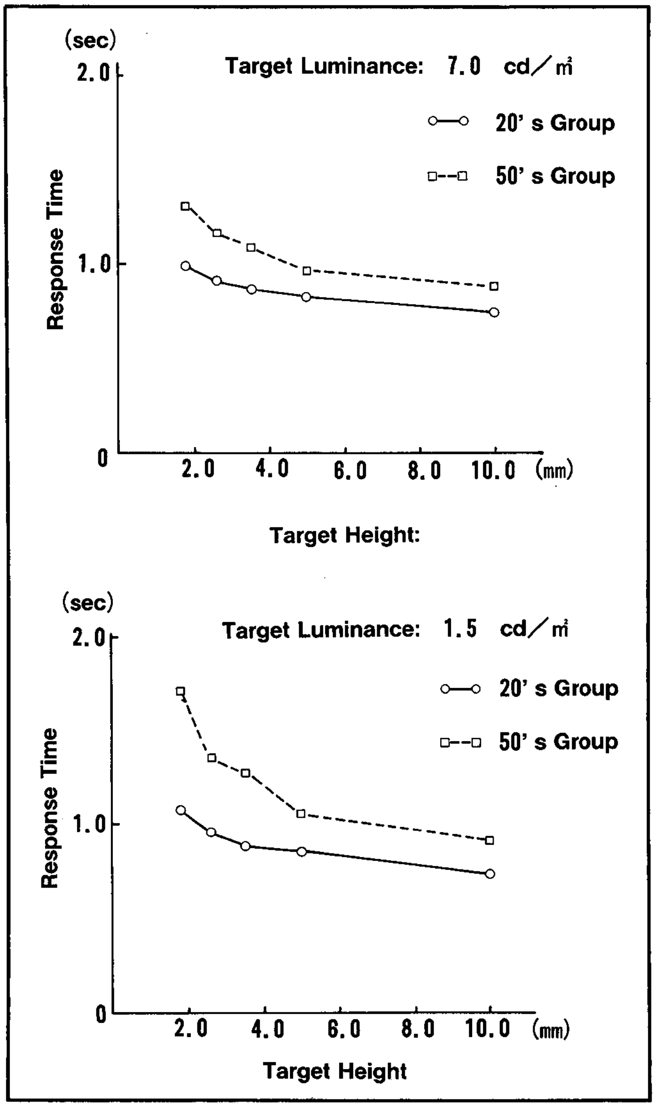

Figure 2. Response time vs. target height

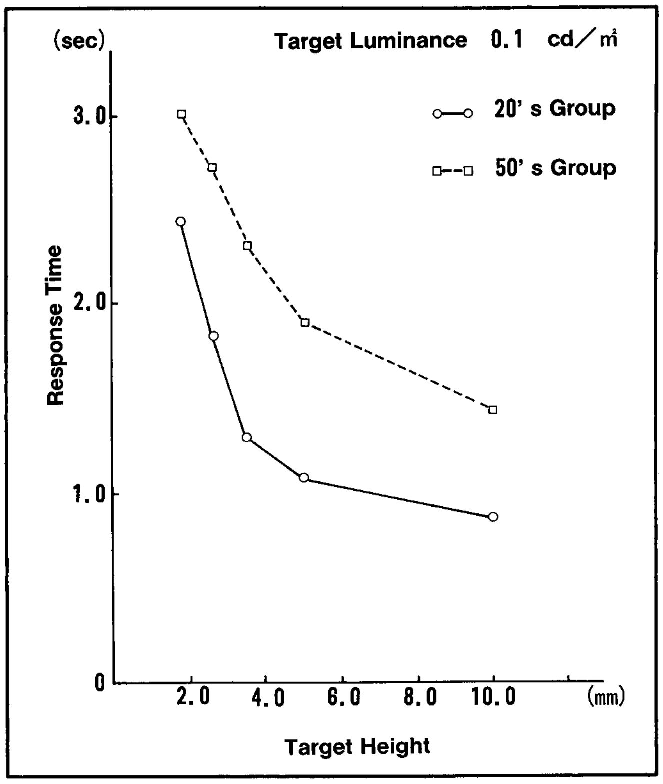

Figure 2. Continued

luminance are reduced. A difference in response time between those in their 20s and 50s has been noted and experimentally confirmed.

(2) *Target distance* Figure 3 contains two graphs relating response time to varied target distances. The target was displayed at eye level and directly in front of the subject. The angle was approximately 0.3° (or 3.5 mm at 70 cm).

1. For the age group in their 20s, there was a small difference in response time for targets within a 40 to 160 cm distance range.
2. For the age group in their 50s, there was a tendency toward lengthened response time as target distance decreased. This tendency was especially marked when target luminance was reduced to 0.1 cd/m ^{2}.

In light of the above results, it appears possible to reduce the response time by lengthening target distance, given reduced luminance and height.

(3) *Target position* The relationship observed between target position and response time is plotted in Fig. 4. The data concerns visual recognition time of targets (height: 3.5 mm; luminance: 1.5 cd/m ^{2}) presented in varying locations within a 30° range to the left or right of the speedometer position. Additionally, Fig. 5 presents data concerning response time for the subjects when presented with targets in varying

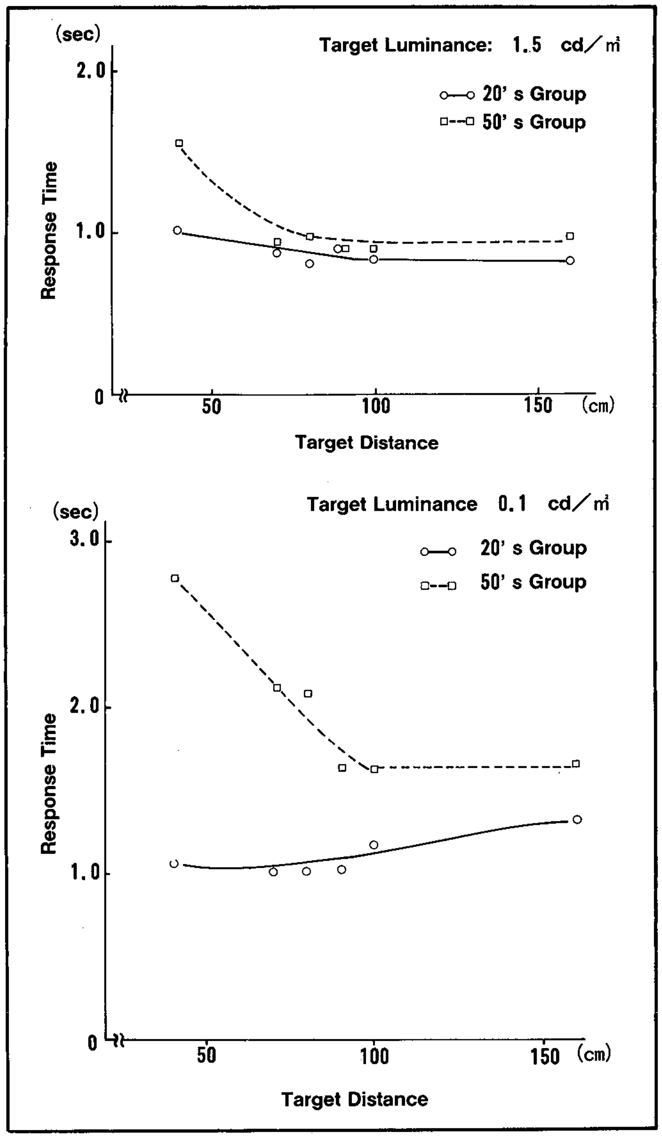

Figure 3. Response time vs target distance

locations within a 30° range above or below their center of vision.

1. For targets presented in various positions within 30° of the horizontal direction, no change in response time was noted for either group.
2. Both groups showed lengthened response time for objects which were presented above or below the direction of vision. For all target positions, the age group in their 50s had a 10% longer response time in comparison to the other group. Accordingly, differences can be said to exist between group response times, but aging cannot be said to effect response time as target objects are placed more distant from the direction of vision.

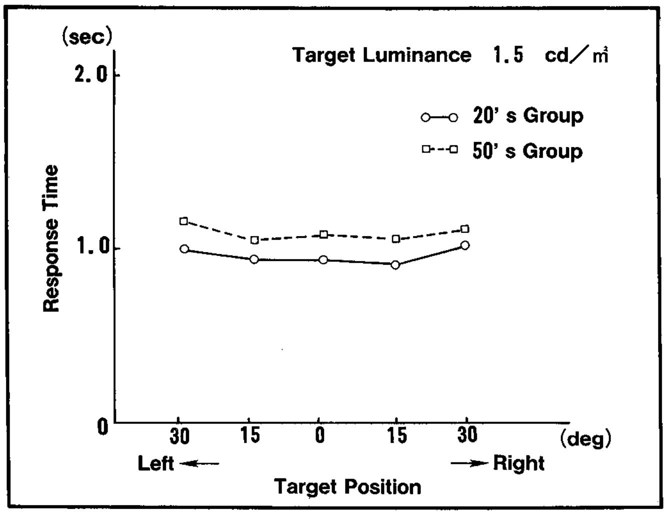

Figure 4. Response time vs. display position (horizontal)

As a result of analyses conducted, the influence of luminance, distance from subject and relative position (up/down) upon the visibility of target objects have been clarified. To improve the visibility of such targets for older drivers, it appears particularly effective to consider target height and luminance.

Analysis of Target Recognition Time in Moving Vehicles

Laboratory tests have been conducted to identify patterns of change in response time for two age groups in their 20s and their 50s when presented with different target objects under various conditions. To confirm the validity of these results under road test conditions, an examination of response time was conducted during a night driving test.

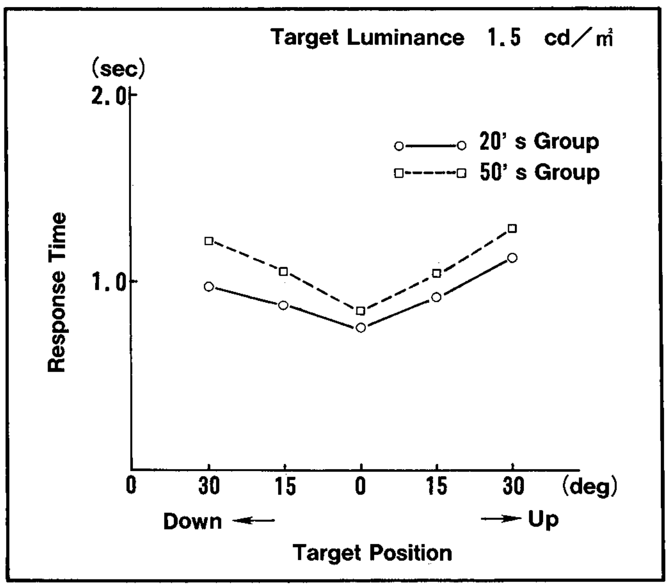

Figure 5. Response time vs. display position (vertical)

The experiment

While the subject is driving at night, a visual target (a Landolt's ring) is presented at random, but simultaneously with the sound of a buzzer. The subject is to identify the position of the gap in the ring (left or right) and indicate this response using the switch provided. The time between the initiation of the buzzer sound and the subject's response is measured as the time required for visual recognition, termed 'response time'. A photograph of the test equipment is given in Fig. 6.

The test was conducted while driving on a proving ground at an approximate speed of 40 km/hr. with visual target displayed in the speedometer position.

Test condition:

Target display distance ..	approx. 70 cm
Target height	2.5; 3.5 and 5 mm
Target luminance........	0.1; 1.5; 7.0 cd/m ^{2}
Panel	Age group in their 20s—5 persons Age group in their 50s—5 persons

Results and discussions

Fig. 7 presents three graphs which plot response time as a function of speedometer position for three degrees of luminance. The results of this road test conform well with the laboratory results. To restate the general conclusion: as target luminance and height decrease, the response time for both groups lengthens. That having been said, the increase on the part of the age group in their 50s is more prominent.

In light of the above, it has been ascertained that the laboratory test results may be effectively utilized

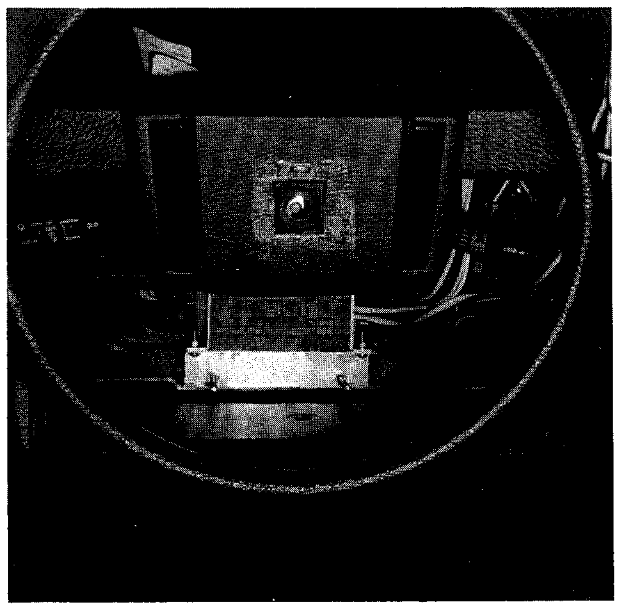

Figure 6. Road test equipment

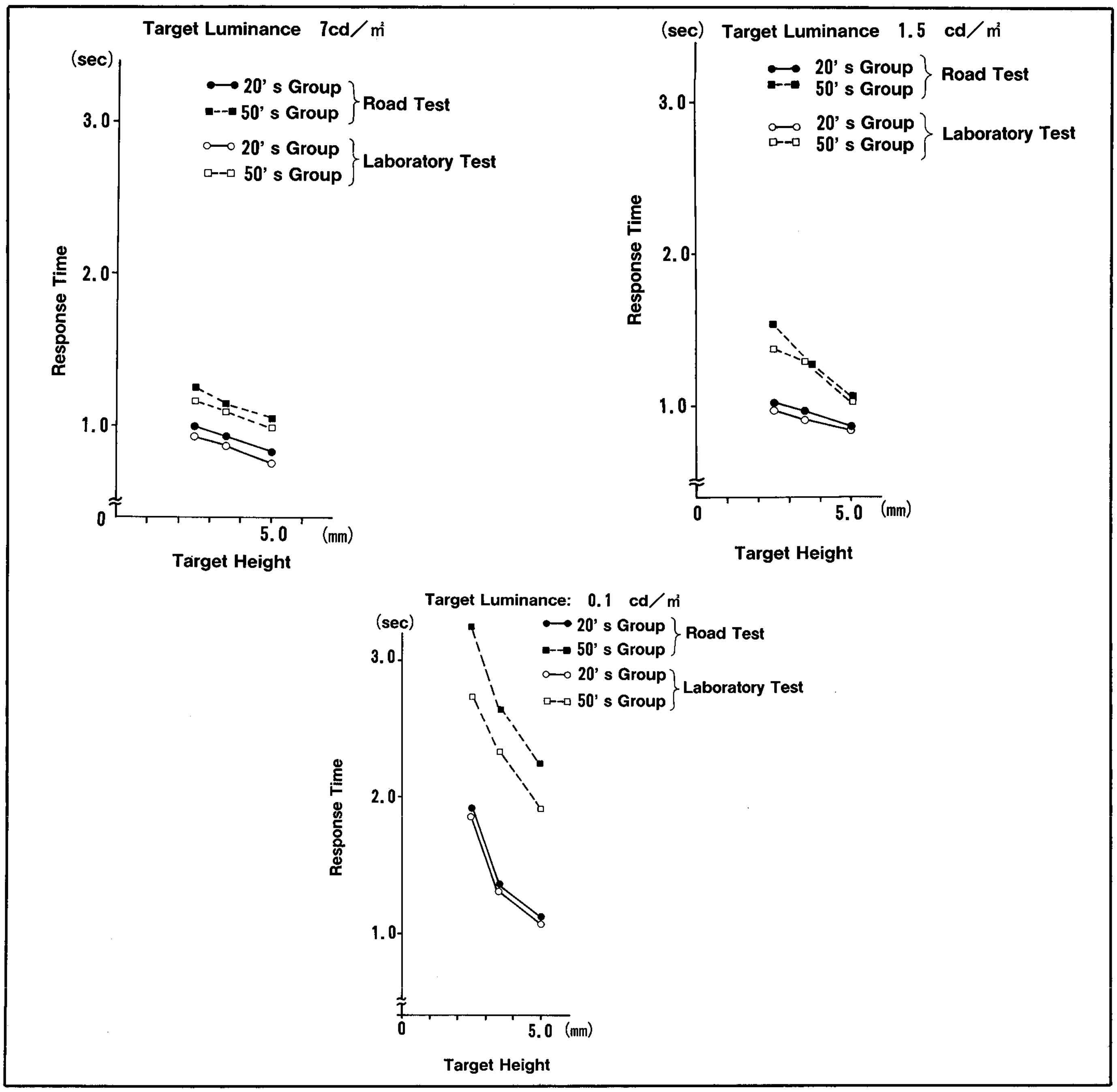

Figure 7. Response time vs. display position

to clarify the target conditions useful in improving the visibility of such objects for older drivers.

Analysis of Conditions Facilitating Visibility

Standard response time

In order to determine the target conditions which facilitate visibility, it is necessary to determine a standard response time. This is the maximum interval judged to be acceptable between target recognition and the switch manipulating response.

For the purpose of this investigation, the time during daylight driving when attention is focused on the speedometer, and the time to respond by switch are calculated separately, and the total value of the two has been set as the standard response time.

In order to avoid having the drivers become self-conscious, they were told of other objectives. By means of a road test utilizing installed eye cameras, the time during which driver focus is on the speedometer is recorded, and an average time period determined. In the present study, this was found to be 0.7 sec. Next, a Landolt's ring is displayed while attention is focused upon the determined target position. The time needed to throw the switch to the left or right is then determined. In this experiment, the average response time is 0.6 sec.

To total value of the two, 1.3 sec., was established as the standard response time.

Conditions facilitating target visibility

The results of a study based upon the values of target height and luminance, for the response time of 1.3 sec. in Fig. 2 (vertical axis), are given in Fig. 8. The area above each age group line represents the region within which the age group in their 20s and the age group in their 50s respectively are able to provide a recognition response to a visual target within 1.3 sec. This becomes, therefore, the range of an optimum target and is identified by oblique lines.

Consequently, the following can be stated regarding illumination conditions for an average night period (adaptive luminance: 0.1 cd/m^2).

1. The age group in their 20s are able to respond to a visual target within the standard response time if target luminance and height respectively equal or exceed 1.5 cd/m^2 and 2.5 mm.
2. The age group in their 50s can recognize a visual target within the standard response time if target luminance and height respectively equal or exceed 1.5 cd/m^2 and 3.5 mm.

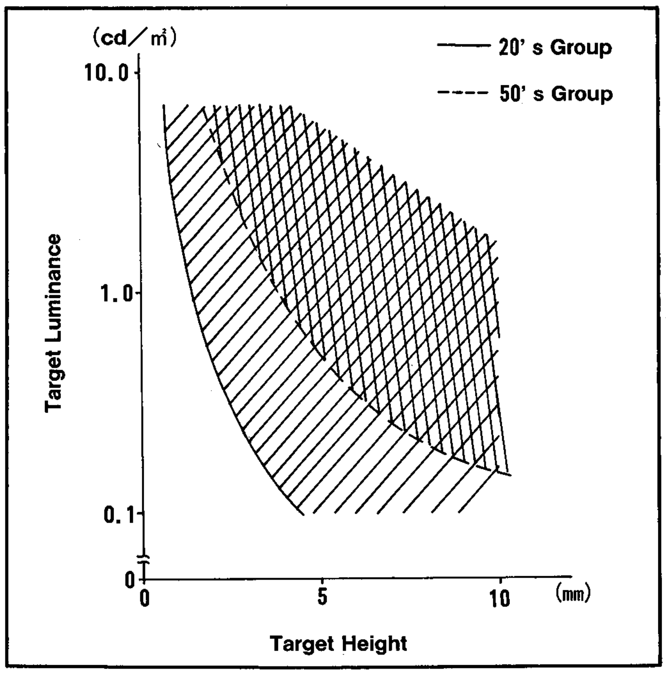

Figure 8. Target luminance vs. height

Influence of Instrument Panel Luminance Upon Forward Visibility

In the preceding paragraph, the optimum conditions were identified for visual recognition, within a standard time period, of displays in vehicle components such as a speedometer. To determine conditions for display of vehicle components, it is also essential to insure that the luminance of these objects within the car does not become a hindrance to the field of forward vision and the ability to spot potentially dangerous objects within this field. With this issue in mind, a study was made concerning the degree of visibility needed for various speedometer luminance conditions.

The experiment

A panel having 0.01 cd/m^2 luminance is set in front of the subject. The subject is then given adequate time for dark adaptation. His/her attention is focused upon an indicator in the upper part of the screen. The sound of a buzzer directs the subject to watch for speedometer units having various degrees of luminance. The units used were, in fact, those of the current Nissan Bluebird model. Following this, the buzzer is used to direct the subject's attention to the screen once again. At the same time, in one of the three points on the screen (visual angle: 0.2° x 0.3°) the target is illuminated for an instant. The subject, as shown in Fig. 9, responds by means of a hand or foot switch, indicating the location of the target: left, right or directly ahead of the subject.

If the subject response is correct, target luminance is reduced. The target luminance is increased if the subject errs. This procedure is repeated 15 times to ascertain the average luminance value needed for target visibility.

Test conditions:

Speedometer target (unit) luminance ... 0, 10 and 40 cd/m^2

Target display position (relative to viewing subject) ..
(relative to viewing subject) left (5°)
directly ahead (0°)
right (5°)

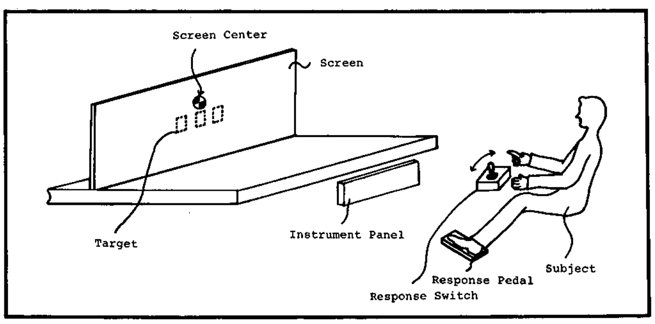

Figure 9. Diagram of panel luminance test

Results and discussion

Fig. 10 indicates the relationship obtained between speedometer luminance and the target luminance differential required for visibility. The difference between target and screen luminance is termed the luminance differential threshold region.

For both the 20- and 50-year-age groups, as speedometer luminance is raised, the luminance threshold region also rises. It is easily understood that it becomes more difficult to notice the target. Consequently, given night driving conditions, reduced luminance of the speedometer and other devices facilitates recognition of obstacles within the forward visual field.

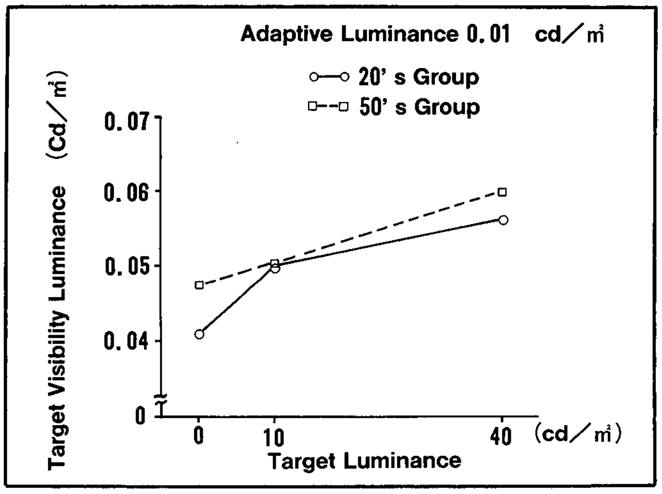

Figure 10. Visibility vs. target luminance

Conclusion

The data obtained from the preceding examination of the factors involved in object visibility for subjects in their 20s and 50s leads to the following conclusions:

1. Object luminance, height, display distance and position all influence visual recognition. The 50-year-age group, compared to the younger group, is greatly influenced by target luminance and height.
2. If target height and luminance are increased, visual response time decreases. This suggests that improvement in prompt recognition is possible. Conversely, this leads to increased difficulty in recognizing obstacles in the forward visual field. Accordingly, the determination of optimum object conditions for older drivers requires careful consideration of both trade offs in determining optimal conditions.
3. While the present work employed 50-year-old subjects in the 'senior' age group, it will be necessary to analyze data for even older subjects.

Recognition of Pedestrians During Nighttime Traffic

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Abstract

Based on the European- and US low-beam situation, tests were made about the recognition distance for pedestrians. The following parameters were investigated

shape of the headlamps
mounting height of the headlamps
inclination of the headlamps
reflectance factor of pedestrians.

Introduction

Starting from the measured contrast of pedestrians in the streets during nighttime, in dynamic experiments in the street the recognition distance for pedestrians illuminated by low-beam headlamps was measured. During the tests several parameters were changed.

Pedestrians in the Street

In Figure 1 schematically a two-lane road is shown as seen by a car driver. The pedestrians are at a distance of d = 50 m in front of the car, illuminating the street by means of low-beam headlamps. C shows the cut-off-line of the two headlamps, correctly aimed with a mounting height of h = 65 cm.

Z describes the border lines to zone III in the European regulations. A is the area of an overhead sign, B of a shoulder mounted sign at 50 m distance. The Figure shows that the pedestrians are mainly illuminated by the light in-between "C" and "Z" (dotted area). This area is without legal requirements.

Light-Distribution of Headlamps

In Figure 2 and 3 the vertical illumination distribution for two different types of headlamps are shown. The distribution is measured through the points 75R and B50L of the European regulations.

In Figure 2 the light-distribution for a typical European low-beam headlamp is shown, in Figure 3 for a SB-headlamp.

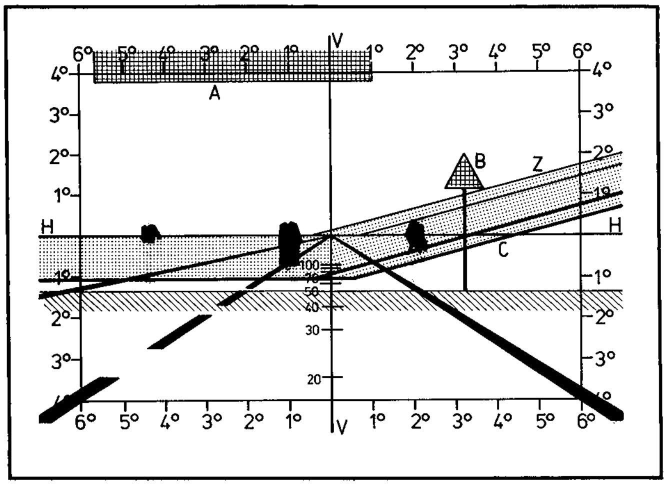

Figure 1. Perspective drawing of a two-lane road
A : overhead sign
B : shoulder mounted sign
Z : border to the illumination-zone III
C : cut-off-lines of two headlamps
Pedestrians in 50 m distance

Reflectance of Pedestrians and Recognition Distance

In the first test the influence of the reflectance factor of the clothing of pedestrians on the recognition distance was investigated. The results are plotted in Figure 4.

The difference in recognition distance for the object-positions left-hand/right-hand side of the street is 1 2 ... 1 5 m. The difference in distance v for black, grey, white object is 25 m. The difference between the object (white, right) and the object (black, left) is more than the factor 4.

Type of Headlamp and Recognition Distance

In Figure 5 the influence of the type of the headlamp, fulfilling the same regulation on the recognition distance is shown.

For the worst situation (object black, left) the difference in distance v between the 3 headlamps is v $\approx$ 10 m. The value increases to roughly 30 m for the object (white, right).

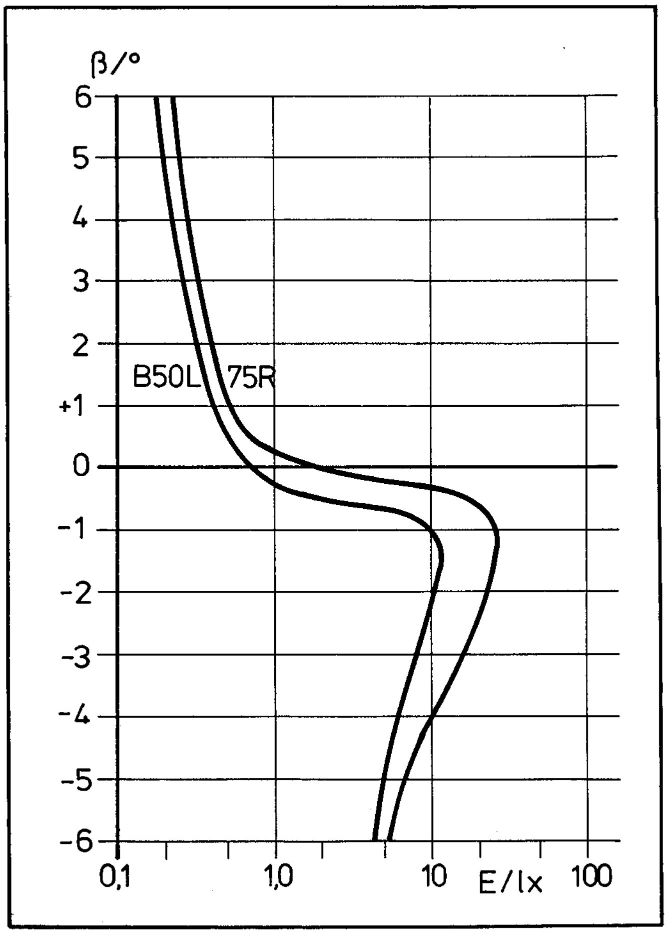

Figure 2. Vertical distribution of the illumination E for an European headlamp measured vertical through the points 75R and B50L

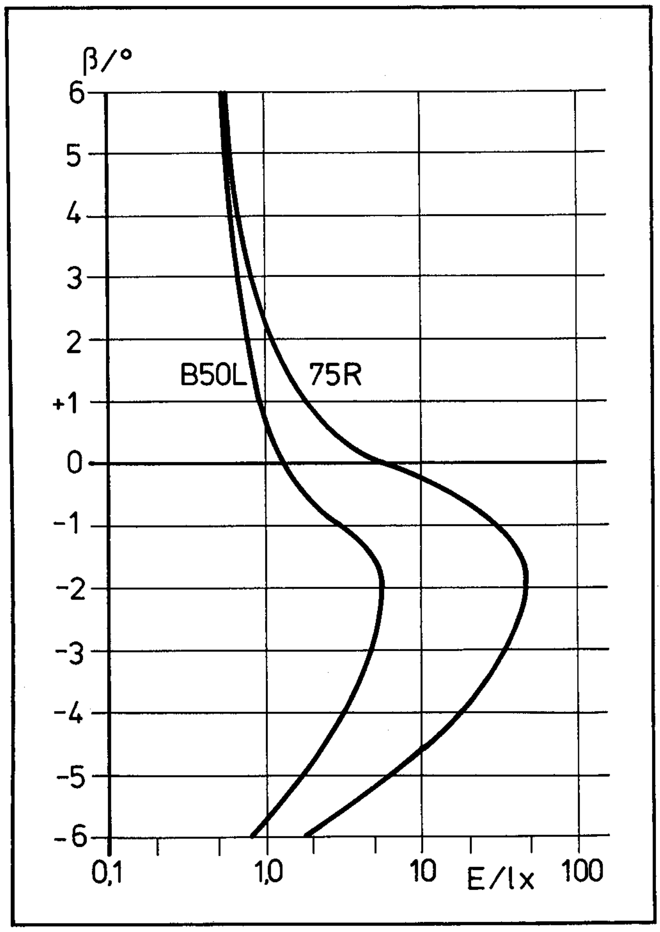

Figure 3. Vertical distribution of the illumination E for a SB-headlamp measured vertical through the points 75R and B50L

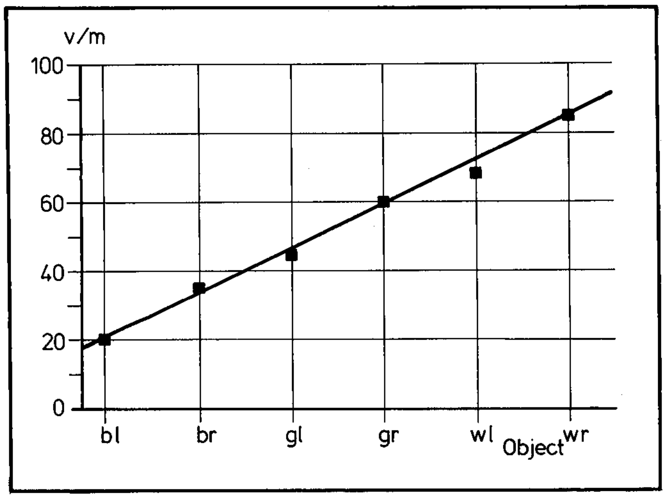

Figure 4. Recognition distance v for different objects
b : black object
g : grey object
w : white object
l, r : left, right position of the object in the street

Mounting Height of Headlamps and Recognition Distance

In Figure 6 the influence of the mounting height h of headlamps on cars on the recognition distance v is shown.

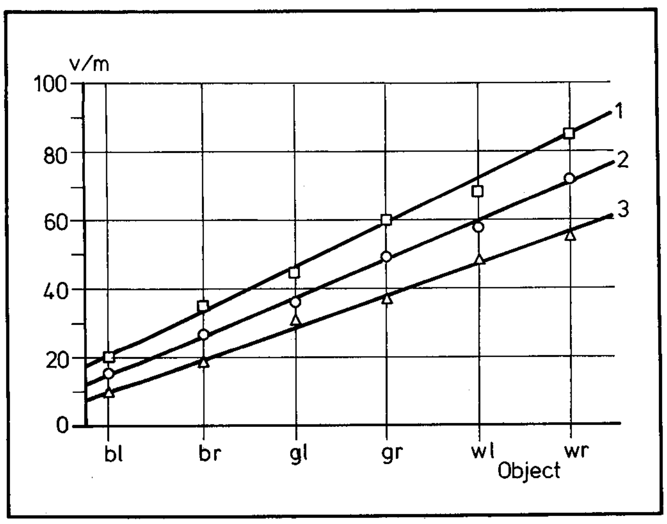

Figure 5. Recognition distance v for object and different headlamps
b : black object
g : grey object
w : white object
l, r : left, right position of the object in the street
1 : rectangular headlamp
2 : 7"-headlamp
3 : 5 3/4"-headlamp

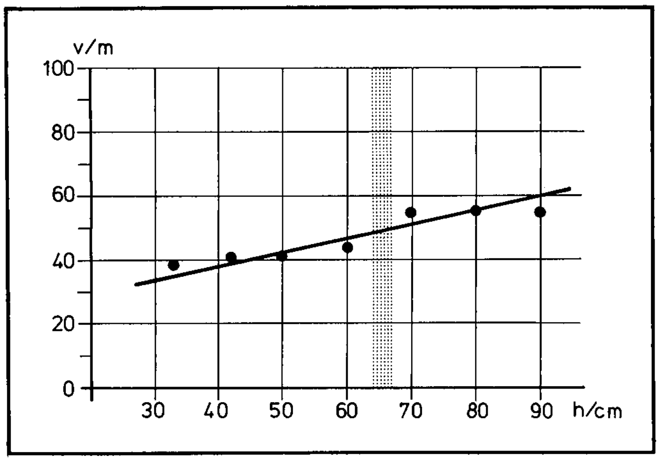

Figure 6. Recognition distance v and mounting height h of headlamps

The dotted area in the mounting height used on several cars. The increase in the height h of 10 cm gives a gain of recognition distance of d ≈ 4,5 m.

Inclination of Headlalmps and Recognition Distance

The influence of the inclination of a headlamp of the recognition distance can be shown in Figure 7.

The dotted area describes the normal inclination of headlamps. With increasing inclination of the headlamp the recognition distance decreases. A change of inclination of ± 1% for the normal inclination gives a change of the recognition distance v ≈ ± 10 m.

Voltage of the Headlamp and Recognition Distance

A very often underestimated effect is the influence of the voltage of a headlamp on the recognition distance. In Figure 8 the influence of the voltage on the recognition distance is shown.

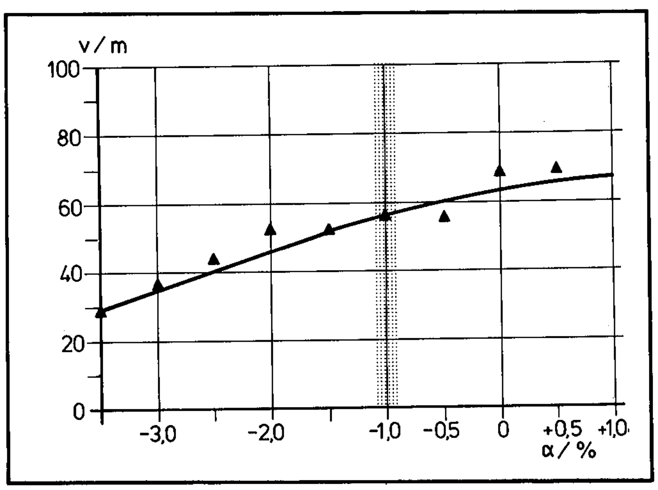

Figure 7. Recognition distance v and inclination of the headlamp

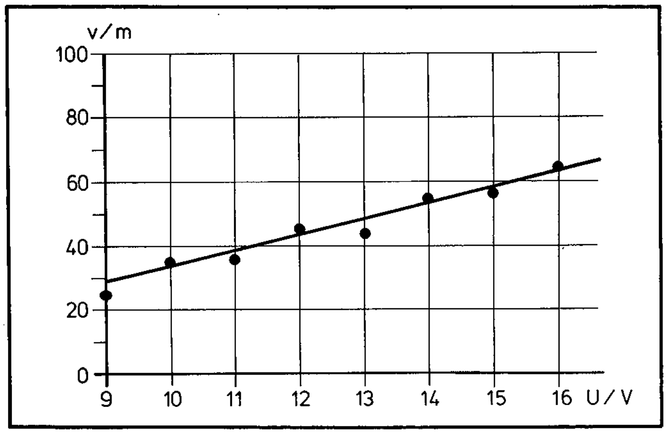

Figure 8. Recognition distance v and voltage at the headlamps

Out of these results a quotient of 5 m/1 V can be derived; that means a change in the voltage of 1 V gives a change in recognition distance of 5 m.

Conclusions

The recognition distance for pedestrians illuminated by low-beam headlamps can be improved by several means,

- optimisation of the geometry and optic of headlamps
- improvement of the reflectance of pedestrians
- optimisation of the mounting height of the headlamp
- introduction of headlamp-leveling-devices
- improvement in the power supply.

The possible improvement in the light distribution can be shown with Figure 9.

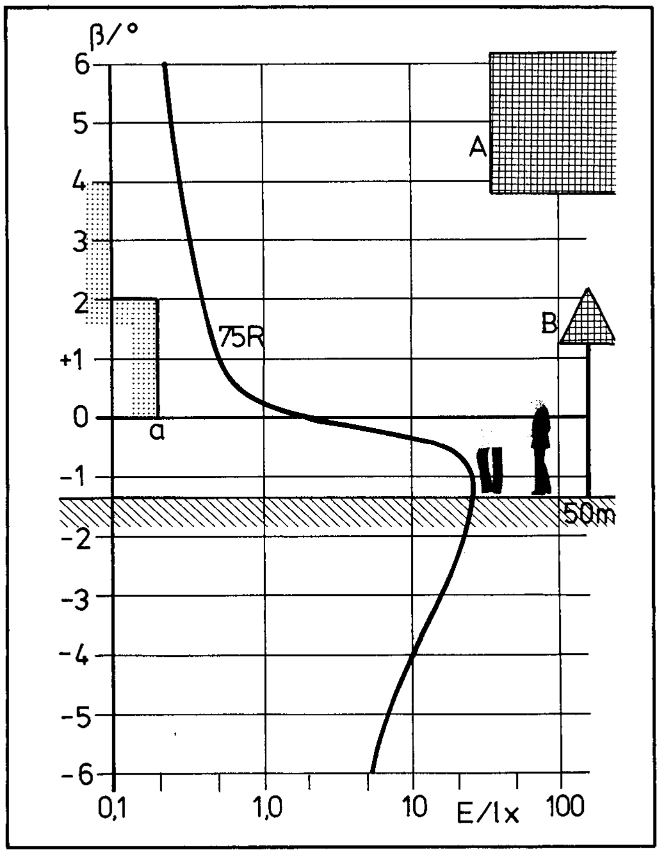

Figure 9. Vertical distribution of the illumination E for an European headlamp measured through 75R
A : overhead sign
B : shoulder mounted sign
a : limits for minimum "glare" values
Pedestrians in 50 m distance

The pedestrians are standing at a distance of 50 m in front of the car. By introducing a controlled amount of light above the horizontal line, pedestrians closer to the car are better recognized. As a sample the dotted boundary a, as proposed in Europe, is plotted in the Figure. For better illumination beyond the horizontal line, the gradient in the cut-off must be increased; this is possible for example with elliptical headlamps.

Contribution of Head-Up-Displays (HUDs) to Safe Driving

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Abstract

Head-Up-Displays (HUDs) have long been used in airplanes; however, it has not yet been verified how the difference in environment would affect their legibility and/or driver visibility if they were to be used in automobiles.

Experiments were conducted based on the Double Task Method in which subjects were made to identify both the display and objects in the forward field of vision in a short time period of 0.3 to 0.5 seconds. The results show that the display recognition error rate is reduced when the display is positioned on a plane from horizontal to 10° below the forward line of vision and at a distance of 0.8 or more from the eyes. They also show that improved legibility obtained by HUDs reduces the recognition error rate in the

forward view. Measuring the time required to recognize the display using an Eye-Mark-Camera, it was shown that recognition time with HUD is 0.2 seconds less than with conventional digital displays. Another experiment involving tinted Combiner on which the HUD is projected was carried out, in which subjects were tested on their perception of objects appearing for 0.27 seconds in the forward field of vision. The result shows that the recognition error rate does not increase due to tinting of the Combiner providing that transmittance is 40% or more.

These results both substantiate the HUDs' high legibility while, at the same time, proving its effectiveness in assuring safe driving from the standpoint of enhanced driver attention to the front view.

Introduction

Head-Up-Displays were first installed in military airplanes in the 1940's. At present they are also widely used in commercial airplanes as well. The typical structure of Head-Up-Displays for airplane use is shown in Fig. 1. A pilot is able to both accurately and quickly perceive the external scene through the canopy along with information such as altitude, velocity and landing point displayed on a combiner. Therefore, HUDs are of great assistance to pilots, especially aiming weapons and landing in adverse weather conditions.

Some studies verifying the advantages of using HUDs in airplanes have been carried out. For example, Naish(1) and Fischer(2) et al. have made it clear that pilots can perceive both objects in the forward view and information on the display more quickly and accurately than with conventional display panels.

In regard to recognition of superimposed visual sources, Neisser and Becklen(3), (4) et al. have reported experiments that indirectly supports the advantages of HUD from the point of view of "selective attention".

In automobiles, number of display items along with the number of older drivers has been increasing while,

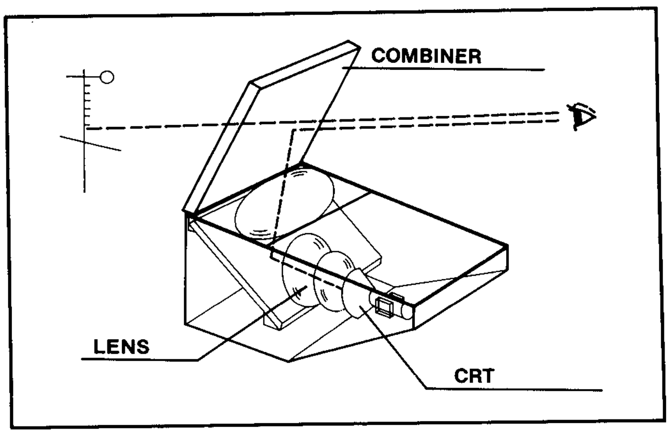

Figure 1. Typical construction of airplane HUD

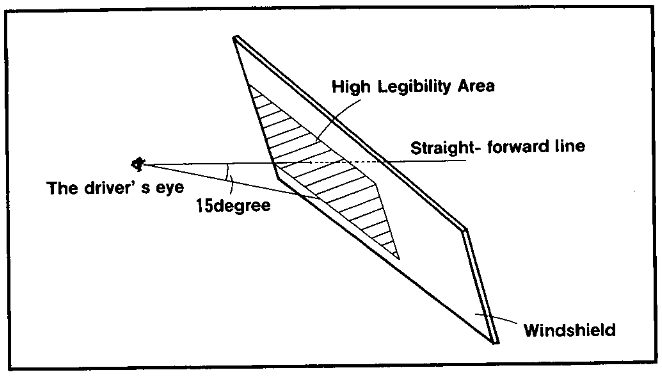

Figure 2. High legibility area of displays

at the same time, vehicle speed is getting faster. Therefore, display systems which satisfy the needs of a variety of customers are required. H. Izuka and Y. Seko et al.(5) reported what they found to be the most suitable place for correctly recognizing information displayed when the driver is looking forward (see Fig. 2).

Both this background and the results of these experimental reports suggest that HUD could provide the answer to new demand trends in automotive displays. This paper, which takes into consideration differences between the airplane and automotive uses of HUD (as shown in Table 1), presents the advantages of using HUDs in automobiles as verified through experimental investigations.

Experiments

Experiment 1. Difference in legibility between HUDs and conventional Head-Down-Displays (HDDs)

The effect on display legibility that is produced by the difference in movement of the visual point between HUD and HDD was examined based on the "Double Task Method". Two visual sources were projected in a short time period of 0.48 seconds. One was a pair of Landolt's rings which was projected on a screen, and was meant to correspond to objects in the forward view. The other was 2 digits of random numbers, which correspond to the display. Fig. 3 shows the setup and Table 2 shows the conditions of the experiment.

Table 1. Difference between automobiles and airplanes.

	Automobiles	Airplanes
Distance of view	Near	Far
Appearance of objects in view	Abruptly	Gradually
Forward view	Road, buildings, vehicles, pedestians, etc.	Sky, clouds runways, airplanes, etc.
Driver/pilot	Not specially trained including unskilled	Well trained and skilled

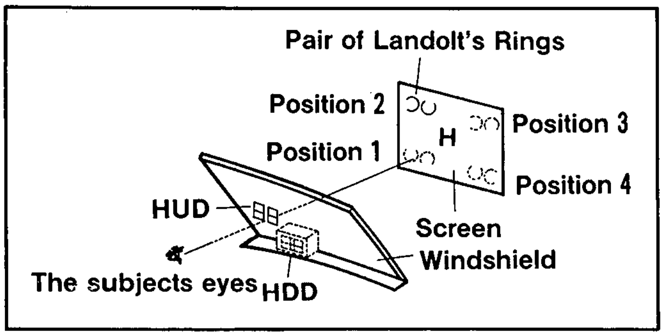

Figure 3. Setup of experiment for examining the difference between a HUD and HDD at a distance of 0.8 m

Seven subjects were used, all males ranging from 20 to 41 years in age. The subjects were instructed to look at the letter "H" at the center of the screen. Then, when the two sources were projected, they first identified the direction of the openings in the Landolt's rings and, next, read the numbers displayed in succession.

Results. The effect of shifting the visual point on displays recognition was shown in Table 3. The error rate with HUD is 30% less than that with HDD. This difference was statistically examined and confirmed to be significant. It can be said, then, that reduction in the movement of the visual point in the Head-Up displaying improves legibility.

Experiment 2. Effect of image distance on HUD legibility

The effect on display legibility that is produced by accommodation of the eyes was also examined based on the "Double Task Method". Two visual sources were projected in a short time period of 0.48 seconds. One was a pair of Landolt's rings which was projected on the screen, and again was meant to correspond to objects in the forward view. The other was the 2 digits of random numbers corresponding to the display. The setup was the same as Experiment 1, and the experimental conditions are shown in Table 4.

Seven subjects were used, again all males from 20 to 41 years in age. As before, subjects were instructed to look at the letter "H" at the center of the screen. Then, when the two sources were projected, they first identified the direction of the openings in the Landolt's rings and, next, read the numbers displayed in succession.

Results. Fig. 5 shows the result of this experiment. The effect of accommodation from the forward view to the display was determined by comparing HUD recognition error rates at various image distances from 0.8m to 10m. These error rates, as shown in Fig. 3, tend to become smaller as the distance of the image is increased. It can be said that reducing accommodation by displaying at a distance improves legibility.

Experiment 3. Effect of display legibility on object recognition in the forward view.

This experiment was also based on the Double Task Method. This method is used to see how varying the load of one task affects the performance of another. Here, the effect of display legibility differences on the recognition of objects in the forward view was examined.

The setup used was the same as that in Experiment 1 (shown in Fig. 3), experimental conditions are shown in Table 5. The subjects were five males of ages from 20 to 40. They were instructed to first identify the display and then the direction of the ring openings. The effect of display legibility difference on

Table 2. Conditions of experiment 1.

Items	Conditions		Remarks
Objects in the forward view	A pair of Landolt's rings		Ring opening directions: Random, Projected position: One of four position as shown in Fig. 3
Distance to the screen	12m		Regarded as being equal to far point from the charateristics of accommodation and convergence
Screen luminance	13 lux		Road luminance when more than 80% of drivers turn on their head lights
Displays	HUD	HDD	
	2 digits of rand numbers	←	
Image distance	0.8m	←	Corresponds to the conventional display
Numbers' height	1.0° visual angle	←	↑
Display brightness	8 cd/m²	13cd/m²	Average of seven subjects' favorite brightness
Display position	10° to the left an 8° below the subject's straight ahead line of vision	20° below the subject's straigh ahead line of vision	HUD: Practical for in - vehicle installation Legibility loss is negligible against straight - ahead. HDD: Corresponds to the conventional display
Projected period	0.48 sec		
Number of trials	80 times		20 times in each of the four position on the screen

Table 3. Difference in recognition error rate between HUD and HDD.

	HDD	HUD	Difference
Error rate (%)	58.7±11.5	27.8±11.5	30.9

the recognition of objects in the forward view was compared by measuring the error rates in recognizing the ring directions in both the HUD and HDD.

Results. Table 6 compares the error rates in ring direction recognition with a 2.5m HUD and 0.8m HDD. The error rate for the HUD is lower than that for the HDD in both brighter and darker displays. The difference in error rates between the HUD and HDD is statistically significant for both degrees of brightness. However, the difference between brighter and darker in each individual display is not significant.

From the results of Experiments 1 and 2, it is obvious that the legibility of the 2.5m HUD is higher than that of the 0.8m HDD. The result, therefore, shows that improved legibility enables the driver to pay better attention to the forward view, which leads to a reduction in the recognition error rate. It can be said, in other words, that high display legibility aids in forward recognition.

Experiment 4. Comparison of display recognition time.

Display recognition time during actual driving was measured by using an Eye-Mark-Camera (EMC). Recognition time as stated here means the period of time in which the subject's visual point remains on the display, whereas it includes the time required to accomodate and converge, it does not include the response time.

The EMC consisted of three TV cameras and an image controller. One of the cameras was placed on top of the subject's head and recorded the forward

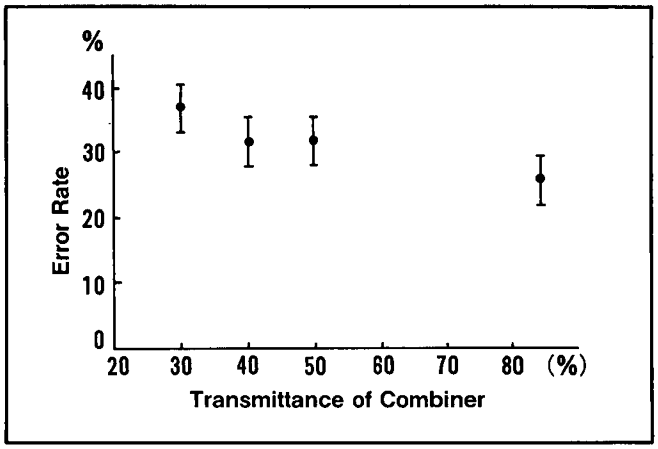

Figure 5. Effect of HUD image distance on display recognition error rate

view and the other two recorded the direction of the subject's left and right eyeballs, respectively. The direction of the eyeballs adjusts to a line of sight at the object viewed so as to place its image in the vicinity of the fovea on the retina. Therefore, by detecting the direction of the eyeball the direction of the line of sight is revealed. The image controller overlapped these three images in their proper relative position and output the subject's visual points in the form of video signals, which were input and recorded in a VTR.

The subjects used were four males of 21 to 38 years in age. Wearing the EMC, the subjects drove the vehicle on the courses listed in Table 7, and while driving, they looked at either the HUD or HDD speedometer in response to oral signals and immediately gave the indicated speed. (See Table 8)

Legibility was evaluated by the time it took the subjects to recognize the speed after their visual points shifted away from the road surface. This time was measured using a digital VTR after the test drives were completed.

Table 4. Conditions of experiment 2.

Items	Conditions	Remarks
Objects in the forward view	A pair of Landolt's rings	See Table 2
Distance to the screen	12m	↑
Screen luminance	13lux	↑
Displays	2 digits of random numbers	
Image distance	0.8m, 2.5m, 5m, 10m	0.8m: Same distance to HDD 10m: regarded as far point
Numbers' height	1.0° visual angle	See Table 2
Display brightness	8 cd/m^2	↑
Displayed position	10° to the left and 8° below the subject's straight - ahead line of vision	↑
Projected period	0.48sec	↑
Number of trials	80times	↑

Table 5. Conditions of experiment 3.

Items	Conditions		Remarks
Objects in the forward view	A pair of Landolt's rings		
Distance to the screen	4.3m		Sufficiently far comparing HUD image distance
Screen luminance	13 lux		See Table 2
Displays	HUD	HDD	
	2 digits of random numbers	←	
Image distance	2.5m	0.8m	HUD: Typical HUD distance HDD: Corresponds to the conventional display
Numbers' height	1.0° visual angle	←	See Table 2
Display brightness	4, 14cd/m²	8, 13cd/m²	Highest and lowest value of seven subjects' favorite brightness
Displayed position	10° to the left and 8° below the subject's straight-ahead line of vision	20° below the subject's straight-ahead line of vision	See Table 2
projected period	0.34 sec		
Number of trials	80 times		↑

Results. Table 9 shows the average of the measured recognition time. In all driving conditions, the HUD demonstrated better recognition time than the HDD, which testifies to the high legibility of HUD.

The recognition time advantage of the HUD over the HDD on straight roads is greater than on curved ones. The reason for this is thought to be that on curved roads the driver's visual point shifts away from the straight-ahead line, which increases the eye movement required in recognizing the HUD. Consequently, the merit which the HUD offers of less movement in the visual point is somewhat diminished on curves. Therefore, the decrease in recognition time difference between the HUD and HDD on curved roads is not due to a reduction in HDD recognition time, but rather to an increase in HUD recognition time.

On a straight road the difference in display recognition time between the HUD and HDD increases along with a rise in average speed. Taking into consideration the tendency that the subject's visual point shifts deeper into the forward view in high speed driving, the results can be understood that HUD makes the recognition time shorter than HDD.

Table 6. Comparison of effect of display legibility on recognition error rate.

Brightness of display (cd/m²)		Brighter HUD: 14 HDD: 27	Darker HUD: 4 HDD: 8
Recognition error rate (%)	HUD	15.2	22.2
	HDD	32.0	37.0
Difference (%)		16.8	14.8

Experiment 5. Effect of combiner tinting

The combiner requires high reflectance to produce sufficient contrast between the display and the external scene. As transmittance would not be made

Table 7. Driving course for experiment 4.

Road	Straight			Curve	
Average speed	40km/h	70km/h	100km/h	40km/h	70km/h
Range of turn radii				30-80R	60-150R

Table 8. Conditions of experiment 4.

Items	Conditions	Remarks
Display item	Digital vehicle speed	
Image distance	HUD:2.5m HDD:0.8m	See Table 5
Visual angle height	1.0degree	See table 2
Luminance of road	2,000-20,000 lux	From cloudy to clear daytime

Table 9. Comparison of recognition time between HUD and HDD.

Road		Straight			Curve	
Average speed		40km/h	70km/h	100km/h	40km/h	70km/h
Recognition time (sec)	HDD	0.47±0.08	0.45±0.02	0.48±0.07	0.48±0.08	0.44±0.02
	HUD	0.34±0.08	0.31±0.02	0.30±0.07	0.37±0.08	0.35±0.02
Difference (sec)		0.13	0.14	0.18	0.11	0.08

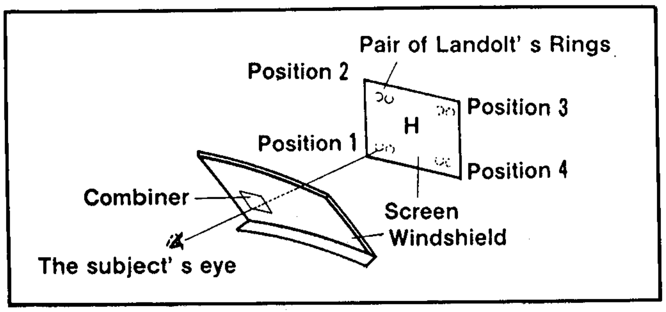

Figure 7. Setup of experiment for determining the effect of combiner tinting on forward vision recognition

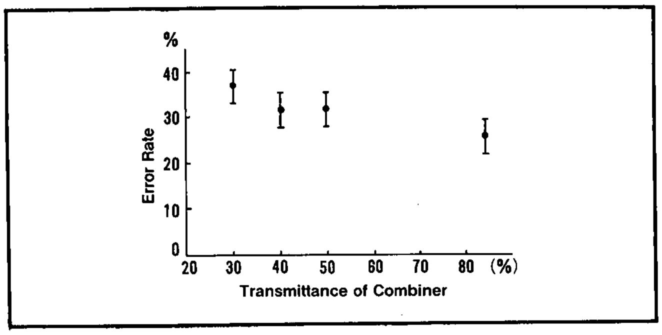

Figure 8. Effect of combiner transmittance on recognition error rate of the forward vision

independent from reflectance, and a combiner with high reflectance would produce lower transmittance than the rest of windshield, it is, therefore, necessary to examine the effect of combiner tinting on forward visibility.

Fig. 7 shows the setup for this experiment. And, the experimental conditions are shown in Table 10. The subjects were six males of 20 to 40 years in age. Here again, they were instructed to look at the letter "H" at the center of the screen and to indicate the direction of the openings when the pair of Landolt's rings was projected.

Results. Fig. 8 shows the change in error rate when the transmittance of the combiner was varied. The error rates for combiners with a transmittance of 50% and 40% do not show significant statistical difference from that of the bare windshield with a transmittance of 84%. However, the combiner with a 30% transmittance demonstrates an error rate significantly higher than that of the bare windshield. Therefore, it can be concluded that combiner tinting need not necessarily impede forward view recognition.

Experiment 6. Optimum HUD brightness considering brightness variation in forward view

Since HUD is superimposed on the forward view, display contrast is greatly affected by brightness variation in the forward view. Therefore, an allowable display brightness range must be determined for each degree of brightness in the forward view. If HUD brightness deviates from this range, a reduction in display legibility, in forward visibility and/or in driver comfort may occur.

Therefore, a sensory test was carried out to determine the allowable brightness ranges for HUD. Fig. 9 shows this experimental setup, and Table 11 does the experimental conditions. The subjects were all males of ages from 20 to 46. They looked at the picture on the screen while taking repeated glances at the HUD. By varying HUD brightness during the process, upper and lower limits of the allowable brightness range were determined.

Results. Fig. 10 shows the results of this experiment. This indicates HUD brightness should range from 2.4 ~ 1,300 cd/m^2 to cover the road brightness variations ranged from 1 cd/m^2 of night driving to 13,000 cd/m^2 of snow condition driving. Since the allowable HUD brightness range for a given degree of forward brightness is narrow compared to that required to cover the gamut of variations in forward brightness, HUD brightness must be more precisely controlled than HDD. (See Fig. 11)

Table 10. Conditions of experiment 5.

Items	Conditions	Remarks
Objects in the forward view	A pair of Landolt's rings	See Table 2
Distance to the screen	4.3m	See Table 5
Screen luminance	13lux	See Table 2
Exposure period	0.27sec	
Combiner size	130×140mm	Requied size assuming of 2 digital HUD and 95percentile eye - ellipse
Combiner position	10°left and 8 °below the subjects' straight - forward line of vision	See Table 2
Combiner transmittance	30,40 and 50% (and a bare windshield of 84%)	
Combiner distance	0.8m	Corresponds to the conventional display
Number of trials	80times	See Table 2

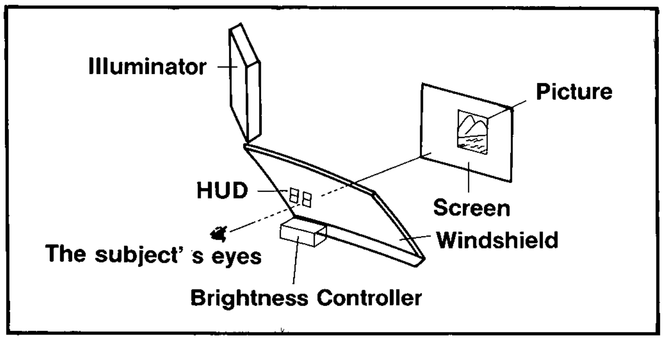

Figure 9. Setup of experiment to examine optimum HUD brightness in respect to various forward brightness

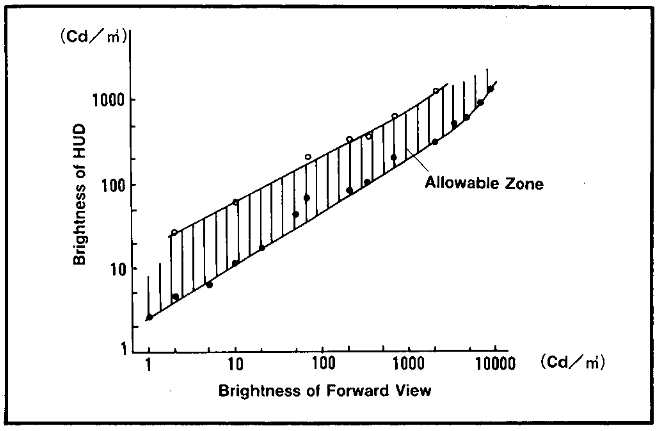

Figure 10. Allowable HUD brightness for various brightness in the forward vision

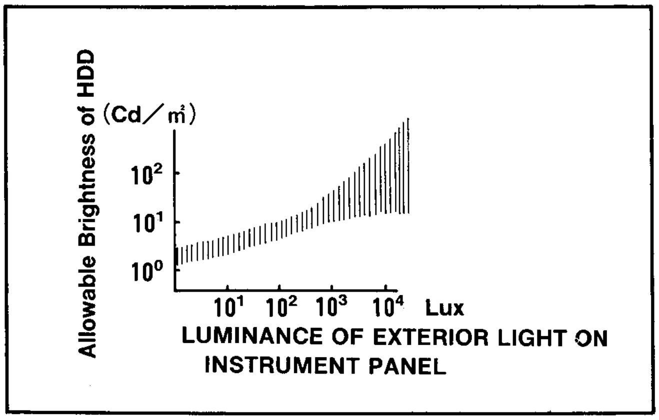

Figure 11. Allowable HDD brightness for various instrument panel luminances

Experiment 7. Sensory test by driving HUD installed vehicle

The feeling which the presence of the HUD on the external scene gave the driver was tested by a question/answer method. The subjects were seven males from 22 to 40 years old. They drove a HUD installed vehicle on a predetermined course at a given speed. During the drive, the subjects scored four evaluation items with five grades in response to tape-recorded questions. The recorded questions were as follows:

(1) Is the display distracting to you?
(2) Does it seem overbearing?
(3) Do you feel any uneasiness due to reduced forward visibility?
(4) Are you troubled by display glittering?

The grades ranged from "1" for no effect to "5" for large effect; "3" represented a level of approval. The image distance of the HUDs were 1m, 5m and 10m, and all tests were carried out in the daytime.

Results. Figs. 12 and 13 show the average grades given in response to the four questions when using a HUD. In all cases, the subjects gave the HUD low (good) grades, which means that HUDs installed in automobiles do not hinder driving ability provided that the HUD brightness is properly adjusted.

In regard to speed, Fig. 12 indicates that uneasiness due to reduced visibility decreases significantly with increase in average speed. Distraction and glittering also tend to decrease with increase in average speed, whereas a sense of overbearingness remains nearly level. Therefore, it can be concluded that HUD legibility is improved with higher speed driving.

Table 11. Condition of experiment 6.

Items	Conditions	Remarks
Objects in the forward view	A picture	
Distance to the screen	4.3m	See Table 5
Screen brightness	1 −13,000cd／m²	From night to snow condition
Displays	2 digits of numbers	
Image distance	2.5m	See Table 5
Numbers' height	1.0° visual angle	See Table 2
Display brightness	Variable	
Displayed position	10° to the left and 8° below the subject's straight - ahead line of vision	↑

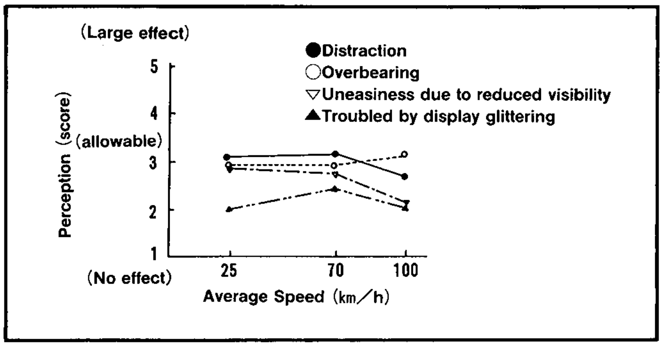

Figure 12. Effect of vehicle speed on perception during drives of HUD installed vehicles

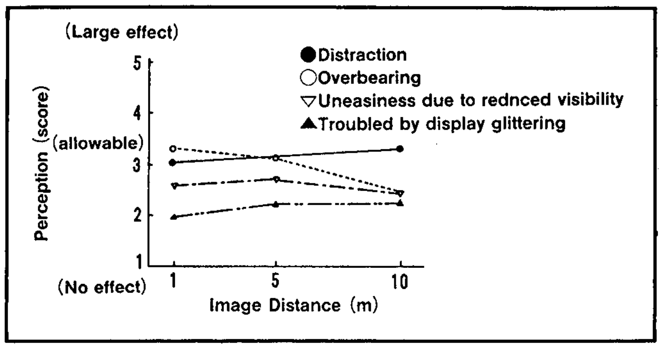

Figure 13. Effect of image distances on perception during drives of HUD installed vehicles

Fig. 13 shows that the sense of overbearingness is significantly reduced with increase in image distance. This reinforces HUD's high legibility effect at longer image distances.

Conclusion

1. The application of HUDs to automobiles can improve both display legibility and forward visibility over that of conventional display panels. As safe driving requires nearly simultaneous recognition of display information and the forward view, HUDs can bring great improvement in driving safety, especially in high-speed driving.
2. Adjusting the brightness of display images is considered very important. Although combiner transmittance is less than that of windshield glass (84%), HUDs do not hinder forward view recognition. In fact, an experiment comparing a combiner with 84% transmittance to one with 40% showed no significant difference in the recognition error of objects in the forward view.
3. By properly adjusting the brightness of display images and properly setting the size and position of display images, HUDs can be guaranteed to never have an adverse effect on driver sensitivities.

Through these experiments and analyses the contribution which HUDs can make to safe driving has been clarified. Further studies into the fields of ergonomics, human factors, electronics are expected to bring about more sophisticated HUD hardware and application techniques.

Acknowledgements

The authors wish to extend their gratitude to Professor Toyohiko Hatada, Tokyo Institute of Polytechnics, for his helpful advice and to the staff of the R & D Department, Central Glass Co. Ltd., for the great assistance they provided us in various kinds of device testing.

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Lotus Active Suspension System

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Abstract

This paper is intended to serve as a brief introduction to the Lotus Active Suspension System and the current status of its development. It starts with an historical and organizational overview. This is followed by a general description of the system and its performance. The fundamental features and attributes are then summarized and finally the steps that are being taken to bring it to the production stage. This generalized suspension system has obvious active safety implications.

Introduction

As reported in the automotive press, the Lotus active suspension has been under development for the past five years. Basically, this is a system which utilizes moderately high pressure electrohydraulic actuators at each wheel location, a variety of sensors on the ''sprung and unsprung'' parts of the vehicle, an on-board computer and specialized Moog servovalves. Mechanical offset springs may support the static weight of the vehicle but the servo system has full authority under dynamic conditions. The system utilizes a modal approach, has been through numerous development stages including some 35 successful prototype vehicles and is currently close to production.

Historical Organizational Overview

Historically, the initial incentive to develop an active suspension came from the high aerodynamic download permitted by the formula for Grand Prix racing cars of a few years ago. Aerodynamic downloads in excess of the static weight of the car were common and had resulted in very stiff conventional suspensions with natural frequencies of several hundred cycles per minute. This essentially rigid suspension caused driver injury and necessitated very smooth track surfaces. In casting about for a solution, Lotus decided to transfer the variable stability aircraft technology as practiced at the Cranfield Flight Research Center. The initial prototype active system, fully analog, was installed on a Lotus Esprit. This system used no mechanical springs to support the static weight. It proved to be extremely effective for ride and also enjoyed a remarkable control over the directional stability characteristics. A Mark II system was then developed for the Lotus 92 Grand Prix car. This was a hybrid system whose algorithm (control strategy) could be rapidly changed. The machine was used to demonstrate the general reliability of the system and to point the direction for further development. The Mark III system was entirely digital with load offset springs. This prototype was constructed on a contract for an American automobile manufacturer. Other prototypes and research vehicles have continued the development. They are fully digital with the exception of certain internal analog loops. In total, more than 35 prototypes have been built for European, American and Japanese firms. Additional prototypes are in work or under negotiation.

A number of significant developments are currently underway: the central processing originally used is being replaced by distributed processing, an economy servovalve and associated patent structure is being developed by Moog, failure mode analysis and experience is under intensive development. Component optimization and productionizing of components is also underway.

Milliken Research Associates (MRA) was retained by Lotus as its USA representative for the promotion of active ride systems and was thus responsible for the introduction of this active control into the USA. MRA was also instrumental in establishing a connection between Lotus and Moog, the latter being the major worldwide producer of servovalves. This led to a technology agreement between these two companies for the furtherance of active ride/handling control. Subsequently, General Motors acquired Lotus but with an understanding that Lotus, while owned by GM, was not a division of GM. As such, Lotus has continued to operate as an engineering development and consultancy firm, performing confidential research and development for automobile companies. This independence of Lotus has permitted the establishment of the joint venture company—Moog-Lotus Systems, Inc.—which is now free to pursue active control development for the motor industry under a completely confidential and proprietary basis. Milliken Research has been retained to assist the new company in further promoting this technology.

Description of Active Control System

Active control is defined as a system to which power has been added. Because of the power level, force and frequency response requirements, the logical implementation is electrohydraulic. A key component in such systems is the servovalve which permits the control of high pressure oil via small electrical signals. The servovalve was invented by Moog in the early 1950's when he was employed by Cornell Aeronautical Laboratory. He was subsequently given the patent rights and has built an international organization performing several hundred million dollars worth of business per year. Moog components and systems are preeminently used in high performance aircraft, missiles, the space shuttle, robots and for many other military and industrial uses.

The Moog valve, by design and internal feedback, is closely a linear flow-control device. Its frequency response characteristic is in the hundreds of cycles per second range and full flow is reached in two-ten milliseconds in typical automotive active systems.

A typical active suspension system utilizes a number of transducers at each wheel and also lateral and longitudinal accelerometers on the "sprung mass" C.G. of the vehicle. These signals are fed into a small software package typically powered by a 600 milliamp battery (and standby) which is completely separate from the vehicle electrical system. The software package contains servo amplifiers, filter networks, signal conditioning circuits, an 8-bit microcomputer and over 100 operational amplifiers.

On the hydraulic side there is an engine driven pump with control system pressure on the order of 2500 psi, accumulators, oil cooler, filters and double acting actuators at each wheel location. Package-wise, the installation at the wheel location is comparable to or smaller than that of a conventional suspension system.

The microcomputer is programmed with the ride and handling algorithms. These algorithms are the result of extensive development by Lotus/Cranfield based on theoretical and experimental considerations. Several hundred algorithms have been explored in the development.

The overall system response is flat to over 20 Hz. Essentially the system enables complete separation of body and wheel motions, i.e., eliminates the design compromise between ride and handling. In essence, the suspension is a fully generalized one, capable of any interconnections, desirable nonlinearities, etc. Although the frequency response is approaching the harshness range, some rubber is still utilized for the control of higher frequency harshness.

Performance

The performance of these active systems is truly remarkable and can only be fully appreciated by a demonstration. Since the body motion is separately controllable from the acceleration inputs, it is possible to call for no roll or pitch under any maneuver. Specifically, the vehicle can be made to roll outward or inward or remain level. Similarly, the antipitch characteristics are fully under control and the vehicle can be given 100% pitch compensation or even have an over-compensation in which the nose of the car rises under braking. Because of the ability to rapidly change the roll and pitch characteristics, a prototype active system is an excellent device for determining what the human driver desires in the way of body motion. Experience to date, indicates drivers universally prefer no roll (thus eliminating the roll transient) and prefer a small amount of dive under heavy braking. With antilock, it seems likely zero pitch would be desirable. With zero roll, camber changes on the wheels are eliminated and this plus the more nearly constant force between the tire and road have resulted in an increase in maximum lateral acceleration on the order 15%. It is perfectly possible to adjust the over-understeer properties by a control accessable to the driver en route. With a nonrolling vehicle, full wheel travel is retained in even the most severe turn, which markedly improves rough road cornering. Maintaining zero roll and eliminating the roll mode, results in improved directional response characteristics and improved controllability throughout the speed range. The roll mode appears as a useless and extraneous form of motion since experience with active vehicles demonstrates that the rolling motion does not contribute to skid warning as heretofore assumed.

Crash avoidance and active safety are improved by an order of magnitude due to:

- Increase in maximum lateral acceleration
- Elimination of extraneous vehicle ride motions due to rough roads and maneuvering
- Elimination of coupling between ride and handling, hence control simplification
- Optimized directional control response and handling characteristics
- Improved control and directional damping at the limit

Large discrete bumps and road waves are completely "swallowed" by the active suspension provided their amplitude is not in excess of maximum suspension travel. The active suspension lends itself to large travel and progressive nonlinear approach to maximum travel. The behavior of the active suspension on rough roads is one of its more remarkable features. Subjectively, the ride improvement is very great and corresponds to objective measurement.

The system is able to distinguish between an aiding load on the wheel or the converse. This feature enables an economy of power. Extensive use can be made of electronic system technology, as for example, in the application of filters to control various vibrational modes. The filters increase the order of the system and a 20th order system is common and practical. Tire/ground contact forces are made much more constant by effectively reducing the "unsprung mass" of the road wheels. Automatic height and attitude leveling are, of course, available. The mechanical design of the suspension is greatly simplified since geometric compromises are no longer necessary. The wheels, for example, can be supported from simple trailing arms with improved packaging.

One of the most dramatic advantages of the active system is the reduction in testing and development time for satisfactory ride and handling. This was well and initially illustrated on the race car application in which the vehicle could be "setup" in the space of a couple of laps, whereas it is not uncommon with a conventional suspension to use an entire practice period. Fast updating of ride and handling requirements in response to market changes is entirely practical. With a good onboard instrumentation system it is possible to explore on a subjective/objective basis, various ride and handling strategies. Experience indicates up-front design costs are more than offset by reduced development time.

Summary of Features and Attributes

Below is a concise summary of the features and attributes of the Lotus active suspension and control system:

1. Addition of Power
   a. Selectivity with load direction
2. Computer Control
   a. Sensed-motion variables input
   b. High-order control strategy
3. Computer Flexibility
   a. Nonlinearities
   b. Manual and automatic adjustments
   c. Algorithm changes (program)
4. Computer Enables:
   a. Feedback control
   b. Electronic system technology(1) Sophisticated filters
5. A Completely Generalized System
   a. Avoids design compromises of conventional suspensions
6. Can be implemented with mechanical springs for static load or no springs. Full servo authority in both cases.

Some Attributes of Active Suspension

1. Body and Wheel Control Independent
   a. Elimination of roll mode
   b. Separation of ride and handling: individual optimization
2. More Constant Tire/Ground Contact Forces
   a. Improve response and damping
   b. Enhanced ride and road holding
3. Body Attitude Control to Accelerations
   a. Large effective wheel travel
4. Progressive Nonlinear Approach to Limit or Travel
5. Manual or Automatic Height and Attitude Leveling
6. Vibration Control by Selective Filtering
7. Design and Operational Control of Directional Response and Trim

Development Time and Cost Factors

1. Simplified Mechanical Design; Improved Packaging
2. Standardized Hydraulic Components
3. Testing and Development Time Drastically Reduced
4. Fast Updating as Requirements Change
5. Rapid Subjective/Objective Evaluation
6. Up-Front Design Costs Offset by Above Items

Steps Toward Production Systems

Moog-Lotus Systems, Inc., has been formed to facilitate:

- Further development of experimental systems (Lotus)
- Further economy servovalve development (Moog)
- Further computer/electronic/algorithm development (Lotus/Cranfield)
- Production components (Moog)

  Servovalves
  Hydraulics
  Electronics

- Systems integration technique (Moog/Lotus/Cranfield)
- Field servicing

Thus the plan is to offer complete active capability—design, development, production and service—at automotive cost levels.

Future Developments

Moog-Lotus believe that "full active control of all vehicle dynamics" is inevitable in the future. This will include:

- Active suspension
- Active four-wheel steer
- Active traction and braking

Adapting Radar to the Automotive Environment

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Abstract

Recent developments in short range radar technology should soon make vehicle radar a reality. A review will be given of the scheme and approach used for radars operating in a complex environment. The difficulties encountered in developing vehicle radar and the headway control logic that provides a safe driving margin will also be covered. Operation of a sensitive radar in a high-clutter environment will be highlighted, as well as the techniques involved in making it work. The methods used for radar accident prevention and the impact automotive radar will have on drivers and society will be discussed.

Introduction

Our goal has been to develop a small vehicle radar system that would actively increase driving safety. Highway safety can be improved electronically by increasing drivers' awareness and by helping them to maintain safe driving margins. Radar will help drivers mitigate and prevent many avoidable accidents.

To achieve a high level of performance, many radar and vehicle factors had to be considered. The level of driving hazard had to be established, and the critical moment it became dangerous had to be determined. A systems approach was carefully selected and further developed to achieve our goal. Operating a short-range radar in a complex vehicular environment required new technology. Many techniques and unique designs were needed to solve the problems associated with operating a vehicle radar system.

This radar system was implemented, road tested, and developed to make it a practical reality. Drivers with radar-equipped vehicles will soon have a new instrument to increase their safety, and that gives them a better chance of avoiding accidents. This new collision avoidance technology will save many lives and reduce injuries. In doing so it will generate a real savings for drivers, insurance companies, and society.

Recent Developments in Automotive Radar

New Radar Technology was Needed for Vehicles

New technology had to be developed for vehicle radar that covers many disciplines—from the small, shaped-beam antenna that transmits 3 mw of power to the specialized signal conditioning circuits, microcontroller and automatic radar braking system. A new approach was taken, and new parameters had to be established. The vehicle radar system needs a sensitivity greater than -100 DBC to work well, even though the radar environment limits the useful sensitivity to a much lower level. The dynamic and linear operating range of the system had to be over 75 DB to handle the radar signals accurately. Such high gain and sensitivity were necessary even though they created additional problems and the need for many new circuits to handle the signals in the radar returns. All the details of this new technology cannot be covered here, but the following should provide a better understanding of vehicle radar technology.

The design for vehicle radar starts with a good antenna and microwave system that can detect small, complex targets. The antenna beam has to be sized, shaped and steered appropriately to cover a vehicle's changing headway. The detected signal is usually distorted as a result of complex multitarget returns. The signal, however, contains all the needed information such as closing rate, range, direction, signal coding, interference, and other data needed by the radar system.

Assuming that we have all the radar data, it must be conditioned and processed to determine how many feet (or seconds) a driver needs for accident prevention. Then the radar's external information is combined with the vehicle's internal data components (vehicle speed, steering, acceleration, stopping, etc.) plus the driver inputs to have all the inputs to make the radar system operational. We refer to all the internal vehicle and driver inputs as driving monitors or driving modifications (DMs), which describes their function. If a radar system were in free space looking at a target ten miles away, no system modification would be necessary. However, in the ever-changing vehicle radar environment, where small and large objects are constantly passing nearby at various speeds, a different solution was needed and developed. The vehicle radar system design had to be adapted to the environment as well as to the vehicle usage.

Once the radar, vehicle and driver inputs are available, they are conditioned and then combined in a digital-analog processor. After this data is in the microprocessor, the system can be programmed to fit the vehicle's driving situation to achieve the appropriate outputs for the driver.

The main output of the vehicle radar system is to alert the driver—who is the vehicle's ultimate safety device. The radar warning system is designed to keep the driver safer and to actively warn him in dangerous situations. Radar braking system (RBS) technology was developed to further demonstrate the radar's capability to logically stop the vehicle. This new radar technology and developments should soon make collision avoidance radar a production reality.

Vehicle Radar Performance Must be Effective

An effective vehicle radar system should benefit the driver in several different ways. However, regardless of how good a radar system is, it has limitations. Accidents will still occur in both controllable and uncontrollable situations. Radar-equipped vehicles will be taking on a very complex problem, and in spite of the fact that it may greatly improve safety, the radar will still be blamed, in many cases, for not being adequate. Regardless of these criticisms, the overall safety benefits anticipated for radar collision avoidance may be as high as 40 percent and this alone will justify production. This high percentage may be achievable simply because distracted drivers are the major cause of accidents, and they obviously need an active system to make them safer drivers.

The ideal vehicle radar system would always alert the driver in time to avoid a collision. However, in real-world traffic situations, this is not always possible. Curving roadways, cross traffic, and many other fast-changing situations make it physically impossible in many situations for the radar to always give the driver the time and distance needed for stopping. But alerting the driver just one second (e.g. 73 feet at 50 MPH) before impact can often provide enough time to steer around an obstacle, or at least allow some additional braking time. The system can provide seconds in many situations that may give drivers the extra time they need.

Some of the safety benefits a vehicle radar system can provide are:

Radar will be constantly on guard to act as a safety back-up for distracted drivers.

Radar should prevent many sideswipe accidents, especially when the road is straight or turns left.

Radar can help drivers establish and maintain safer driving margins in traffic situations.

Radar can respond very quickly in accident situations to alert drivers.

Radar, to be truly effective, must perform well enough to gain the driver's confidence. To be most effective and meaningful for a driver, the system should communicate in the same manner as a passenger would warn an unsafe driver. A passenger knows when a driver is unsafe and he will respond accordingly. If a driver is careless, a passenger may say something; if a collision is imminent, he may yell. The radar warning system should also communicate its measured level or interpretation of the hazardous situation and alert the driver in words and statements. The system can now say "look out," "be careful," and alert the driver in appropriate situations.

The primary control for vehicle safety will always be in the hands of the driver. However, when a driver is not paying attention, the radar will be working for him in many critical situations. This means it should not miss too many opportunities to avoid preventable accidents; yet it must not disturb the driver or passenger needlessly. Therefore, the output of the radar warning system should be focused toward keeping the driver attentive in hazardous and dangerous situations and alerting him to take action. Collision avoidance radar now looks very promising because of what it can do for drivers.

The Difficult Environment for Vehicle Radar

The Vehicle Radar Environment

The complex, moving environment the vehicle radar sees, and must operate in, poses many difficulties for reliable and optimum system performance. With a small high-gain antenna mounted two feet above the roadway, many nearby objects must and will be seen by the radar system. Radar energy bounces around when it hits nearby objects, much like water from a nozzle would bounce back from a nearby object. A sensitive radar will see the road, cars, posts and any other nearby and distant objects in the radar beam. Some small, poorly shaped radar objects (referred to as targets) are hard to see. Other targets are very large, and some large targets can be seen miles away. The necessary dynamic operating range of the system must, therefore, be great to see both small and large targets. For this reason, many ships switch to low-power radar systems when they are coming into harbor. Many vehicle radar systems can only see or respond to larger targets at close range. A good short-range radar system must detect smaller targets and also work well when large targets are blindingly close. A radar can blind itself by its own return.

All of these environmental difficulties are manifested one way or another as signal problems such as distortion, saturation, multipath, multitarget, multibounce, and multiple returns. Other related problems are false alarms, radar blinding, electrical interference, road noises, vehicle noise, bounce and vibration, weather, and similar phenomena.

These environmental problems are some of the primary reasons why vehicle radar has been such a challenge to develop. Regardless of environmental restrictions, it is very important for a system to recover all the information available in the radar signal. The difficulties of extracting the information from a weak, saturated or distorted return signal have caused vehicle radar designers to look for alternate solutions.

It was for this and other reasons that a doppler system was selected over the pulse and frequency modulation approaches. The doppler signal provides more direct information than other radar approaches tried, and it has the primary component of closing rate. New circuits and designs were developed for the vehicle radar system to help resolve the environmental radar problems.

Roadway Environment Should Improve

The road performance of the vehicle radar antenna and the circuits that back it up must be very good in order to see smaller radar targets. The major factor in any vehicle radar system will be how well the radar sees and recognizes the weaker targets. Some objects, at times, appear almost invisible to the radar. The normal signal strength range or radar cross-section of these radar returns now varies about 50 DB. Some poor radar targets can be improved with ease, but detecting a small child will always be a challenge for any radar system.

Roadways are now enhanced with all types of visual aids to help the driver's vision. Radar visibility can also be enhanced by adding a radar reflecting device or material to existing vehicles and roadside markers. Visible light and radar images are comparable in many ways. Radar returns respond somewhat like looking at yourself in a mirror. A slight turn of the mirror and your image disappears, even though you can still see the mirror. Radar returns don't suddenly disappear like the visual image, but they do become very weak in many cases.

As vehicle radar becomes a reality, vehicles may have improved radar cross-sections, or they may be augmented with one or more radar reflectors to enhance their radar image. It takes a very small change to make poor radar targets many times stronger, especially at higher frequencies. A radar system should still be able to perform in the present automotive environment. However, when the roadway environment is improved for radar, it will be better for drivers as well.

The Headway Control Scheme

The Headway Control Logic

In simple terms, the headway control zone represents the space or distance a driver needs in front of his vehicle to stop it or regain control. Since vehicles are always turning in various ways as they are driven through both a moving and stationary environment, this complex motion must be taken into account in all radar driving situations.

One of the keys to collision avoidance is a scheme or algorithm that can always distinguish between a safe or dangerous driving situation. The degree of safety or danger can be thought of as a measure of hazard level (HZL). The critical hazard level is determined by the total value of all the major factors in the algorithm. Additionally, the dynamic algorithm allows us to differentiate between many different hazard *levels*. With the control zone scheme, the radar system can determine any hazard level within the control zone, while disregarding radar targets outside of the zone.

The major components used in the control zone logic are:

Range (R). This is a measurement of the distance from the radar to the object it sees, referred to as the target. Range, in feet, is weighted at about 50% of the value of the algorithm. (Line R, Figure 1)

Closing Rate (CR). The closing rate is the relative speed difference between you and the target, which may be moving or stationary. The closing rate, in MPH, is weighted at about 33% of the value of the algorithm. (Line CR, Figure 1)

Vehicle Speed (VS). The vehicle speed is how fast the vehicle is moving on the road. Vehicle speed, in MPH, is weighted at about 17% of the value of the algorithm. (Line VS, Figure 1)

Driving Modifiers (DMs). These modifying factors have to do with the operation of the vehicle, and with what the driver is doing in traffic. Since the radar environment is constantly changing, the DMs are specifically applied to adapt the radar's control zone logic to fit these vehicle driving situations. Specifically, the DMs include factors such as turning as a function of vehicle speed, slow closing rates and vehicle speeds, the status of the brake pedal (whether the driver is braking or not), and target direction. These inputs, along with other processor-generated modifiers, are used to fine tune the algorithm in order to yield the appropriate output for any driving scenario. All the DMs, as weighted, may range from +15% to −30% in the algorithm. When the vehicle is going straight in a normal situation, the DM value is usually zero.

The Headway Control Algorithm. When the ordinate value units of the range R, closing rate CR and vehicle speed VS are chosen separately and then added together, the total will yield the value of the hazard level. (Please refer to Figure 1, page 7.) Hazard levels are considered safe under 400 units and dangerous above 400 units. (The DM value has been omitted for clarity.) The formula for the HZL is: $HZL = R + CR + VS \pm DM$. For working examples let $DM = 0$.

The Dynamic Safety Margin

A safer driving margin can be maintained if several key factors are controlled. To establish an appropriate safety margin, we must, in this case, weigh and combine two of the major inputs to yield the desired safety or range margin. (Refer to Table 1, page 7.) The dynamic safety margin table shows vehicle speed (VS) along the left vertical edge and closing rate (CR) along the top. At the intersection point of any VS and CR, in the middle of Table 1, the safe distance (R) that should be maintained for a given set of conditions is shown. For example, a situation where VS and CR are both 40 MPH (going 40 MPH into a stationary object) yields a range R on the diagonal line of 171 feet. The table of margins can be used for any speed up to 70 MPH and any range up to 300 feet, and for anything from tailgating to high closing rate situations. The diagonal Line R represents the stopping distance needed if you are going toward a stationary object, such as a parked car. If the other car is moving in the same direction as you are, and at the same speed, the safe following distance is shown toward the left edge. If you are going slow and the other vehicle is rapidly approaching you, the warning range will be shown in the upper right of the table. In addition, the needed or desired range values can be scaled up or down as much as 100 feet for different vehicle and driving situations.

The Technology in Vehicle Radar

The Radar System

The radar system is divided into two major assemblies: the antenna assembly in the grille of the vehicle and the driver's console located in the dash.

The antenna assembly is the eye for the radar system and must always have a good view of the road ahead. The radar antenna beam is narrow and is shaped somewhat like a pencil. The antenna assembly, which can be either fixed or steerable, is approximately the size of a vehicle headlight assembly. It must be rugged and weatherproof since it will be mounted in the vehicle's grille. Several different

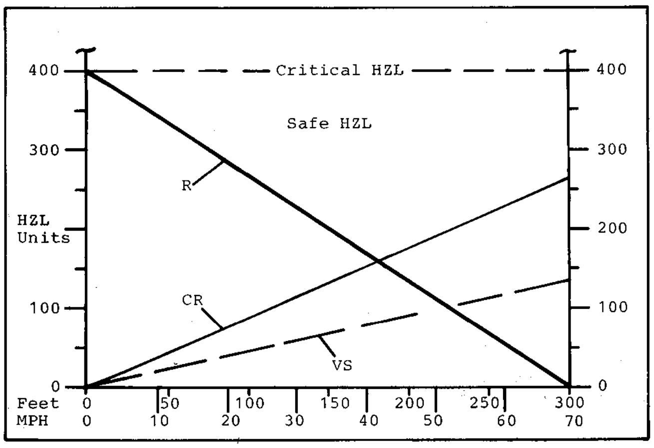

Figure 1. Headway control algorithm

Table 1. Dynamic safety margin.

VS Vehicle Speed in MPH	CR Closing Rate in MPH: 0	1	10	20	30	40	50	60	70
	1	2	29	57	86	114	143	171	200
	10	14	43	71	100	128	157	185	214
	20	29	58	86	115	143	172	200	229
	30	43	72	100	129	157	186	214	243
	40	57	86	114	143	171	200	228	257
	50	71	100	128	157	185	214	R	271
	60	86	115	143	172	200	229	257	286
	70	100	129	157	186	214	143	271	300

antenna models will be manufactured for styling and performance reasons. We currently use a shaped-beam reflector antenna with an on-axis feed.

The active microwave components in the antenna assembly are the low-power Gunn transmitter diode and a mixer diode. Both are connected to a waveguide circuit mounted on the back of the radar antenna. The antenna and microwave assembly are shock mounted to increase system performance and sensitivity. The antenna is steerable right or left 20 degrees by a servo positioner that appropriately maintains a constant cord height in turns in order to look at a spot two seconds ahead of the vehicle. A single cable connects the antenna assembly to the driver's console.

The driver's console will contain all the integrated electronic circuits for the entire radar system. The best and most reliable technology and techniques must be used in a vehicle radar system. The electronics for the radar system will include custom integrated circuits, such as ASIC's and PLA's, along with other high quality discrete devices. The console will be approximately 2 x 6 x 9 inches deep. It will be located to allow the driver easy access to see and adjust the radar system. The console controls and switches will give the driver the ability to adjust the system to fit his driving preferences and needs. The radar's performance and operating characteristics can be adjusted over a very broad range, and several indicator lights will keep the driver informed about the radar's status. The radar's console will be "user friendly" so the driver can easily select different safety margins with a single switch. The console will have a speaker for voice and audio warnings. (The car radio speaker could also be used for this purpose.)

All radar system data will be processed by a microprocessor that will control the system's output commands. One advantage a radar system will have over the driver is the speed and quantity of data it can accurately process, and the ability to convert it into useful output commands. The radar console may become the "black box" for vehicle accidents because it will have all the radar and vehicle data in non-volatile memory. All external inputs and internal vehicle data are scanned 50 times a second, then processed, stored and used by the system. The system will also continually test itself for operational failures and alert the driver if necessary. The radar, with a microprocessor that can be programmed to fit various driver preferences, will be a dynamic instrument that will extend a driver's capability. We have many patents pending covering the technology used in our radar system. Vehicle radar required many design innovations to overcome all the problems and to make it work well, especially at low power.

A good practical test for a vehicle radar system can be conducted on any roadway or city street. Traffic congestion is an ideal situation for a radar system because the target vehicle is close and directly ahead. Standard cars can be seen beyond 300 feet and even a poor target like the beetle-shaped Volkswagen can be seen beyond 200 feet in many situations. A vehicle radar system may provide drivers the extra margin of safety they need to make safer driving possible.

Figure 2 shows a functional block diagram of the vehicle radar system. The radar signal processor can be either analog-digital or microprocessor based. A host computer can be connected through an RS-232 serial port to communicate directly with the radar's microprocessor. For details about the radar system's specifications, please refer to Table 2.

Collision Avoidance for Drivers and Society

The Impact of Active Prevention

Automakers are doing many things to make vehicles safer. Seatbelts, airbags, and improved braking systems are now available. Governmental agencies and insurance companies are doing everything possible to improve driver awareness. Today's cars and highways are much better than 40 years ago, but speeds are higher and roadways are more complex. Driving has been made easier and more convenient, but something more will still be needed as long as accidents occur. Although passive devices can reduce injuries, this approach will not prevent accidents or reduce their number. An active collision avoidance system in vehicles could prevent or mitigate accidents. A collision avoidance system could easily pay for itself if it prevents just one accident.

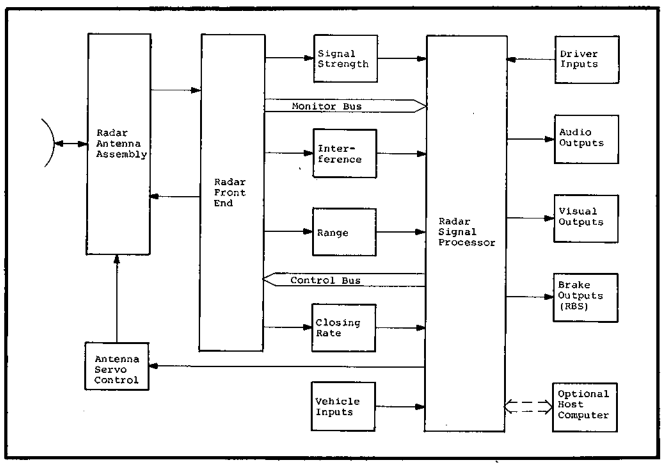

Figure 2. Vehicle radar system functional block diagram

Table 2. Summary of vehicle radar specifications.

OPERATING FREQUENCY	24.15 GHz
TRANSMITTED R.F. POWER	3 mw typical
MODULATION BANDWIDTH	1 MHz
SYSTEM SENSITIVITY	-85 DBC
ANTENNA	
Beamwidth	5° @-3 DB
Sidelobes	-22 DBI minimum
Size	7" dia.
Steering	20° left/right
OPERATING RANGE	1-300 feet
AMBIGUOUS RANGE	6000 feet
CLOSING RATE CAPABILITY	.1-200 MPH
VEHICLE INPUT	
Speed	1-100 MPH
Tire angle	35° left/right
Braking	off/on 12VDC
DRIVING MODIFIERS	8 typical
DRIVER INPUT	
Programs	4
Sensitivity levels	2
Driver selection	3
SYSTEM PROCESSOR	
Microprocessor	8088
System Clock	5 MHz
Cycle Rate	7 MS
ROM	8 K
RAM	32 K
POWER INPUTS	16 watts typical
COMMUNICATION PORT	RS 232, 9600 bps
SYSTEM RESPONSE TIME	100 MS typical
SYSTEM OUTPUTS	
Audio Statments	9
Visual Indicators	3
Braking (RBS)	proportional

Apparently, what the driver needs now is help—help to prevent accidents before they happen. Collision avoidance is active prevention, which means it will do some or all of its work before the collision. With collision avoidance to alert the driver, he will have a better chance to lessen the severity of accidents. This lifesaving radar instrument will save lives and be unduly criticized if it doesn't. Drivers may rely on their radar for protection in many unrealistic situations. Vehicle radar systems should be able to establish their own worth and safety limitations within one or two years. Regardless of the criticisms, radar should prove to be a very substantial safety improvement.

The vehicle radar system is also perceived as an instrument to help drivers maintain better control of their driving safety. If drivers paid attention and drove more carefully at all times, collision avoidance equipment wouldn't be as effective. Drivers will never be able to pay attention at all times—even when they should. The vehicle radar could be the catalyst that makes driving, and drivers, safer. Collision avoidance technology could make a big difference because it will help the driver directly.

There are now good reasons to believe that the immediate benefits of vehicle radar will make it a reality. It can be demonstrated that a vehicle radar system may be able to reduce the consequences of accidents by 50%. This would double driving safety, and thereby result in significant savings to insurance companies and drivers. Insurance companies could reduce insurance premiums 25% and still maintain their level of earnings.

We predict that once radar systems are mass produced, they should cost between $400 and $700 each, depending on model and driver options. Initially, the consumer cost of a radar system in a new luxury vehicle is estimated to be higher at about $700 to $900. Let us assume that in 1989 a new luxury car will cost $24,000 and the vehicle radar will cost about $800 extra. Also assume the owner keeps his vehicle for four years and then resells it. The owner's yearly insurance premium of $800 may be reduced by approximately 25%, thereby resulting in a savings of $200 every year, or $800 over a four-year period. This $800 insurance savings will have completely paid for the radar system. We also assume that the radar system will be worth at least $400 after four years, which means the owner will be $400 ahead over the four-year period.

During this same four-year period, the insurance company could save an additional 15% or $120 per year, or a total of $480 over four years. Based on this example, the radar owner would save $400 and the insurance company $480 over four years. That is a possible $880 savings for the driver and insurance company in four years, and the radar system should last 10 to 15 years.

These assumptions may be conservative for several reasons. The cost of a new radar system may quickly drop to $600. It is believed that radar systems in used vehicles will command a premium of perhaps 80% to 90% of their new cost. The vehicle radar would, therefore, have a greater value to the second owner because of its accident reduction capability and lower insurance premiums. If one typical accident can be avoided over a five-year span, the radar's cost would be recovered many times over. The net savings generated by a vehicle equipped with radar could be significant every year, regardless of its age.

There may be other important benefits for society that may result from the use of radar-equipped vehicles. These benefits could include less traffic congestion because of fewer accidents, and a reduction in the number of additional lanes needed during peak hours. Computer-controlled interval spacing technology is already in the radar system. The resulting roadway efficiency, and greater overall driving safety, could make radar a real value to society. Radar-equipped vehicles can give drivers the active help they need to prevent many avoidable accidents. The vehicle radar system could make a difference because it can focus the driver's attention to keep him in touch with his driving. People will have greater peace of mind knowing that their radar-equipped vehicles make them safer drivers.

Objective Directional Response Testing (Written only paper)

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Abstract

This paper summarizes the technology and methodology used at General Motors for the objective measurement of vehicle directional response characteristics. It serves to update previous publications regarding this subject and presents information on new test procedures that have not previously been published. Trends noted from statistical analysis of production vehicle performance data are also discussed.

Introduction

Vehicle directional response characteristics, driver skill, and road/environmental influences are the three primary factors that comprise the active safety performance of motor vehicles. These factors can be viewed as forming a simple closed-loop model of vehicle operation (Figure 1), where the vehicle interacts with the road surface and environment, and the driver closes the loop through application of inputs to the vehicle, based on cues obtained from both the vehicle and the road system. Of these three factors, only vehicle directional response characteristics are quantifiable with any degree of precision. Various test programs using driver subjects have shown a wide variety of driver skill levels, reaction times, and strength. The driving environment is even more variable, with a wide variety of pavement friction levels, surface roughnesses, and weather conditions.

The interaction of these stochastic influences frustrates attempts to quantify directional response characteristics using closed-loop test methods. Determination of vehicle performance characteristics during closed-loop operation is only possible if large amounts of data are taken in order to normalize the random influences of the driver and road system. This is an unsatisfactory solution outside the research environment. During the development process of new vehicle designs, limited test time is available for assessing the directional response characteristics of prototype vehicles, since these vehicles are in high demand.

A solution is the use of open-loop directional response test procedures, where the driver and road influences are minimized through the use of carefully prescribed and repeatable driver inputs, with testing conducted on well defined road surfaces, under limited environmental conditions. This situation is depicted in Figure 2.

Vehicle dynamics investigations at General Motors date from the 1930's and originated with the work of Maurice Olley, one of the earliest vehicle dynamicists. The procedures and techniques described in the first half of this paper represent an overview of the current stage in the development of objective, instrumented test procedures for the quantification and measurement of vehicle directional response characteristics at General Motors.

In addition to assessing the directional response performance of new vehicle designs throughout the development process, tests are conducted to measure the directional response characteristics of both GM and competitive production vehicles. The second half of this paper will present information obtained from tests of a variety of production vehicles.

Objective Directional Response Test Procedures

Background

Although research into the quantification of vehicle directional response characteristics dates from the early part of the century, most developments have taken place in the past 30 years. Many of these developments have been made by four groups: U.S. Government sponsored research programs, activities of the Society of Automotive Engineers (SAE), activities of the International Organization for Standardization (ISO), and internal activities of various automakers. The outputs of each of these groups, and

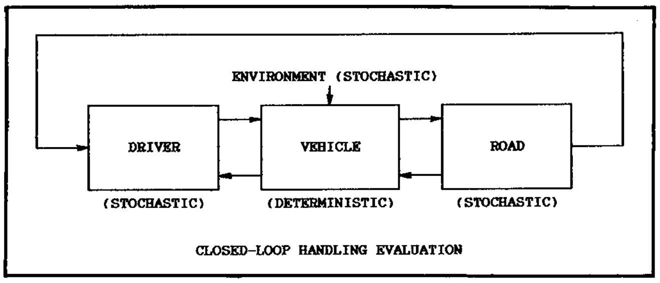

Figure 1

their applicability to the design process, are discussed in the following paragraphs.

Substantial government-sponsored research in vehicle dynamics took place in the late 1960's and throughout the 1970's. This research was sponsored by the National Highway Traffic Safety Administration (NHTSA) with the primary purpose of improved vehicle safety through enhanced vehicle handling performance, mandated by government regulation. This research resulted in the creation of a variety of test procedures and an array of performance metrics, most directed at the limit performance aspects of vehicle directional response. Some of the procedures produced by these studies are the tests and performance goals proposed during the Experimental Safety Vehicle (ESV) and Research Safety Vehicle (RSV) programs. Another set of test procedures proposed by the NHTSA are the eight Vehicle Handling Test Procedures (VHTP's). An overview of these procedures is given in Reference 1. These procedures are again directed at limit performance and were aimed at establishing measures of vehicle limit handling that could be used as the basis for statutory regulation. The performance metrics chosen for these procedures have limited applicability in the general vehicle design process. These procedures have the additional disadvantage of being relatively complex and cumbersome, with substantial requirements in both test equipment and test time for completion. As a group, the government test procedures have limited applicability in the vehicle design process.

The Society of Automotive Engineers Vehicle Dynamics Committee has developed a set of four vehicle dynamics test procedures with the goal of standardizing techniques used for performance measurement throughout the industry. These four procedures are

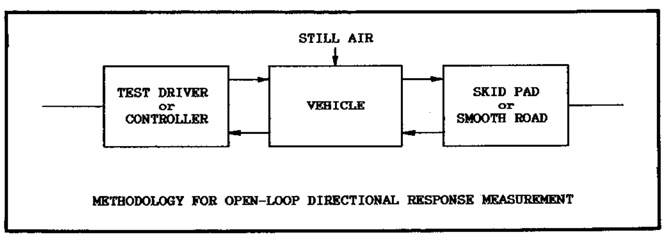

Figure 2

currently in draft form(2) and have been validated by a number of manufacturers and independent organizations as providing representative data. The proposed test procedures are the constant radius method, the constant steering wheel angle method, the constant speed method, and the constant throttle method. These procedures all measure steady state directional response properties of a vehicle and employ similar performance metrics. The last of the procedures measures transient response properties as well and is virtually identical to one of the standard test procedures used by General Motors (the Control Response Test, discussed below).

The International Organization for Standardization (ISO) Vehicle Dynamics and Road Holding Ability Subcommittee (TC 22/SC 9) has also developed a series of vehicle dynamics tests. These tests are generally directed at methods for measuring vehicle responses on test facilities available in Europe, where large skid pads are less available than they are in the United States. One of the procedures (ISO 4138) is an accepted International Standard(3) and is similar to the constant radius test method contained in SAE XJ266. The remaining procedures (ISO/DP 7401), relating to the measurement of transient directional response characteristics, are in draft form(4). The procedures are basically conducted on straight roads and emphasize the evaluation of performance in the frequency domain. An exception is the step input test, which is similar to the SAE constant throttle test and the control response test used by GM.

The last major source of technology for the quantification of directional response characteristics is the manufacturers themselves. The procedures and facilities used by General Motors are described in the following paragraphs.

Data Acquisition Equipment

The testing needs of a vehicle manufacturer are different than those of a research organization. The latter can afford the time to design and fabricate special transducers to suit the particular requirements of a specific vehicle and can carefully instrument a vehicle with numerous individual transducers. In addition to instrumentation time, test turnaround time is also a minor constraint in the research environment, with testing of a vehicle taking days or weeks to complete.

For the vehicle manufacturer, overall test time for objective performance measurement must be minimized due to the high demand for prototype vehicles. This need to minimize test time has led to the development of specialized test and data acquisition equipment. This equipment has evolved into compact, modularized packages that can be quickly installed in a vehicle.

For most testing, four equipment modules are used: a data acquisition system, a transducer and signal conditioning module, an instrumented steering wheel, and a velocity transducer. This equipment is general purpose and can be applied to a variety of tests. With the modular equipment, a vehicle can be fully instrumented and ready for test in less than one hour.

The data acquisition system which has been developed consists of a commercially available IBM PC compatible microcomputer, equipped with additional commercially available cards to configure it for use as a data acquisition device. A custom 12 volt DC power supply card completes the package, enabling the system to be operated from a standard vehicle electrical system. The system has the following specifications: 702 kB RAM, two 360 kB floppy disk drives, 512 kB non-volatile bubble memory, and the capability of monitoring up to 16 analog data channels, with analog-to-digital signal conversion. Digital sampling rates of 40 samples per second or less are used for all tests, as this sampling rate has been determined to provide the desired accuracy without the need to take (and reduce) excessive amounts of data.

The transducer and signal conditioning module contains most transducers used in standard tests: a free gyroscope, used for measuring vehicle roll angle; a yaw rate gyroscope; a lateral accelerometer; and a fore-aft accelerometer. The package also contains all necessary power supplies and signal conditioning to support these transducers plus other standard transducers (steering wheel angle, steering wheel torque, and vehicle velocity). The module does not use a stabilized platform for the accelerometers, as stabilized platforms have been found to require excessive maintenance when used in routine testing. Errors resulting from vehicle roll and pitch motions are corrected during data processing.

Two different instrumented steering wheel modules are available for use in different types of testing. Both of these modules require the removal of the production steering wheel. One steering wheel has the capability of measuring steering wheel angle and incorporates adjustable steering stops to enable repeatable step steer inputs to be made. The other steering wheel measures both steering wheel angle and steering torque.

The velocity transducer module currently employed is a standard fifth wheel, which has been found to be accurate and durable in the production test environment.

The complete instrumentation package which has been developed weighs approximately 60 kg and has an operating current draw of approximately 10 A.

General Motors Tests

As stated above, the test needs of a manufacturer are different from those of a research organization.

Directional response tests used by a manufacturer must be relatively routine and not overly complex, so that vehicle testing can be accomplished expeditiously.

Objective directional response tests used by General Motors are intended to quantify the performance of vehicles to permit comparison of the directional response characteristics of new vehicle designs with the characteristics of existing vehicles. The procedures currently in use are directed at the quantification of both limit and sub-limit performance and have found increasing application in the GM design process. The availability of objective test information permits vehicles to be measured quantitatively throughout the design process.

The following objective tests are discussed in this paper:

- Control Response Test
- Frequency Response Test
- Maximum Lateral Acceleration Test
- On-Center Handling Test
- Lift-Dive Test
- Center of Gravity Test

Control Response Test

The control response test is a fixed control test (i.e. - vehicle path is not defined) run at highway speed (100 km/h). This test speed was selected in order to magnify vehicle performance differences as much as possible, consistent with constraints imposed by test facility size and test safety. This test can be run to the limits of vehicle performance in relative safety. The test procedure is virtually identical to those contained in SAE XJ266 (constant throttle test) and ISO/DP 7401 (step steer test), which are both based on the GM procedure. The test consists of a series of step-like steering inputs of increasing magnitude applied from an initial condition of zero steering wheel angle and 100 km/h forward velocity. In each test run, the steer input is held until steady-state turning conditions are established, then quickly removed. The test provides information on steady-state turning performance as well as transient characteristics both entering and exiting a turn.

Because the vehicle path is not defined in this open-loop procedure, a large test surface is required. General Motors is able to run this test year-round, with the 67 acre Vehicle Dynamics Test Area at the Milford Proving Ground in Michigan permitting testing during the spring, summer, and fall months, and a similar, smaller facility located at the Desert Proving Ground in Arizona permitting tests to be conducted during the winter months.

A complete control response test consists of 30 or more test runs, half in each turn direction. Steering wheel angle increments used range from $2\frac{1}{2}$ degrees to 10 degrees. During the test, the following vehicle motion variables are continuously monitored: forward velocity, steering wheel angle, roll angle, yaw velocity, fore-aft acceleration, and lateral acceleration.

The specific metrics evaluated are: understeer gradient, roll gradient, sideslip gradient, steering sensitivity, yaw velocity response time, and lateral acceleration response time. This test is utilized extensively in the development process.

Frequency Response Test

The frequency response test provides gain and phase relationships in the frequency domain describing the dynamic response of a vehicle to steering wheel angular displacement. The test procedure used is similar to that described in the random input method in the ISO/DP 7401 test procedure. The test is conducted at specified constant speeds (typically 100 and 140 km/h) on a straight, three-lane road, with vehicle steer excitation using a broadband pseudo-random steering input. The steer input used is a "swept-sine" input, comprising steer input frequencies from near-DC to the maximum frequency which the test driver is capable of achieving. The swept-sine input has been chosen to insure adequate input power across all steer frequencies.

During the test, steering wheel angle, yaw velocity, lateral acceleration, and roll angle are continuously recorded. The recorded data are processed through a digital spectrum analyzer to define the spectral content of each signal.

The transfer functions describing the yaw velocity, lateral acceleration and roll angle responses to a steering angle excitation are obtained for each test segment and ensemble averaged for the entire test. The test is conducted within a lateral acceleration limit of 0.4 g. The vehicle gains are normalized with steering wheel angle.

Maximum Lateral Acceleration Test

The maximum lateral acceleration test measures the maximum steady-state cornering capability of a vehicle on a constant radius circular course. The test course used is similar to that specified in the constant radius part of the SAE XJ266 test procedure and to that specified in the ISO 4138 test procedure. The purpose of the test is much different than the SAE and ISO tests, however, and concentrates only on the vehicle's limit performance characteristics. The single test metric is maximum lateral acceleration. The test is run by driving the vehicle at the maximum speed attainable on a fixed radius circle, while still remaining on the circle. The test is run under driver control. This is one of the few closed-loop objective tests performed by General Motors, and is somewhat driver-sensitive.

During the test, vehicle velocity and yaw rate are continuously monitored. When these two variables are multiplied, the result is lateral acceleration. The lateral acceleration is averaged for each test run (approximately one lap of the test circle). The turn direction is then reversed, to equalize tire wear, and the test is repeated in the opposite direction. A complete test consists of six to eight test runs, half in each turn direction. The average of all the individual lap values determines the maximum lateral acceleration capability of the test vehicle. This value represents the performance limit of the tire-vehicle-driver combination.

On-Center Handling Test

The on-center handling test has been developed as a means of quantifying control quality in the low lateral acceleration regime typical of expressway type driving. This test was described in a recent SAE paper(5). The test consists of a low frequency (0.2 Hz) sinusoidal steer angle input at a constant vehicle velocity of 100 km/h. The steer input magnitude is limited so that lateral acceleration does not exceed 0.2 g. The following performance variables are monitored throughout the test: forward velocity, steering wheel angle, steering wheel torque, yaw velocity, and lateral acceleration.

A variety of performance metrics are obtained from cross-plots of the measured performance variables. Among these are minimum steering sensitivity, steering sensitivity at 0.1 g, lateral acceleration at zero steering torque, steering torque at 0 g and at 0.1 g, steering torque gradient at 0 g and 0.1 g, and steering work sensitivity. This test has rapidly gained popularity within GM.

Lift/Dive Test

The lift/dive test is intended to quantify suspension movement and vehicle pitch characteristics during acceleration and braking. The test metrics are front and rear lift gradients and vehicle pitch gradient, under both acceleration and braking. The lift/dive test instrumentation consists of string potentiometers on each wheel, measuring the relative vertical displacement between the wheel center and the body, plus an accelerometer mounted inside the vehicle to measure longitudinal acceleration.

The test is performed in two phases—constant accelerations at approximately 0.1 g intervals up to wide open throttle, and constant decelerations at 0.2 g intervals up to impending wheel lock-up. Front trim and rear trim values (averaged for the front and rear axle) are recorded for each acceleration level. Pitch is calculated as the difference in front and rear trim readings, and measured acceleration is then corrected for the pitch angle to determine true longitudinal acceleration.

The measured data are plotted as front trim change, rear trim change, and pitch angle change versus acceleration. The performance of vehicles may be compared by taking the slopes of the fitted lines on these plots.

Center of Gravity Test

The center of gravity test provides the spatial location of the total vehicle center of gravity at a given loading. The test is performed using platform scales capable of simultaneously measuring the weight of all four wheels, which allows determination of the plan view location of the center of gravity, plus a tilt table to measure the height of the center of gravity. The vehicle chassis is rigidly attached to the tilt table at a given trim height with cables and screw jacks. The table is then tilted through a series of angles, with the table overturning moments being measured by means of a platform scale. The tilt angles are measured with a precision inclinometer. The small lateral displacement of the vehicle body relative to the table is recorded as a correction factor for the tilt data.

A computer program fits a curve to the tilt table data and analyzes wheel load data to obtain the vehicle center of gravity location. The computed center of gravity height has been shown to be accurate to within 3 mm.

1980-1987 Production Passenger Car Handling Characteristics

The customer makes many demands of a motor vehicle, requiring different handling performance for vehicles with different missions. The customer expects that a sports car will have different handling characteristics than a light truck. Therefore, it becomes very important that objective measurements and comparisons can be made, to determine the range of performance that is acceptable to the customer, as represented by the marketplace.

The previous sections of this paper provided descriptions of several General Motors vehicle directional response test procedures. The remainder of this paper reviews test results for vehicles manufactured in the model years 1980 through 1987. Included are production vehicles manufactured by General Motors, domestic competition, Japanese competition, and European competition. These vehicles do not represent actual market or sales distributions. Test summaries do not represent the same vehicle population.

Control Response Test

The Control Response Test results include production vehicles manufactured during the 1980 through the 1987 model years. Of these vehicles, 51 percent were GM manufactured, 7 percent were manufactured by domestic competition, 21 percent by Japanese competition, and 21 percent by European competition.

At a constant speed of 100 km/h, the steering sensitivity is a measure of the rate of change of lateral acceleration with steering wheel angle. The value is determined in the linear range of the vehicle (0.15 g) and linearly extended to the lateral acceleration level expected at 100 degrees of steering wheel angle.

A distribution plot of the steering sensitivity of these vehicles is presented in Figure 3. The average value of steering sensitivity is 1.17 g's/100 degrees of steering wheel angle at 100 km/h. Values ranged from 0.59 to 2.17 g's/100 degrees, indicating a wide range of acceptability in the marketplace.

Roll gradient is a measure of the tendency of the vehicle to roll or "lean" during cornering. This value is determined by the rate of change of a roll angle versus lateral acceleration curve at 0.15 g lateral acceleration.

A distribution plot of the roll gradient of these vehicles at 0.15 g's lateral acceleration is presented in Figure 4. This distribution is skewed toward roll gain levels of 6-8 deg/g. The average value of roll gain is 6.4 deg/g. Values ranged from 1.5 to 11.3 g's/100 degrees. It is apparent that a wide range of roll gain is accepted in the marketplace.

A distribution plot of the understeer gradient at 0.15 g lateral acceleration is presented in Figure 5. This distribution is truncated on the low understeer side. No vehicles have been found to be oversteer in this range of maneuvering severity. The average value of understeer is 3.2 deg/g. Values ranged from 0.7 to 7.0 deg/g.

Lateral acceleration response time is defined as the time required for the lateral acceleration to reach 90 percent of its eventual steady state value from the time at which 50 percent of the steering input is reached for a step-like steering input. This metric has been found to correlate well with subjective opinions. Lateral acceleration response time differences are easier to discriminate because they are typically longer than yaw velocity response times, and are felt more directly as lateral force by the driver.

A distribution plot of the lateral acceleration response time at 0.30 g is presented in Figure 6. The lateral acceleration response time is summarized at 0.3 g's to reduce signal to noise ratio influences on the summarized value. The average value of lateral acceleration response time is 0.39 sec. Values ranged from

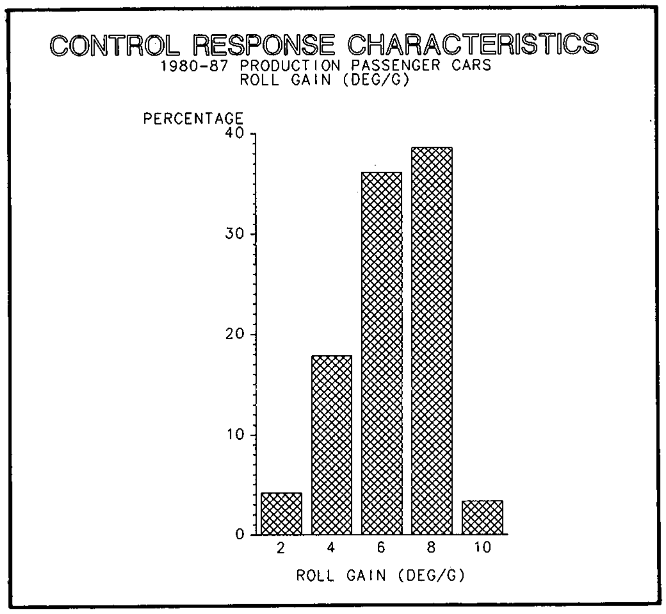

Figure 4

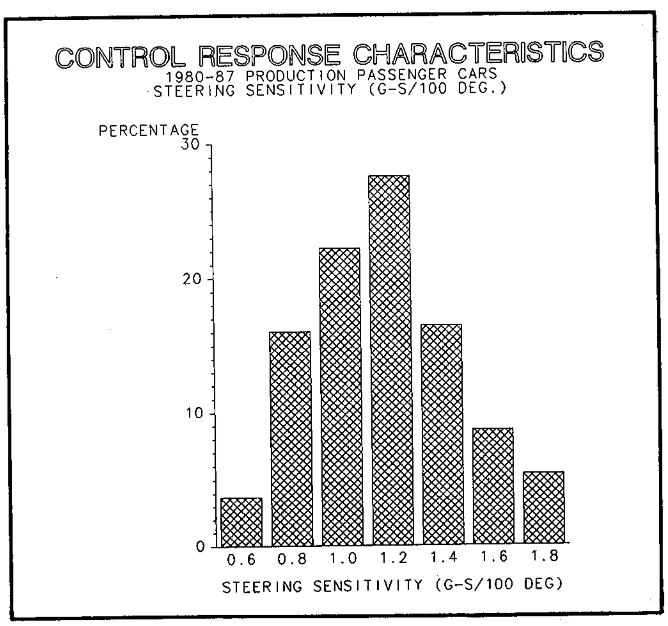

Figure 3

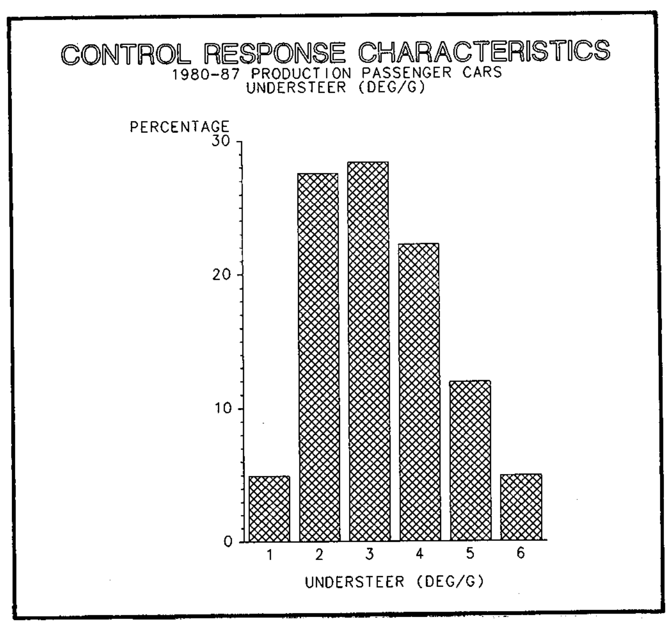

Figure 5

0.23 to 0.77 sec. Short lateral acceleration response times are perceived as better.

Frequency Response Test

The Frequency Response Test results include production vehicles manufactured during the 1983 through the 1987 model years. Of these vehicles, 33 percent were GM manufactured, 9 percent were manufactured by domestic competition, 33 percent by Japanese competition, and 25 percent by European competition. Many of these vehicles are equipped with optional handling suspensions and tires.

Lateral acceleration bandwidth is the frequency at which the lateral acceleration response of the vehicle is reduced by 3 decibels. A distribution plot of the lateral acceleration bandwidth of these vehicles is presented in Figure 7. The average value of lateral acceleration bandwidth is 0.99 Hertz. Values ranged from 0.66 to 1.32 Hertz. High lateral acceleration bandwidths are perceived as better.

Yaw velocity bandwidth is the frequency at which the yaw velocity response of the vehicle is reduced by 3 decibels. A distribution plot of the yaw velocity bandwidth of these vehicles is presented in Figure 8. The average value of yaw velocity bandwidth is 2.8 Hertz. Values ranged from 2.1 to 3.5 Hertz. Again, high yaw velocity bandwidths are perceived as better if the yaw damping is acceptable.

The peak to steady state yaw velocity ratio yields a measure of the damping of the vehicle yaw velocity response. A distribution plot is presented in Figure 9. The average value of the peak to steady state yaw velocity ratio for these vehicles is 1.4, ranging from 1.1 to 2.1. Low ratios are perceived as better if the yaw velocity bandwidth is acceptable.

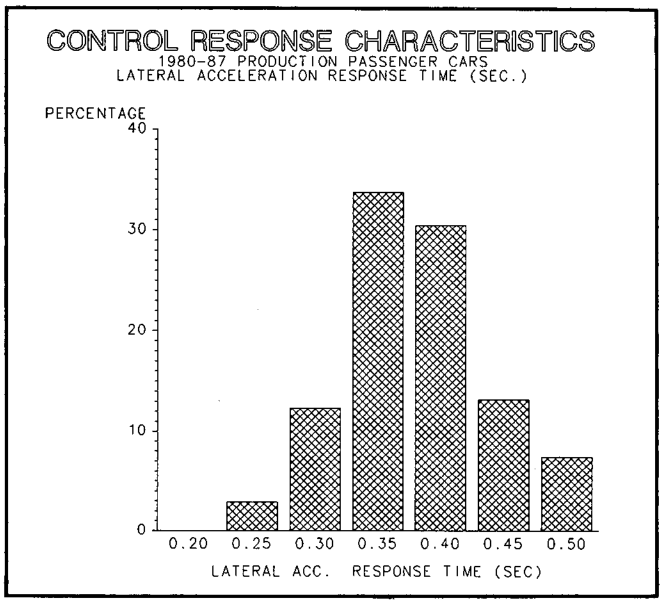

Figure 6

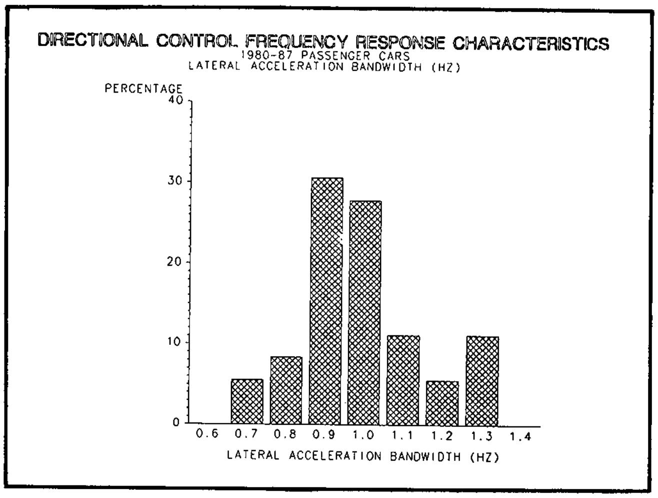

Figure 7

Both the yaw velocity and roll responses of vehicles exhibit a peaked frequency response. The ratio of the frequencies at which the corresponding peaks occur can be used to indicate roll-yaw velocity coupling in vehicles with lightly damped roll modes. A distribution plot of the peak roll-yaw velocity frequency ratios is presented in Figure 10. The average value for these vehicles is 2.0, ranging from 1 to 4. Values higher than 1 are perceived to be better for vehicles with lightly damped roll modes.

It has been noted in the literature that subjective opinions of vehicle handling improve when the phase difference between yaw velocity and lateral acceleration is reduced. This parameter can be determined from the frequency response test results. The actual parameter is the average phase angle difference between the yaw velocity and lateral acceleration between 0.2 Hertz and the lateral acceleration bandwidth of the vehicle. A distribution plot of the phase differences measured is presented in Figure 11. The average value for these vehicles is 17 degrees. The

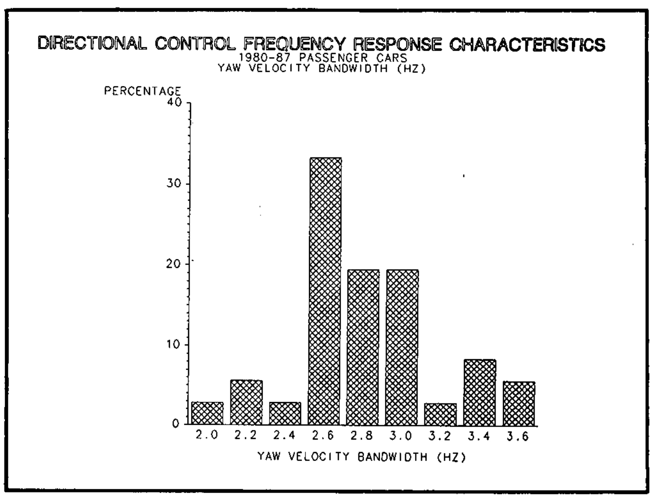

Figure 8

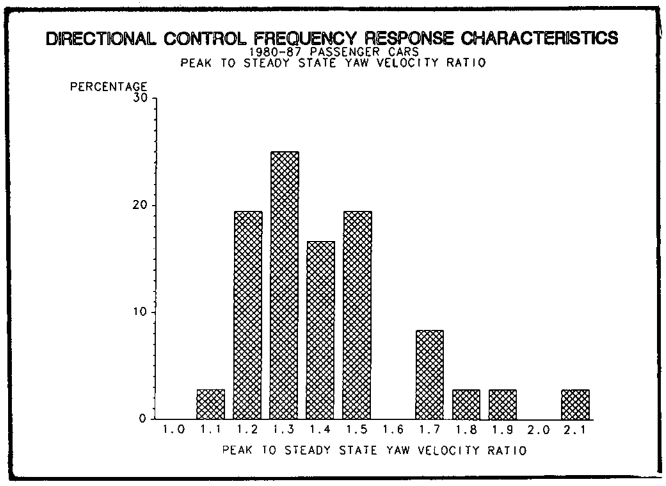

Figure 9

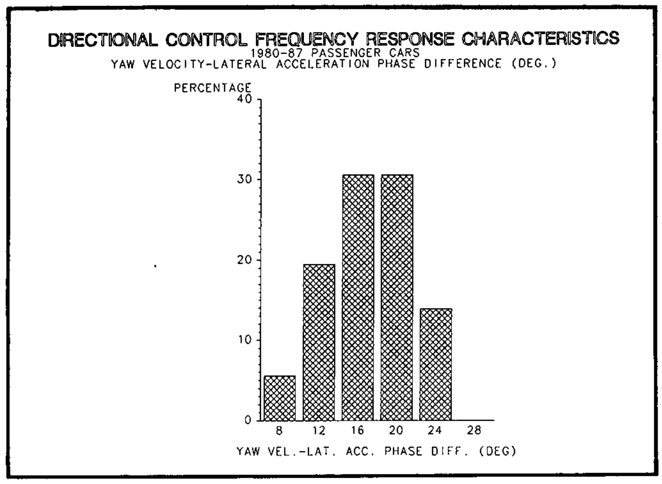

Figure 11

phase differences range from 8.9 degrees to 24.9 degrees.

Maximum Lateral Acceleration Test

The maximum Lateral Acceleration Test database consists of production passenger cars manufactured during model years 1980 through 1987. Most of these vehicles are equipped with optional handling suspensions and tires. Fifty five percent of the vehicles were manufactured by General Motors. Of the remaining vehicles, 3 percent were domestic competition, 26 percent were European competition, and 16 percent were Japanese competition.

Although maximum lateral acceleration is often used as a measure of ultimate handling performance levels, the customer will almost never operate the vehicle at these levels on public roads. A more significant factor indicated by high maximum lateral acceleration levels is the linear range of the vehicle performance. Vehicles with high maximum lateral acceleration performance tend to maintain linear steering characteristics over a wider range of lateral acceleration, a characteristic typically found in high performance vehicles.

A distribution plot of the maximum lateral accelerations is presented in Figure 12. The average value of maximum lateral acceleration is 0.77 g's. Values range from 0.59 to 0.91 g's.

On-Center Handling Test

The On-Center Handling Test results include production vehicles equipped with power steering and manual steering manufactured during the 1980 through the 1987 model years. Of these vehicles, 49 percent were GM manufactured, 5 percent were manufactured by domestic competition, 22 percent by Japanese competition, and 24 percent by European competition.

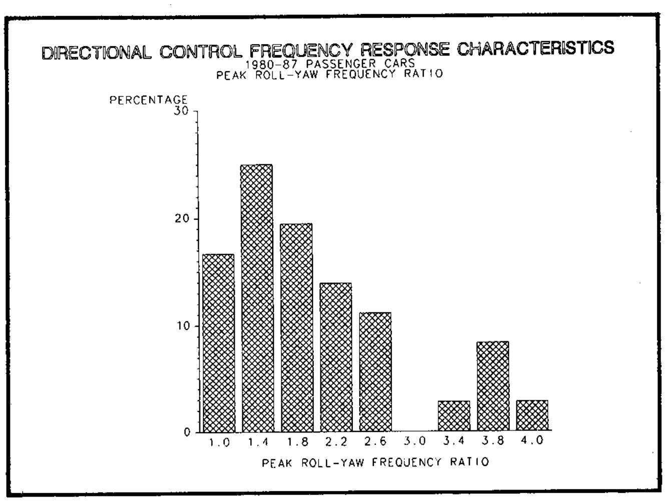

Figure 10

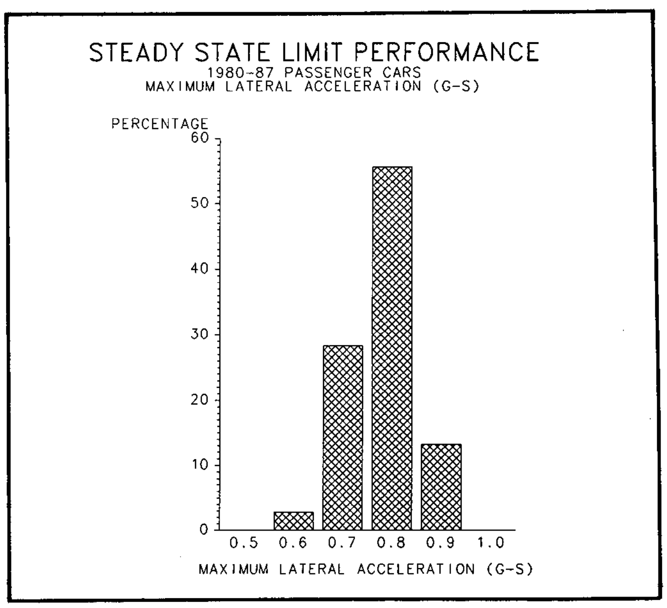

Figure 12

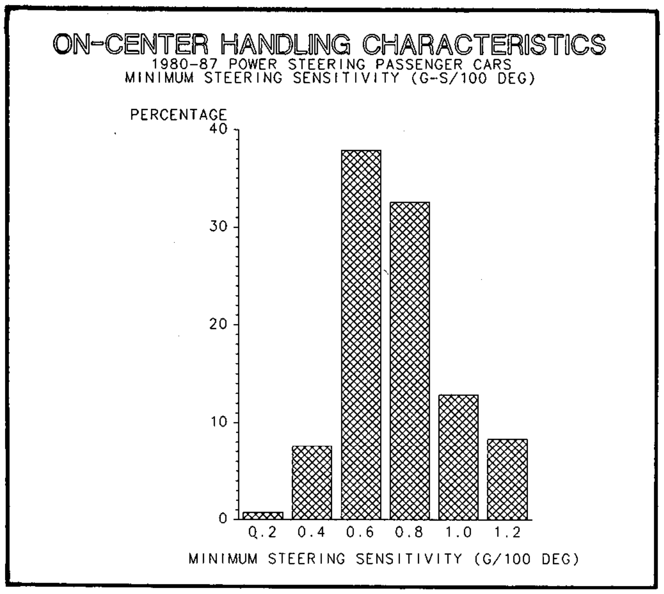

Figure 13

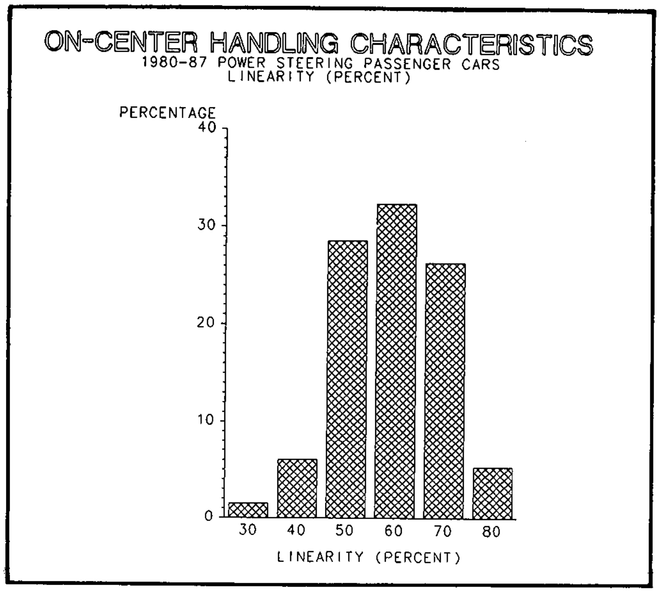

Figure 15

Minimum steering sensitivity is determined as the highest gradient of the steering wheel angle versus lateral acceleration curve within the 0.2 g range, left and right. This value is usually found close to the straight ahead position (0 lateral acceleration). High values represent vehicles subjectively determined to have a "crisp" feel. Subjectively desirable levels are above 0.5 g's/100 degrees steering wheel angle.

A distribution plot of the minimum steering sensitivities for power steering vehicles is presented in Figure 13. The average value of minimum steering sensitivity for power steering passenger cars is 0.74 g's/100 degrees steering wheel angle. Values ranged from 0.27 to 1.42 g's/100 degrees steering wheel angle.

A distribution plot of the minimum steering sensitivities for manual steering vehicles is presented in Figure 14. The average value of minimum steering sensitivity for manual steering passenger cars is 0.86 g's/100 degrees steering wheel angle, slightly higher than that for power steering passenger cars. Values ranged from 0.41 to 1.21 g's/100 degrees steering wheel angle.

Minimum steering sensitivity cannot be used by itself to judge a vehicle's on-center handling properties. High minimum steering sensitivities can be achieved by designing the vehicle with a high off-center steering sensitivity. Therefore, we must also review the ratio of the minimum steering sensitivity to the steering sensitivity at a higher lateral acceleration

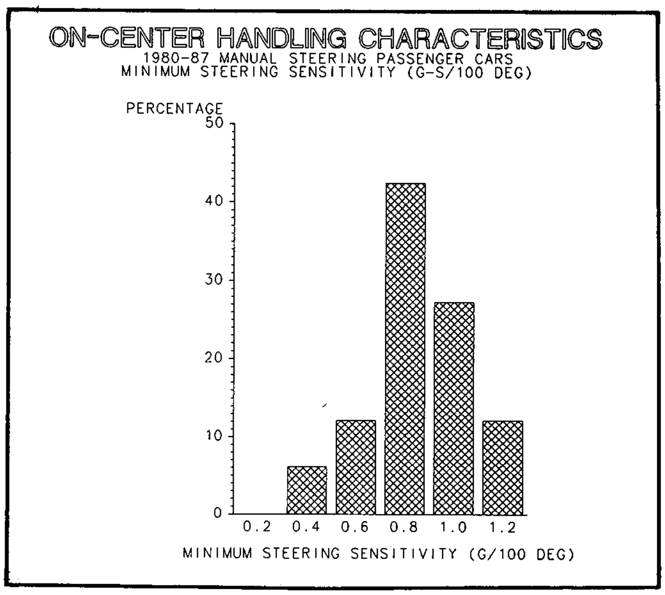

Figure 14

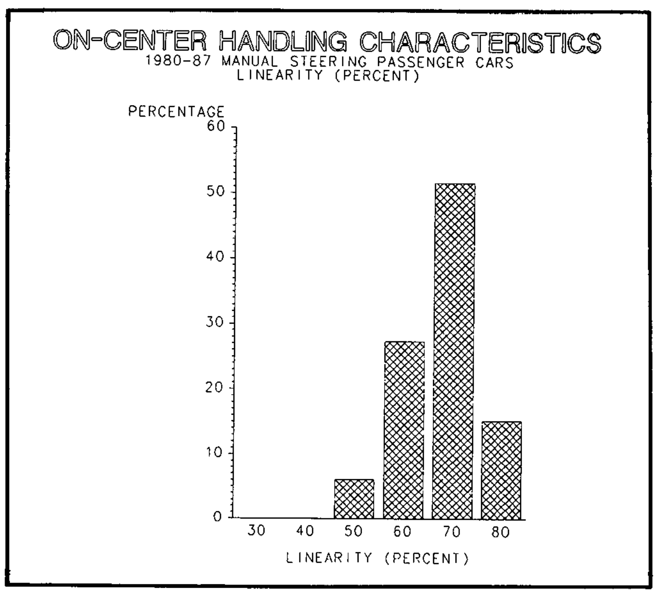

Figure 16

level (0.1 g's). This ratio has been called "linearity". One hundred percent represents an exactly linear steering wheel angle to lateral acceleration relationship within the 0.1 g left and right lateral acceleration window.

A distribution plot of linearity for power steering vehicles is presented in Figure 15. The average value of linearity for power steering passenger cars is 60 percent. Values ranged from 25 to 83 percent. Sixty percent or higher is usually subjectively rated as desirable.

A distribution plot of the linearity for manual steering vehicles is presented in Figure 16. The average value for linearity of manual steering passenger cars is 68 percent, slightly higher than for power steering vehicles. Values ranged from 54 to 81 percent. Note the absence of linearity values below 54 percent for manual vehicles, indicating that at high speed, most manual steering vehicles will be rated subjectively better than many power steering vehicles for "linearity".

During the On-center Test, the steering wheel torque is measured. The torque is then cross-plotted with the lateral acceleration to determine the steering torque characteristics. This generates a hysteresis loop with crossing points on the lateral acceleration axis. At these points (one left and one right) the steering wheel torque is zero. Therefore, the hands can, theoretically, be removed from the steering wheel, and the vehicle will maintain the indicated level of lateral acceleration. The average values of these two points represents a measure of the vehicle's "returnability".

A distribution plot of the returnability for power steering equipped vehicles is presented in Figure 17. The average value of returnability for power steering passenger cars is 0.06 g's. Values range from 0.01 to 0.13 g's. Levels lower than 0.07 g's are usually subjectively rated satisfactory.

A distribution plot of the returnability for manual steering vehicles is presented in Figure 18. The average value of returnability for manual steering passenger cars is 0.06 g's. Values range from 0.02 to 0.12 g's. These levels are similar to those of the power steering vehicles.

Another factor that is measured during the On-center Test is the rate of change of steering wheel torque with lateral acceleration at zero lateral acceleration. This measurement is one indication of the "feel" of a vehicle.

A distribution plot of the steering torque gradients for power steering vehicles is presented in Figure 19. The average value of steering torque gradient for power steering passenger cars is 18 N-m/g. Values range from 7 to 33 N-m/g.

A distribution plot of the steering torque gradients for manual steering vehicles is presented in Figure 20. The average value of steering torque gradient for manual steering passenger cars is 18 N-m/g identical to that of the power steering vehicles. Values range from 10 to 28 N-m/g.

A typical steering wheel torque characteristic for a power steering vehicle is for the torque to increase rapidly at low lateral acceleration levels, and then to limit at higher lateral acceleration levels as the steering assist becomes active. This transition is usually well advanced by 0.1 g's of lateral acceleration. Although this reduction in steering torque gradient is necessary to attain reduced parking efforts, it generates a non-linearity in the steering wheel torque versus lateral acceleration curve. The ratio of steering wheel

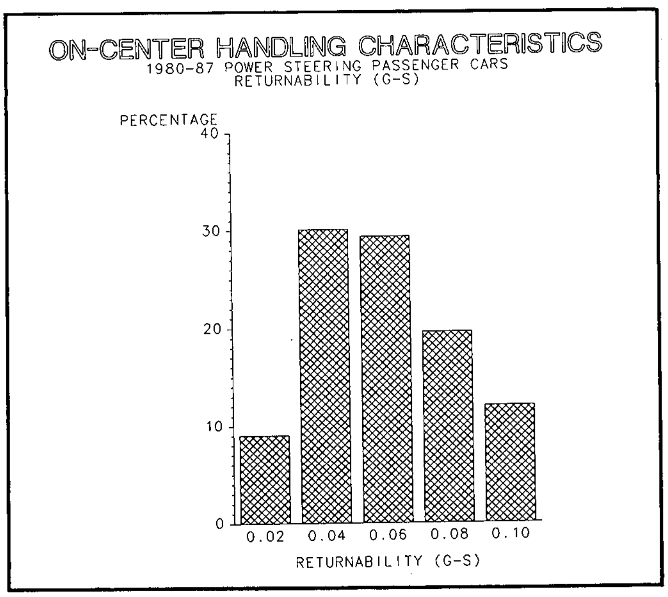

Figure 17

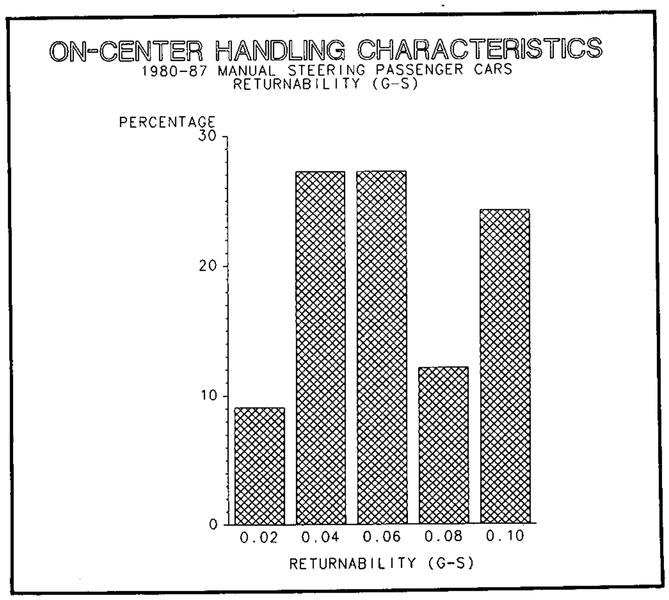

Figure 18

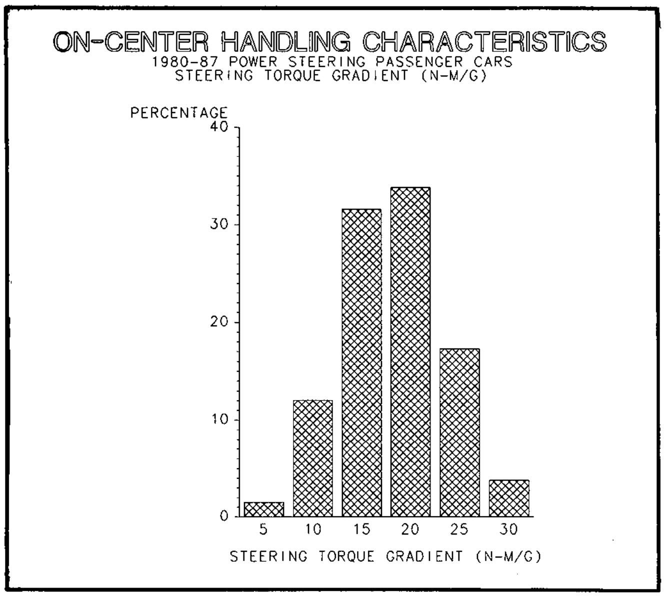

Figure 19

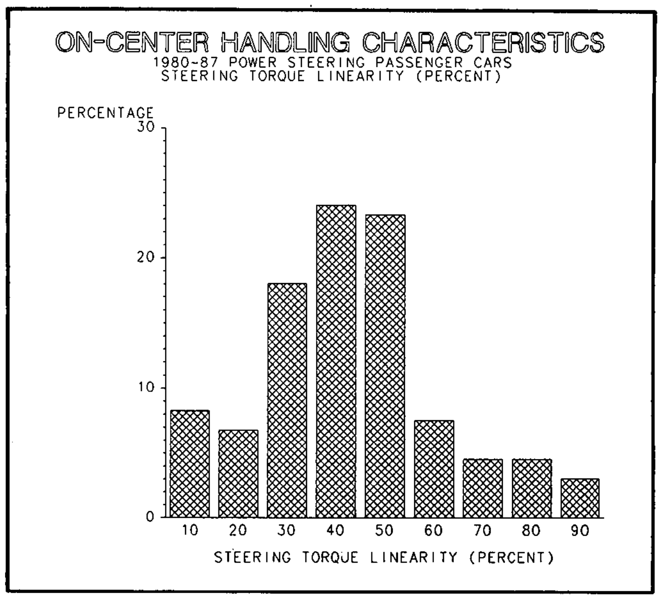

Figure 21

torque gradient at 0 and 0.1 g's of lateral acceleration is called "steering torque linearity".

A distribution plot of the steering torque linearity for power steering vehicles is presented in Figure 21. This distribution indicates a wide range of steering torque linearity in production power steering vehicles. The average value of steering torque linearity for power steering equipped passenger cars is 44 percent. Values range from 6 to 121 percent.

A distribution plot of the steering torque linearity for manual steering is presented in Figure 22. This distribution is highly skewed toward 100 percent, as one would expect for manual steering, since no steering boost is available. The average value of steering torque linearity for manual steering passenger cars is 89 percent, much higher than for power steering vehicles. Values range from 30 to 129 percent.

Steering work sensitivity is a measure of the work expended by the driver in order to generate lateral acceleration of the vehicle. It is a measure of the balance achieved by the vehicle between steering torque gradients and steering sensitivity. A vehicle with low work sensitivity requires high amounts of work to maneuver it, either due to high steering torque levels or low steering sensitivity. Conversely, a vehicle with high work sensitivity requires low levels of work input to maneuver it, due to low steering torque levels, or high steering sensitivity.

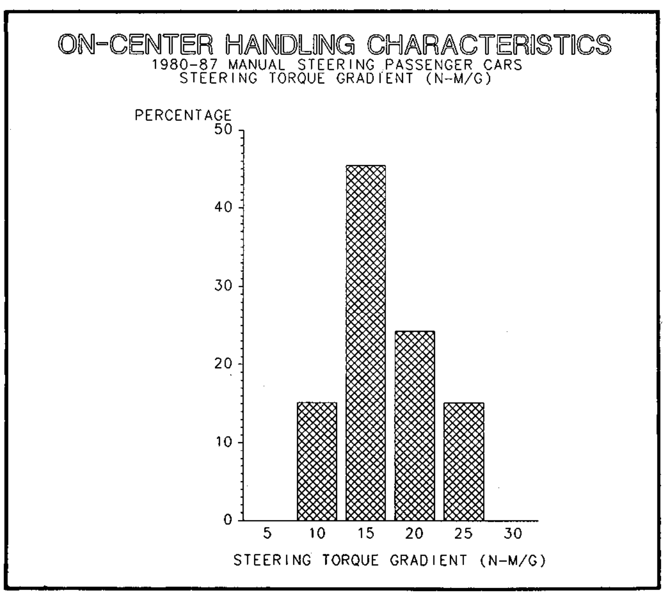

Figure 20

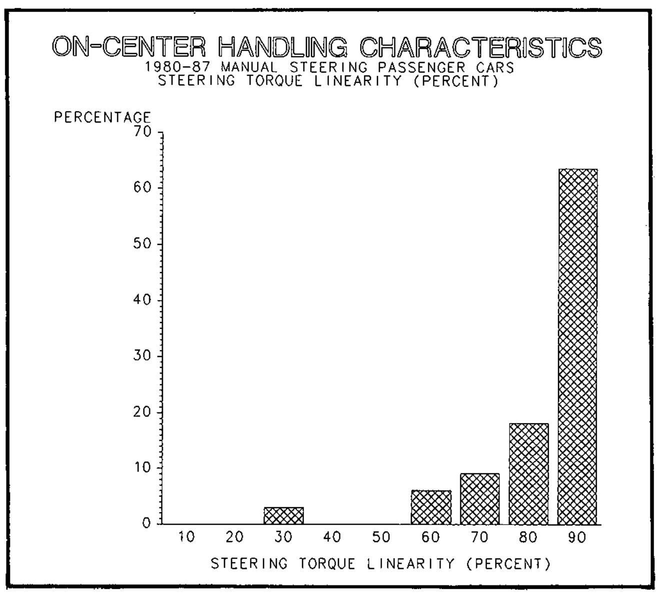

Figure 22

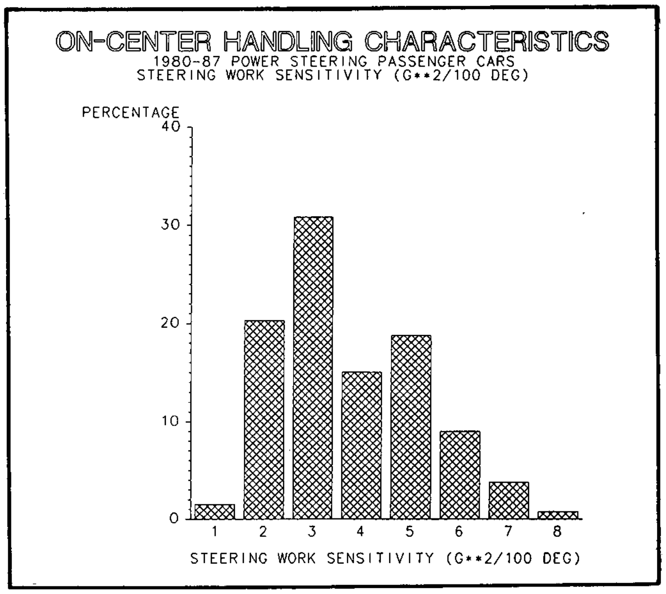

Figure 23

A distribution plot of the steering work sensitivity for power steering vehicles is presented in Figure 23. This distribution is skewed toward low work sensitivity levels, although a wide range of performance is noted in production power steering vehicles. The average value of steering work sensitivity for power steering passenger cars is 3.8 $g^2/100$ N-m. Values range from 1.3 to 8.5 $g^2/100$ N-m.

A distribution plot of the steering work sensitivity for manual steering vehicles is presented in Figure 24. This distribution is more normally distributed than the distribution for power steering vehicles. The average value of steering work sensitivity for manual steering passenger cars is 4.2 $g^2/100$ N-m, somewhat higher than for power steering vehicles. This implies that the average manual steering vehicle at 100 km/h requires less work to maneuver on center than the average power steering vehicle, although the ranges of performance overlap extensively. Values range from 1.1 to 9.8 $g^2/100$ N-m.

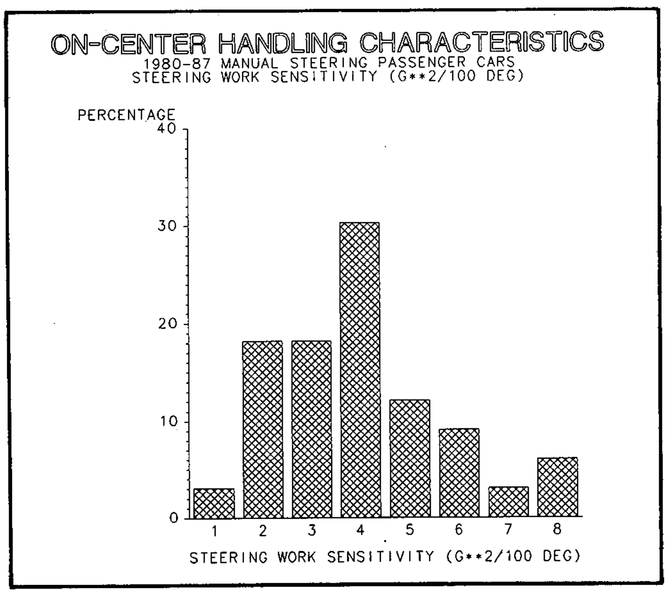

Figure 24

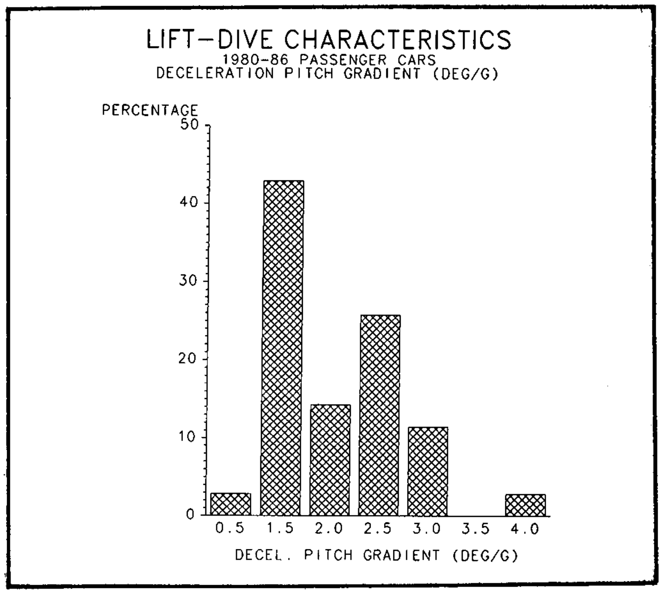

Figure 25

Lift-Dive Test

The Lift-Dive test database consists of production passenger cars manufactured during model years 1980 through 1986. Thirty four percent of the vehicles were manufactured by General Motors. Of the remaining vehicles, 20 percent were domestic competition, 3 percent were European competition, and 46 percent were Japanese competition.

A distribution plot of the acceleration pitch gradients is presented in Figure 25. This distribution is skewed toward the higher pitch gradients, due to the preponderance of strut suspension cars in the database. The average value of acceleration pitch gradient is 3.3 deg/g. Values range from 1.3 to 5.2 deg/g.

A distribution plot of the deceleration pitch gradients is presented in Figure 26. This distribution represents an approximately normal distribution, with the exception of a preponderance of vehicles in the 1.5 deg/g range. The average value of deceleration pitch gradient is 2.0 deg/g. Values range from 0.9 to 3.9 deg/g. The average value for deceleration is less than the value for the acceleration pitch gradients, since both axles contribute to the antidive properties of the chassis during braking, but only the drive axle contributes antilift during acceleration (No four wheel drive vehicles were included in the data sample).

Center of Gravity Test

The Center of Gravity data consists of production passenger cars manufactured during model years 1980

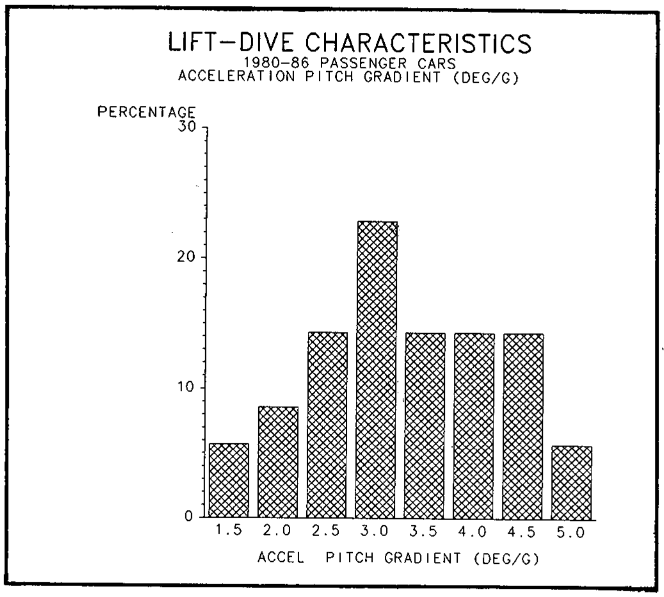

Figure 26

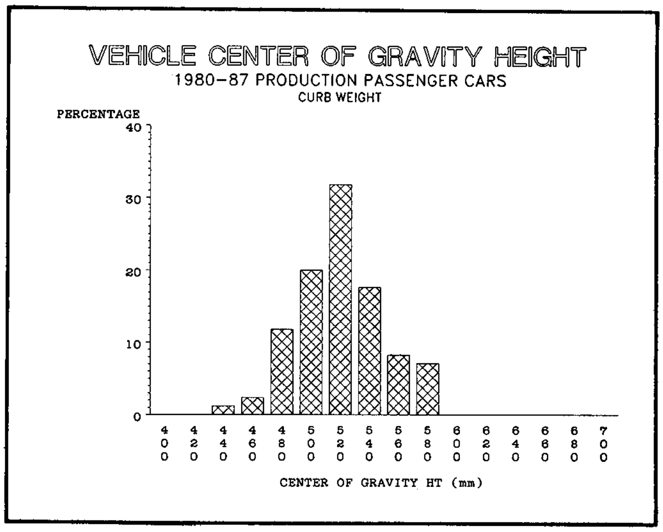

Figure 27

through 1987. Twenty percent of the vehicles were manufactured by General Motors. Of the remaining vehicles, 15 percent were domestic competition, 29 percent were European competition, and 36 percent were Japanese competition.

A distribution plot of the center of gravity heights is presented in Figure 27. The average value of vehicle center of gravity height is 520 mm. Values range from 437 mm to 589 mm.

Summary

This paper documents the usefulness of open-loop test procedures in the measurement of vehicle directional response characteristics. The development of directional response test procedures in the U.S. was discussed. An overview of some of the objective test procedures used by General Motors for the assessment of vehicle directional response characteristics was provided, together with a discussion of the key performance metrics obtained from these tests. A statistical analysis of test data for 1980 through 1987 model year production vehicles was performed, noting averages and ranges for these performance metrics.

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The Ergonomics of Driver Information Systems for Maintained Safety (Written only paper)

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Abstract

The development of a safe vehicle must recognize the requirement to maintain that level of safety for the design life of the vehicle. Whilst secondary safety, i.e. occupant protection is to a large part determined by the design team, primary safety, i.e. accident avoidance, has in the past been the ultimate responsibility of the owner/driver.

In recent years with new developments this situation has begun to change with the provision by the manufacturer of systems to aid accident avoidance: antilock braking is perhaps the most obvious example.

This paper proposes that another important aspect of primary safety where the motor manufacture can

significantly aid the owner is in the area of vehicle maintenance. Recent major developments in automotive electronics hold the potential to provide real assistance to the driver in monitoring the condition of the vehicle and thereby encourage good maintenance and safety.

The paper refers to new research by the Automotive Ergonomics Unit at the Institute for Consumer Ergonomics in the area of in-car driver information systems. It briefly discusses the ergonomics of alternative warning or attention getting devices and driver information systems. Drawing further upon the latest research it compares alternative methods of message presentation such as voice synthesis and alphanumeric displays. The paper concludes that ergonomic recommendations for driver information systems can be valuable in encouraging maintenance and thereby enhancing vehicle safety.

Introduction

Automobiles today leave the factory with a high level of built-in safety. Manufacturers invest considerable resources to research, develop and implement both primary safety features such as braking systems, tyres, steering, handling and secondary safety features such as crumple zones, seat belts, collapsible steering columns. The responsibility for monitoring and maintaining safety features and ensuring that they function effectively, as designed, becomes the task of the owner/driver, whether this is, for example, to maintain the braking system or to ensure that doors are properly closed. The manufacturer's role in 'maintained safety' is not finished once the vehicle comes into service however. From the very earliest vehicles a handbook of some form was provided to explain to the owner how to care for the vehicle and maintain its components. In recent years this form of "assistance beyond the showroom" has started to be supported by electrical/electronic systems built into the car to monitor vehicle components and inform the driver of the status of those components. Warning and information systems are also extending beyond indicating vehicle status, for example it is possible to include systems to alert the driver to the state of the environment, e.g. ice and fog warning.

This is an area of vehicle development which should and will grow significantly in future years. However, if the investment in research development and implementation is to be effective it is essential that some minimum ergonomic criteria for the driver information systems are met.

The objective of these criteria are to ensure that driver information systems provide pertinent information, at the right time, in a form which will be acted upon by the driver, and that they are perceived by the driver as a real aid to vehicle safety and reliability. Systems which do not meet these criteria may fail to fulfil their safety role not only by poor information presentation but also due to lack of perceived utility which will lead to disuse or even disablement of the system (Appleby and Bin*tz*).

There are three primary requirements for effective driver safety information systems:—

1. Alert the driver to change in status of component, system or environment.
2. Inform the driver which component, system or environmental factor is involved, e.g. brakes, doors, ice.
3. Inform the driver about the necessary actions they should take.

There are also secondary features which may prove important in maximising the utility and effectiveness of the system:—

- a demonstration mode which enables a preview of the warning system information (sometimes called "showroom mode").
- pre-drive safety checks which permit the verification that both the safety systems and monitoring systems are functioning.
- reminder systems where the action is to be taken at some later time, or where the driver fails to respond to an urgent warning.

Alerting the Driver

It is necessary to quickly attract the driver's attention to the warning. There is clear evidence of the superiority of auditory warnings (tones) over visual warnings (lamps) in this role. During a series of road trials with a standard production car and a range of drivers the authors measured the time it took drivers to respond to an unexpected red warning light. This light was of large area and conspicuously located in the main instrument pack. It was found that drivers' reaction times varied from 1 second to 58 minutes with nearly 60% of drivers taking longer than 30 seconds to notice the appearance of the warning light. Clearly a warning lamp is less than adequate in alerting drivers to critical situations. In a direct comparison of audible and visual attention getters, an audible "bing-bong" was highly superior. Compared to the response times of over 30 seconds to the visual lamp, the reaction times to the "bing-bongs" were virtually instantaneous.

Whilst there is not space to go into the details of tone generation in this paper the amplitude, frequency and envelope are critical to ensuring a tone is both sufficiently alerting whilst at the same time not evoking a startle response or panic in the driver.

The use of other types of display such as tactile displays may be suitable for specific warnings but not as a general attention getter. ABS systems typically indicate that they are operational by a pulsed feedback via the brake pedal. Such a device is too highly control specific to serve as a generally alerting medium.

Informing the Driver About the Nature of the Warning

The traditional ISO symbols have been shown to be less than adequate for this purpose. The Automotive Ergonomics Unit has found a high proportion of drivers confuse the various symbols. We must not forget that drivers are inexpert and warnings are infrequent so they cannot learn through experience (and do not study sufficiently to recall information from the handbook). Clearly more informative means are required at the time of warning presentation. Thus alpha-numeric displays and voice synthesized messages must be considered, preferably in addition to the ISO symbols.

From an ergonomics perspective these are not alternatives because they each fulfil different ergonomics requirements. The alpha-numeric display is permanently available whereas the audible display is temporary unless frequently repeated. If it is frequently repeated than it may prove distracting while the driver is attempting to deal with other driving tasks. It may also prove annoying. On the other hand, while the alpha-numeric display is permanent the driver must divert his eyes from the road scene in order to read the message.

The textual information presented on an alpha-numeric display may be read and interpreted more quickly than a spoken message. During tests in the vehicle simulator at the Automotive Ergonomics Unit an alpha-numeric display was compared directly with voice synthesis in a vehicle condition monitor which provide information simultaneously in both forms. It was found that upon first hearing the alerting tone drivers immediately looked to the display to read the information rather than keeping their eyes on the road and waiting for the spoken message to be completed. The drivers' reasons for doing this were stated that they could capture the information much more quickly from the visual display.

In conclusion it is recommended that an alpha-numeric display is used for all such information even if a full voice synthesis system is provided. A more integrated approach would be to retain the voice messages for supporting high priority warnings particularly if these require action while driving under difficult conditions.

The role of tactile displays in this is severely restricted to traditional areas. For example the steering wheel may be used to indicate changes in the condition of the tyres or brake pedal vibration to indicate that the ABS system is operating. However, tactile displays do not have a general applicability as even if the driver can interpret the information, considerably greater attentional effort is required for this than with visual or auditory displays.

Inform the Driver About What Actions to Take

For both safety and driver confidence it is necessary to tell the driver not only what remedial action is required ("have the brakes serviced") but also when this should be done ("within 3000 miles").

It is, of course, critical to tell the driver whether he can continue driving, or whether he should pull over now. If driving can continue he should be clearly told when action is required, for example, at the next filling station, or at the next service.

Ultimately the effectiveness of the system is dependant upon the driver acting upon the information presented. From our own customer research we have clearly found that such systems must be wholely reliable and provide practical assistance if they are to be respected and used properly by owners.

As vehicles and their systems become increasingly complex and DIY repair less viable then the role of the driver is to identify when faults arise with support form on-board electronic systems. Information systems which inform upon the status of components will give the driver confidence that the potentially expensive service or repair is actually necessary and has been properly attended to by the garage. This again adds to driver confidence and trust in the systems.

Summary of Recommendations

Alert the driver—auditory "bing-bong" preferred to "siren" to avoid startle and panic.
—auditory preferable to warning light.
—warning light should be provided as a back-up.

Inform the driver of the:

Nature of the warning
—alpha-numeric
—ISO symbols not highly informative

Action to take
—alpha-numeric
—timescale of action ('stop driving now' or 'at next service')

Repetition
—if no action is taken
—increase the urgency only as appropriate

Demonstration facility
—provide a means of drivers sampling all the warning functions upon request.

Systems verification
—enable pre-drive check on components.
—enable pre-drive check on the monitoring system.

Conclusion

It has been the objective of this paper to draw attention to the fact that safety design can and should include provision to enable the owner/driver to maintain his vehicle's safety. The development systems which provide information about vehicle components which are likely to require attention during the normal use of the vehicle must be included within the ambit of the Experimental Safety Vehicle.

Of course, we cannot guarantee that the driver will respond to the information. Indeed studies have shown that drivers may ignore, for example, seat belt warning or reminder devices (Robertson et al, 1974, Robertson, 1975). However, from personal communications with the U.K. Department of Transport there is evidence that the response may be more positive to more directly safety-related components. It appears that owners' responses to recalls concerning the braking system are much higher (around 80%) than they are to recalls due to faulty seat belt mounting (around 30%). Thus, amongst the many ergonomics specifications which pertain to driver information systems we must not forget the importance of achieving perceived as well as actual practical utility.

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Study on Easy Adjustability of Outside Mirrors for Crash Avoidance (Written only paper)

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Abstract

An easy adjustability of outside mirrors is important not only for the convenience but also for the crash avoidance. This paper describes the experimental study on the calculation of the driver's eye position from the inside mirror's angles adjusted by himself. The accuracies of the calculation of the driver's eye position were found to fall within ±20 mm in the vertical direction and ±40 mm in the longitudinal direction of the actual eye position. In addition, the accuracies were influenced by the sideways supporting of a seat. The visibility of the outside mirrors with those angles determined by the calculated eye position for the inside mirror angles was satisfactory when the outside mirrors with the convex of radius around 1000 mm were used. Results of the study indicated the feasibility of a mirror control device that would give a driver the proper rearward visibility through outside mirrors automatically.

Introduction

A lot of efforts have been made to improve the indirect rearward view. But, no other system has ever replaced the current mirror system commercially. The rearward viewing device should be a compromise among various factors, such as visibility, appearance, cost, weight, safety and so forth. Although the current mirror system is one of the practical compromises, it has blind zones as shown in Fig. 1. To minimize the blind zones, it is important for a driver to adjust each mirror properly.

The objective of this study is to find the accuracy of calculation of the driver's eye position for the inside mirror angles and to research the feasibility of an automatic adjusting device of outside mirrors through the signal obtained from the inside mirror angle.

Calculation of Eye Position

Generally, almost all drivers adjust an inside mirror before driving, while they do not always adjust outside mirrors because of their laziness or absentmindedness. An inside mirror is usually so designed that a driver can see the rearward through the whole area of the rear window. When a direct line of sight reflected at the center of an inside mirror passes through the center of the rear window as shown in Fig. 2, and the position of the driver in the lateral direction is restrainted by the seat, the driver's eye position will be calculated for the angles of the inside mirror.

Test Method

A Test vehicle as shown in Fig. 3 and the six seats were prepared for the investigation of the driver's eye position. The vehicle was equipped with two cameras

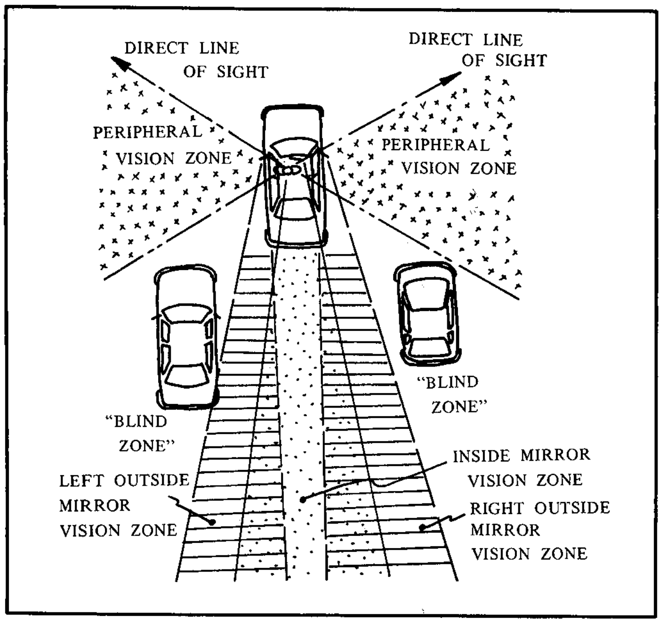

Figure 1. Vision zone by current mirror system

to measure the driver's eye position and the seats were selected to cover different characteristics of the sideways supporting.

Fig. 4 shows the stature of the test subjects, who are fifty Japanese males. The tests were conducted as the following procedure.

1. Test subjects get in the vehicle and adjust positions and angles of the seat, the steering wheel, the inside mirror and the outside mirrors before driving.
2. Test subjects drive the vehicle in the designated driving course for several minutes.
3. Test Subjects readjust the mirrors and others if necessary and then the eye positions and mirrors' angles are measured.
4. Tests are conducted for fifty subjects with six seats variation.

The instrumented mirrors with the angular sensors were used to measure mirrors' angles as shown in Fig.

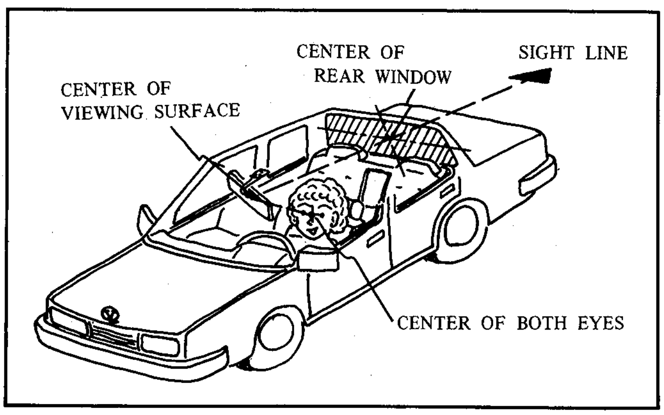

Figure 2. Ordinary adjustment of inside mirror

Figure 3. Test vehicle

5. The angular sensor consists of a magnet and two hall elements and measures the horizontal and vertical angles. The data of the angles can be read on the indicator installed on the passenger seat as shown in Fig.6.

Test Results

Fig. 7 shows the geometric relationship between the driver's eye and the inside mirror for which the driver's eye position is calculated.

P_1	:	Neck Pivot of Driver (assumed)
P_2	:	Eye Point viewed Inside Mirror (Center of Both Eyes)
P_2'	:	Projected Eye Point of P_2
P_2''	:	Eye Point viewed Straightly
P_3	:	Inside Mirror Pivot
P_4	:	Center of viewing Surface of Mirror
P_5	:	Center of Rear Window

The position of the neck pivot and the rotation of the head were assumed to have following conditions.

1. The distance from the eye point to the neck pivot is 85 mm (as the average of Japanese).
2. The driver rotates his head horizontally and vertically by the horizontal and vertical angles of the inside mirror respectively.

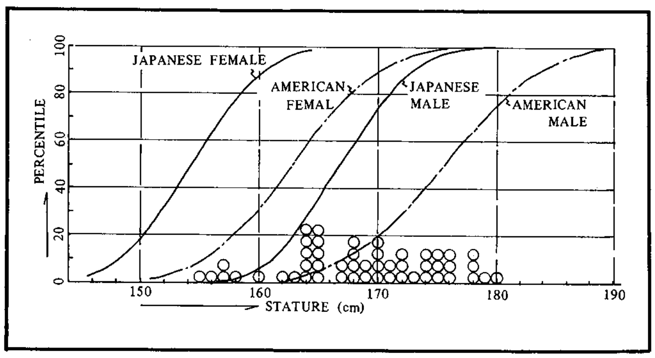

Figure 4. Stature and number of test subjects

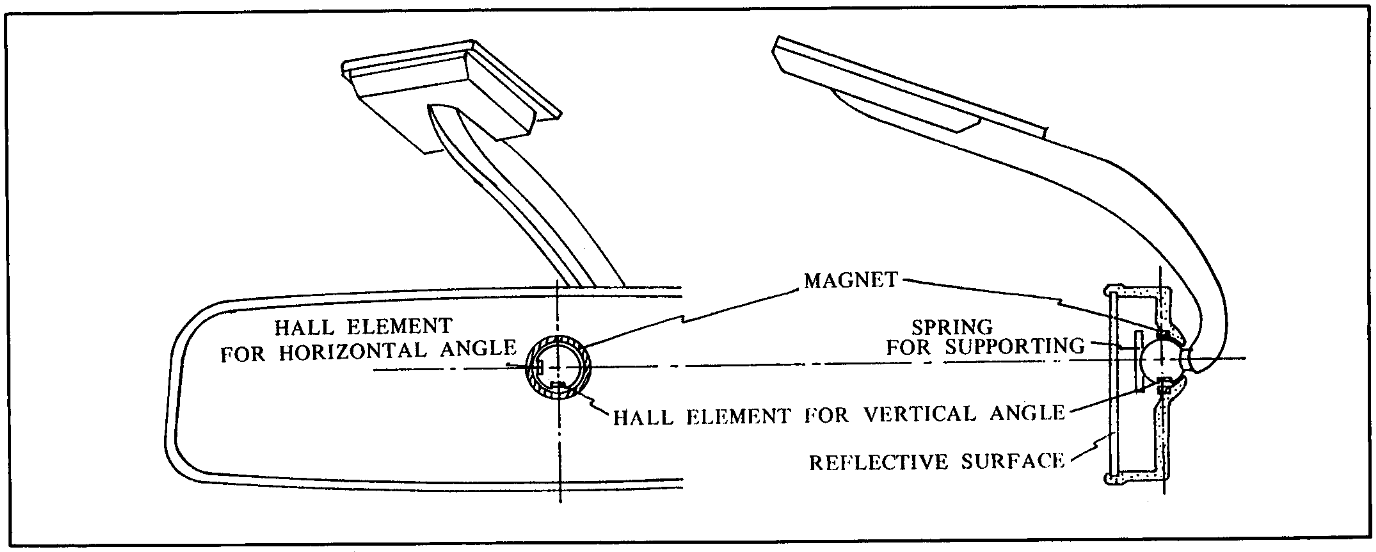

Figure 5. Inside mirror with angular sensor

The eye point (X_2'', Z_2'') viewed straightly for the angles (B, F) of the inside mirror was calculated as follows.

Since P_3 and P_5 are given, the angle of the depression (θ) of P_5 from P_4 in VIEW A is obtained by the following equations.

$$T_1 = Z_3 - D_1 \sin B - D_2 \cos B - Z_5$$

$$T_2 = [X_5 - X_3 - (D_1 \cos B - D_2 \sin B) \cos F] \cos F$$

$$\theta = \arctan (T_1/T_2)$$

where,

Z_5 = Z co-ordinate of P_5

X_5 = X co-ordinate of P_5

and, using the given Y_1, P_4, it follows that

$$YP = Y_1 - 85 \cos B \sin F - (D_1 \cos B - D_2 \sin B) \sin F$$

$$KP = Yp/\sin F$$

$$KV = 0.5\, KP/\cos B$$

$$GV = KV \sqrt{T_1^2 + T_2^2}/(T_2 \cos B + T_1 \sin B)$$

$$FV = GV \sin\theta$$

$$ZV = 2\, KV \cos B - FV$$

$$Z_2 = Z_3 - D_1 \sin B - D_2 \cos B - ZV$$

thus, $Z_2'' = Z_2 - 85 \sin B$

and, $DV = GV \cos\theta$

$$AV = KP - DV$$

$$HP = AV/\cos F$$

$$XP = HP - YP \tan F$$

$$X_2 = X_3 + (D_1 \cos B - D_2 \sin B) \cos F + XP$$

Thus, $X_2'' = X_2 + 85 (\cos B \cos F - 1)$

On the other hand, for the purpose of quantifying the sideways supporting which may affect the eye position, two figures (H_1 and H_2) shown in Fig. 8 were measured. From the subjective evaluations, the index for the sideways supporting of the seat was set

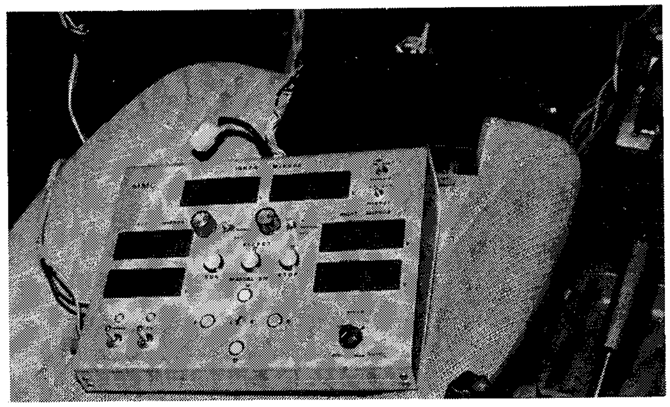

Figure 6. Indicator for angles of mirrors

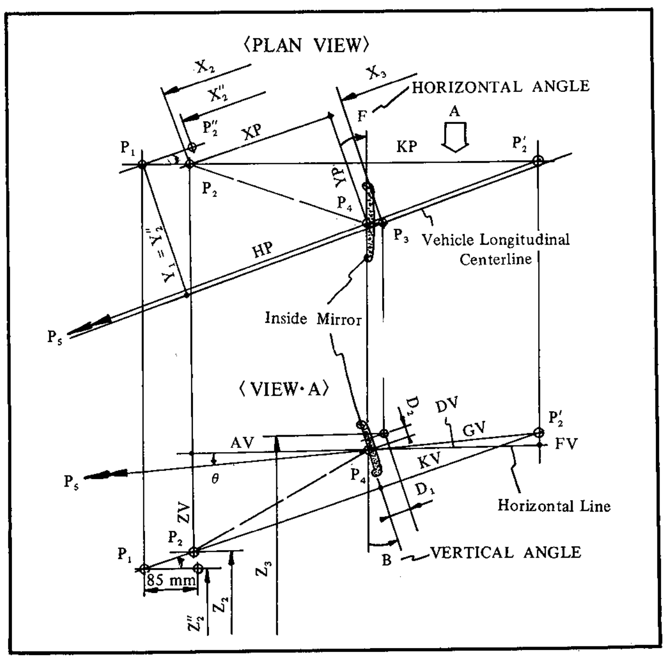

Figure 7. Eye-inside mirror geometry

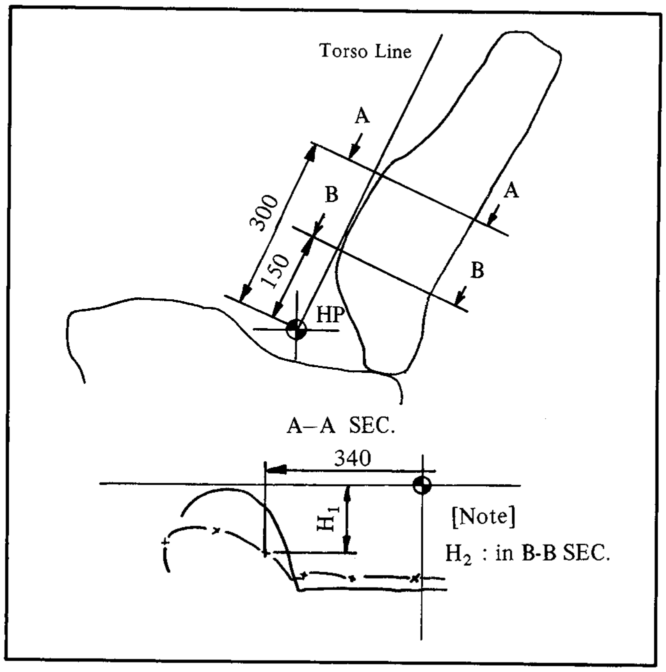

Figure 8. Bucket shape of seat

up as shown in Table 1, where the seat with smaller value of index showed stronger sideways supporting.

Calculation of Eye Position for Inside Mirror Angles Fig. 9 and Fig. 10 show the relationship between the measured eye positions when viewed straightly and those calculated for the inside mirror angles in case of Seat A with the "Very Strong" sideways supporting. The calculated eye position fell within ± 20 mm in the vertical direction and ± 40 mm in the longitudinal direction of actual eye positions. The error is caused by individual variations, such as, lateral seating posture, rotation of the head when viewed the inside mirror, and adjustment of the inside mirror for the rearward view.

Regarding the influence of the sideways supporting on the eye position, Fig. 11 shows the correlative coefficient between the measured eye positions and those calculated for the supporting indexes. Stronger sideways supporting has shown higher correlative coefficient.

These results indicate that the accuracy of the calculated eye position will be improved by the use of

Table 1. Index of sideways supporting.

	A	B	C	D	E	F
INDEX $\frac{H_1 + H_2}{2}$	30	50	55	65	55	75
Sbjective Evaluation	Very Strong	Fairly Strong	Rather Strong	Rather Weak	Normal	Very Weak

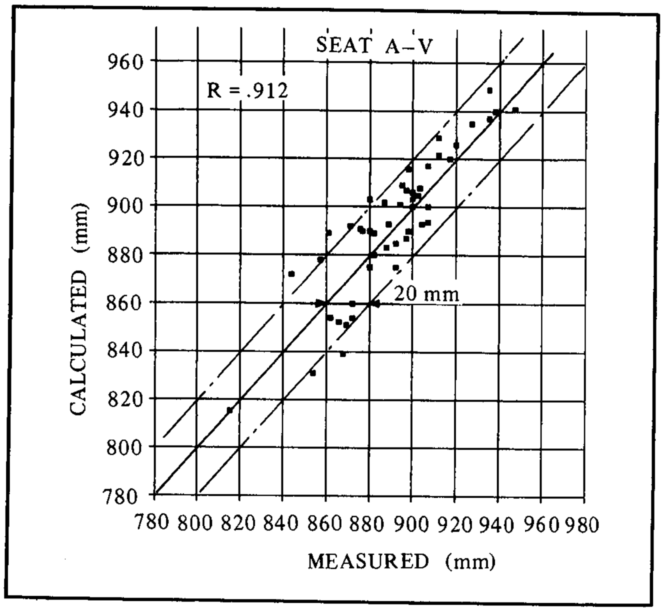

Figure 9. Vertical eye position (seat A)

the fairly strong sideways supporting for a driver's seat.

Adjustment of Outside Mirror. The other study was made on the rearward visibility through the outside mirror adjusted improperly. Fig. 12 shows the degree of driver's satisfaction in the case of the adjustment of the outside mirrors with the convex of radius 1000 mm, the width of 150 mm, and the height of 95 mm. The result showed that the range within $\pm 1°$ of the optimal adjustment angle was evaluated as 'Fairly Satisfied', namely the error less than $\pm 1°$ should be necessary in order to adjust the outside mirrors automatically.

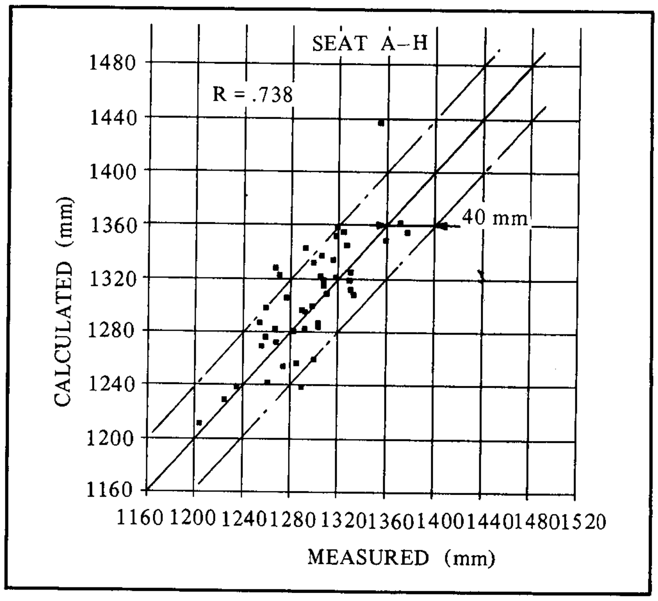

Figure 10. Longitudinal eye position (seat A)

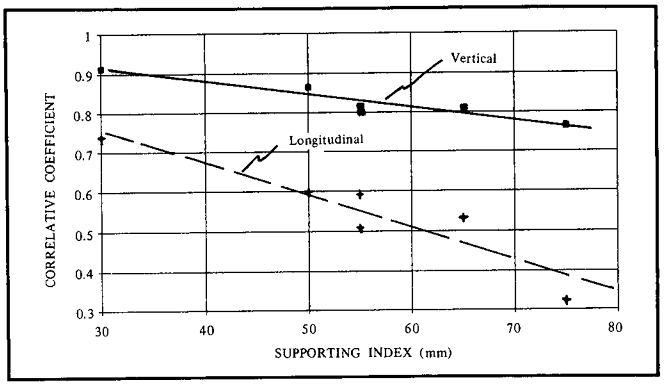

Figure 11. Correlative coefficient vs sideways supporting

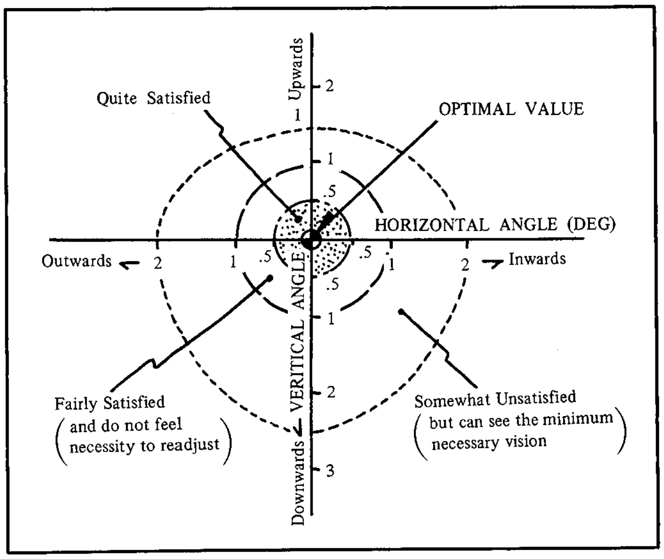

Figure 12. Subjective evaluation of angle of outside mirror

Table 2. Influence on outside mirror angle by calculation error of eye position.

Mirror Angle / Calculation Error	Vertical Angle	Horizontal Angle
Vertical Position (±20mm)	± 0.8°	± 0.07°
Longitudinal Position (±40mm)	± 0.5°	± 1.2°

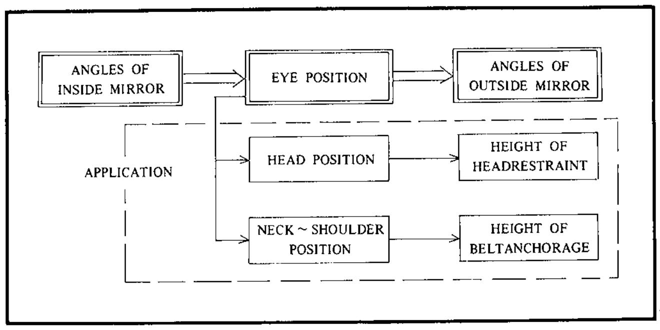

Figure 13. Application of this concept

Table 2 shows the influence on the angular deviation of the outside mirrors by the maximum errors of the calculated eye positions in the case of the mirror with the convex of radius 1000 mm. As a result, the errors of the calculation were approximately with the tolerance range (± 1°) described above.

In the case of the plane outside mirror mainly used on the driver's side in U.S.A., the tolerance range will be smaller than that of the convex mirror because of the narrower field of view.

Conclusions

The calculation of the driver's eye position for the angles of the inside mirror was found to be possible especially in the case of the seat with 'Fairly Strong' sideways supporting. The errors of adjustment of the outside mirrors determined by the calculated eye position is approximately within the tolerance range of the rearward visibility. A new mirror control system, the automatic adjustment of the outside mirrors, will be feasible for driver's convenience and crash avoidance by utilizing the calculated eye position for the inside mirror angles.

Further, if other parts of a driver, such as the positions of head and neck are calculated for the eye position, the optimal heights of the headrestraint and the belt anchorage will be also determined automatically as shown in Fig. 13.

Reference

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The Effect of Vehicle-Speed Sensing Four-Wheel Steering System on Handling Performance

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Abstract

"Yawing center" may be considered significant information for the driver to control vehicle steering performance.

This paper introduces the concept of "yawing center" of a vehicle in yawing motion, with the following factors analyzed by computer simulation with two-degrees of freedom:

- How "yawing center" of two-wheel steering car varies while running.
- How it is improved by adopting four-wheel steering system.

The results indicate that it is possible to control the position of "yawing center" by using a four-wheel steering system, which improves vehicle response to steering input and reduces cornering kinetic energy.

Introduction

The driver controls a car in general with many pieces of information on vehicle motion such as vehicle forward velocity, yaw velocity and lateral acceleration. We assume that the driver's feeling of "yawing center" of vehicle motion is also an important factor. "Yawing center" means the point around which driver feels that the vehicle is rotating in yawing motion.

From our experiences, the driver feels "yawing center" is around the rear axis at very low vehicle speeds, and moves forward as the vehicle speed increases.

Strictly speaking, the actual "yawing center" of vehicle motion varies according to steering action, but the driver feels some fixed "yawing center" exists.

In this report, we will discuss how the position of "yawing center" varies in two-wheel steering car and how it is improved by adopting a four-wheel steering system.

This is followed by discussions of correlations between vehicle response and "yawing center" and correlations between cornering kinetic energy and "yawing center".

These discussions are based on computer simulations using the two-degrees of freedom mathematical model.

Studies on "Yawing Center" of Two-Wheel Steering Cars

Definition of "Yawing Center"

As shown in Figure 1, we define "yawing center" as the point on the longitudinal axis of the vehicle which can be vertically lined to instant center of the motion. And "yawing radius" is the distance between center of gravity (C.G.) of vehicle and yawing center.

It is shown in Figure 1 that "yawing radius " is given by

$$\lambda = Ric\,\beta \qquad \text{for } |\beta| \ll 1$$

Furthermore, Ric = U/r, hence

$$\lambda = \frac{\beta}{r} U$$

where

Ric	:	Distance between C.G. of the vehicle and instant center of the motion
β	:	Vehicle slip angle
r	:	yaw velocity
U	:	Vehicle forward velocity

Yawing radius λ is defined as positive when yawing center is behind the C.G. of the vehicle.

Yawing Center of Two-Wheel Steering Cars

Steady State Cornering

Figure 2 shows yawing radius of steady state cornering relative to vehicle speed for a two-wheel

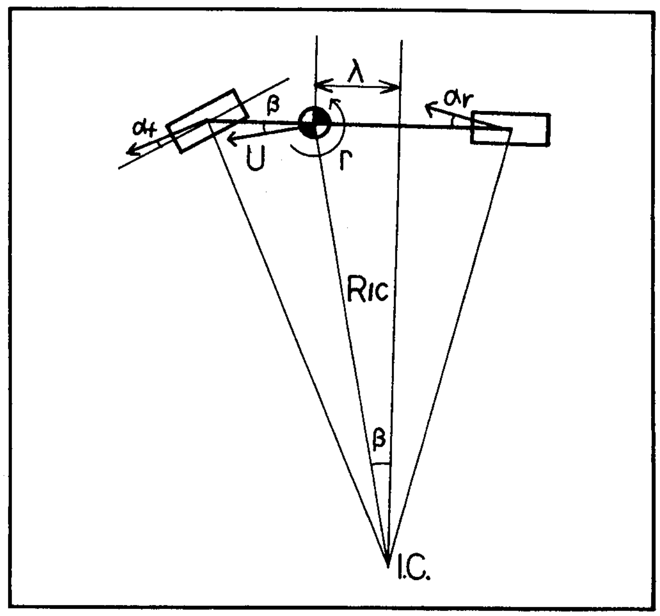

Figure 1. Definitions of yawing center and yawing radius

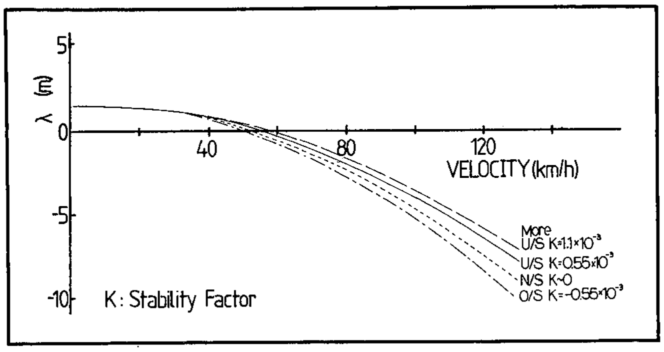

Figure 2. Steady state yawing radius relative to vehicle speed

steering car. The vehicle stability factor was set at four different levels, ranging from slight understeer to slight oversteer. (See Appendix B for full details of vehicle specifications and the stability factor.)

The findings are as follows:

- The higher the vehicle speed, the more yawing center moves forward.
- The more oversteer characteristic the vehicle has, the more yawing center moves forward.
- The vehicle forward velocity in which steady state yawing radius becomes zero (called "Tangent Speed") is higher for greater degrees of understeer.

These findings are considered to be coincident with the normal driver's feeling of yawing center of conventional two-wheel steering cars.

Periodic Steering Input

From the results of computer simulation using a mathematical model with two-degrees of freedom, yawing radius varies from minus infinite to plus infinite. But the driver could feel yawing center somewhere around the vehicle. The reason might be as follows: when yawing radius becomes infinite, whether plus or minus, yaw velocity becomes quite small, so there is no need for the driver to be conscious of where yawing center is. The driver must be aware of yawing center when yawing velocity is rather great.

In this report, yawing center and yawing radius of periodic steering input are represented by those of the time when yaw velocity reaches maximum value.

Figure 3 shows yawing radius of periodic steering input relative to vehicle speed for a two-wheel steering car. The vehicle stability factor was set at the four different levels as mentioned above. The frequency of steering input was selected at two different levels, 0.3Hz and 0.75Hz.

The analytical findings are as follows:

- Yawing center of 0.3Hz steering input has similar characteristic as steady state cornering.
- Yawing center moves backwards according to the frequency change from lower to higher.
- At 0.75Hz steering frequency, the more understeer characteristic the vehicle has, the more yawing center moves forward.

Yawing Radius of Step Steering Input

Figure 4 shows yawing radius of step steering input as follows:

$\delta sw = 0$ for $t < 0$

$\delta sw = \delta o$: const. for $t \geqq 0$

Where

δsw:steering wheel angle

The vehicle stability factor was set at the four different levels as before.

The following characteristics are shown in Figure 4.

- Comparing the steady state level of yawing radius, the more oversteer, the more yawing center moves forward.
- At vehicle speed 100km/h, the more oversteer, the slower yawing radius rises up.
- The higher vehicle speed and the more understeer, the greater the vibrational tendency.

Effects of Four-Wheel Steering System

Effects of Four-Wheel Steering System

Figures 5 and 6 show the effects of a four-wheel steering system on yawing radius in the same conditions as a car with a two-wheel steering system. The

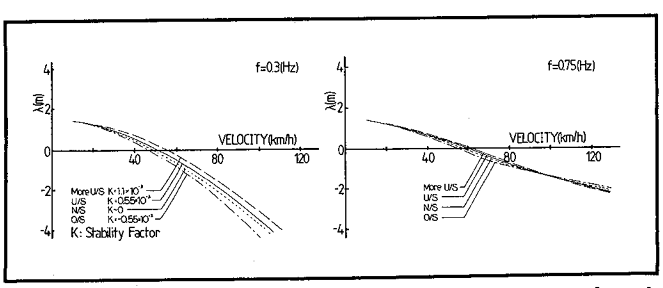

Figure 3. Yawing radius of periodic steering input relative to vehicle speed

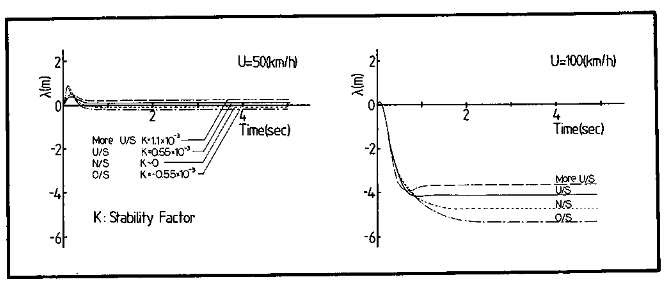

Figure 4. Yawing radius time histories of step steering input

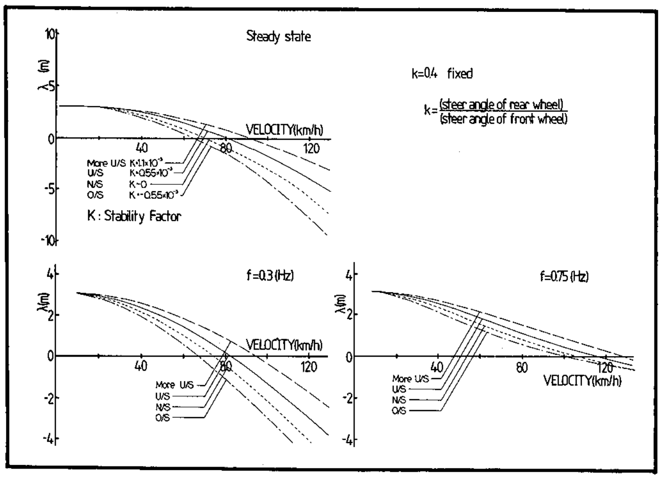

Figure 5. Yawing radius of four-wheel steering car relative to vehicle speed

ratio of rear wheel angle to front wheel angle k was fixed at 0.4 in this calculation. The ratio of rear wheel angle to front wheel angle k is defined as positive when the rear wheels are steering in the same direction as the front wheels. The mathematical model used in calculation is shown in Appendix A.

The following characteristics may be derived from figures 5 and 6:

- By steering the rear wheel in the same direction as the front wheel, yawing center is kept rather more backwards than that of a two-wheel steering car.
- The more understeer, the bigger the observable effects.

Effects of Steer Angle Ratio of Rear to Front on Yawing Radius

Figure 7 shows the effects of steer angle ratio of rear wheel angle to front wheel angle k on yawing radius at vehicle speed 50km/h and 100km/h. The frequency of steering input was selected at three different levels, 0Hz (i.e. steady state), 0.3Hz and 0.75Hz.

The following characteristics are shown in Figure 7.

- The bigger the k, the bigger the effect on yawing radius.
- At 50km/h, yawing radius changes a little by frequency variations. But at 100km/h, a bigger change of yawing radius can be seen by frequency variations.

The discussions in sections 1 and 2 may be summarized as follows:

The concept of yawing center is introduced as one of parameters of vehicle motion which are considered to give rather significant influence on the driver's feeling of steering control. It is shown that yawing center changes with vehicle speed in accordance with the driver's general feeling that vehicle stability changes with vehicle speed, and that adopting four-wheel steering system may make it possible to control yawing center. In other words, to select the proper steering ratio of rear to front k according to vehicle speed and steering frequency may make it possible to keep yawing center in the desirable range.

Correlation Between Yawing Radius and Vehicle Response

Steady State Yawing Radius and Phase Delay of Vehicle Response

Figure 8 shows correlations between steady state yawing radius and phase delay of lateral acceleration and yaw velocity response to steering input at vehicle speed 50km/h and 100km/h. The frequency of steering input was selected at two different levels, 0.3Hz and 0.75Hz for the calculation of phase delay.

The ratio of rear wheel steer angle to front wheel angle k was set at -0.4 to +0.3 and +0.3 to +0.6 for calculations at 50km/h and 100km/h, respectively.

- Phase delay of yaw velocity response to steering input increases slightly as yawing radius λ increases.
- Phase delay of lateral acceleration reponse to steering input decreases remarkably as yawing radius λ increases.
- The point of yawing radius λ exists where phase delay of lateral acceleration response to steering input equals that of yaw velocity.

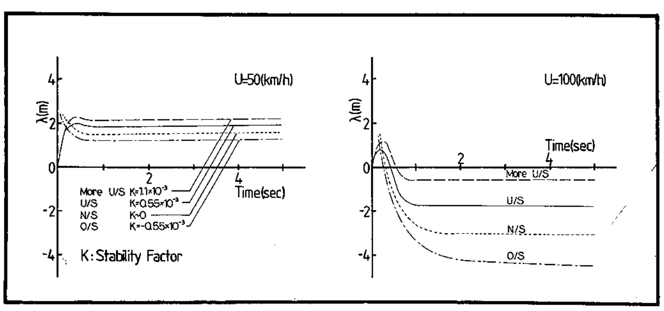

Figure 6. Yawing radius time histories of four- wheel steering car of step steering input

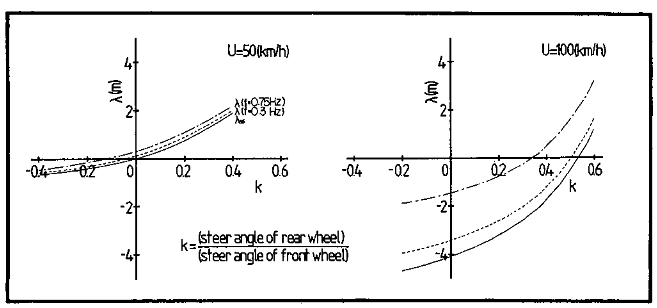

Figure 7. Effects of steer angle ratio of rear to front k on yawing radius

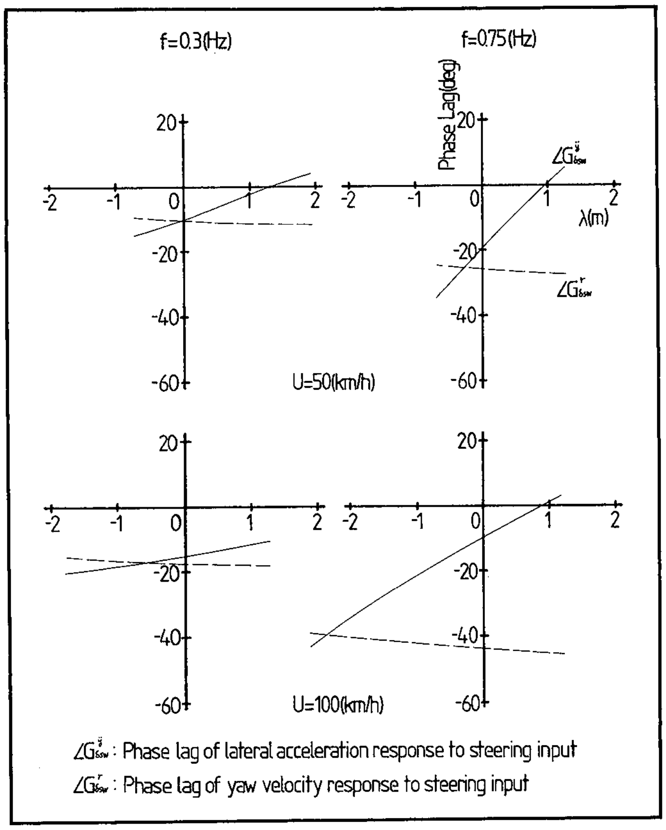

Figure 8. Phase delay of lateral acceleration and ayw velocity response to steering input relative to steady state yawing radius

Steady State Yawing Radius and Phase Difference of Lateral Acceleration and Yaw Velocity Response to Steering Input

Figure 9 shows phase difference between lateral acceleration response to steering input and yaw velocity relative to steady state yawing radius. Parameters used in the calculation are the same as before.

The findings are as follows:

- At the frequency 0.3Hz, the differences become zero where steady state yawing center is a little bit forward of the center of gravity of the vehicle.

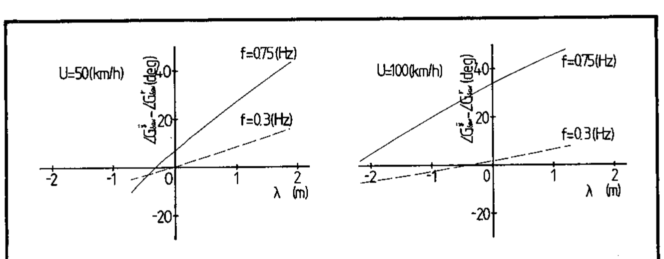

Figure 9. Steady state yawing radius to phase delay difference between lateral acceleration and yaw velocity response

- At the frequency 0.75Hz, the point of steady state yawing radius where phase difference is zero is more forward than that at 0.3Hz.

Yawing Radius of Periodic Steering Input and Phase Difference

Figure 10 shows the correlation between phase difference of lateral acceleration to yaw velocity and yawing radius of periodic steering input as defined in the former paragraph. Parameters used in the calculation are the same as in former paragraphs.

From figure 10, following can be clearly seen:

- To equalize phase delay of lateral acceleration response to steering input with that of yaw velocity has nearly the same meaning as to make yawing radius of the corresponding periodic input zero.

In other words, to equalize the phase delay of lateral acceleration with yaw velocity response to steering input has nearly the same meaning as to make the vehicle slip angle β zero.

Studies on the Cornering Kinetic Energy

Cornering Kinetic Energy

Translational kinetic energy TKE of a vehicle running straight with a constant speed, U, is

$$TKE = \frac{1}{2}mU^2$$

However, when a vehicle is performing a cornering motion and maintaining the same speed, U, energy will be put into the cornering motion, and the cornering kinetic energy, CKE will be (See Reference (3))

$$CKE = \frac{1}{2}mu^2 + \frac{1}{2}Ir^2$$

where

m	:	vehicle mass
U	:	vehicle forward velocity
u	:	vehicle side slip velocity at C.G.
$I = m\rho^2$	:	moment of inertia
r	:	yaw velocity
ρ	:	radius of gyration

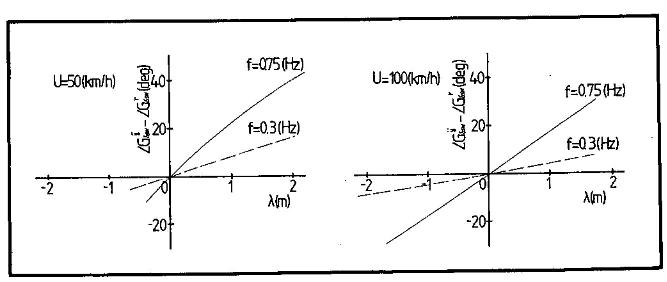

Figure 10. Phase differences relative to yawing radius of periodic steering input

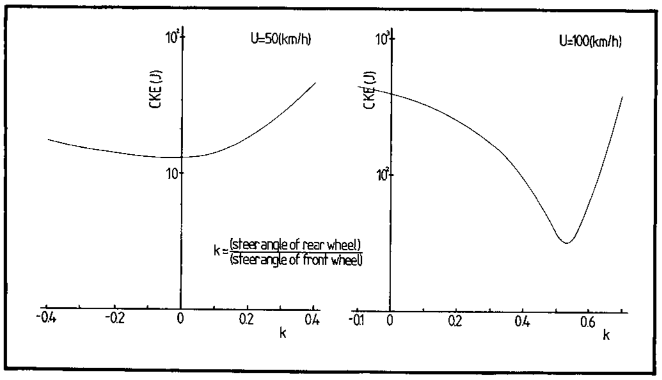

Figure 11. Effects of steer angle ratio of rear to front k on steady state cornering kinetic energy (CKE)

For $|\beta| \ll 1$, the following expression will be possible,

$$\beta = \frac{u}{U}$$

Using these equations, CKE will be expressed with TKE as follows:

$$CKE = TKE(\beta^2 + (\frac{r\rho}{U})^2)$$

Effects of Steer Angle Ratio of Rear to Front on Cornering Kinetic Energy

Steady State Cornering

Figure 11 shows the steady state cornering kinetic energy at 30 degrees of steering wheel angle relative to steer angle ratio of rear to front k. The steer angle ratio of steering wheel to front wheel n was adjusted according to the change of steer angle ratio of rear to front k in order to make the steady state gain equal to that of the two-wheel steering car. (See Appendix B)

The findings are as follows:

> Steady state cornering kinetic energy (CKE) has minimum value at around zero of steer angle ratio of rear to front k at vehicle speed 50km/h. At 100km/h, steady state CKE has minimum value at around 0.53 of steer angle ratio of rear to front k.

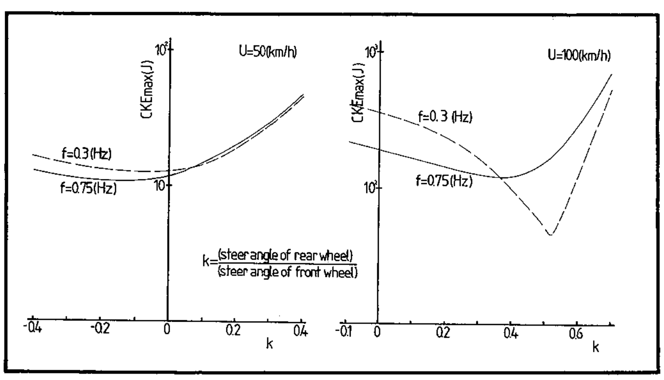

Figure 12. Effects of steer angle ratio of rear to front k on CKE of periodic steering input

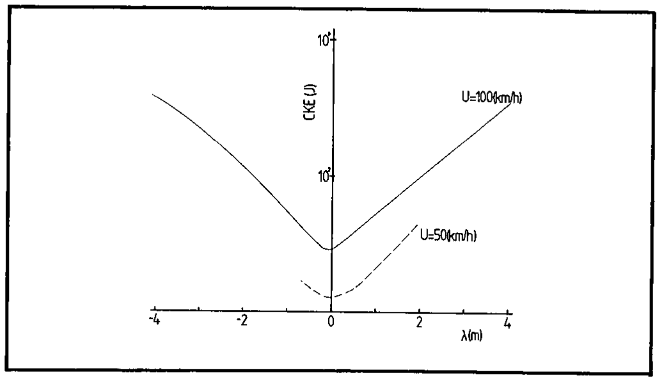

Figure 13. Steady state cornering kinetic energy relative to yawing radius

Periodic Steering Input

Figure 12 shows the cornering kinetic energy of periodic steering input relative to steer angle ratio of rear to front k. Amplitude of steering wheel angle was 30 degrees and the steer angle ratio of steering wheel to front wheel n was adjusted as in the previous calculation.

Actual cornering kinetic energy of periodic steering input varies in accordance with steering input. But in this report, it is represented by the maximum value of cornering kinetic energy.

The frequency of steering input was set at two different levels, 0.3Hz and 0.75Hz.

- The value of steer angle ratio of rear to front k at which cornering kinetic energy takes minimum value decreases as the frequency of steering input changes from 0.3Hz to 0.75 Hz.

Discussions in paragraphs 4-2-1 and 4-2-2 may be summarized as follows:

The value of steer angle ratio of rear to front k at which cornering kinetic energy becomes minimum varies as vehicle speed and steering frequency change.

Correlation Between Yawing Radius and Cornering Kinetic Energy

Figures 13 and 14 show cornering kinetic energy relative to yawing radius. Yawing radius of periodic steering input was calculated based on the definition of yawing radius in paragraph 1-2-2.

The analytical findings are as follows:

- In any conditions of vehicle speed and steering frequency, cornering kinetic energy takes minimum value when yawing radius is zero.

It may be suggested from calculated results that if yawing radius λ is taken as one of the parameters to control steer angle ratio of rear to front k, it may be possible to control the minimum cornering kinetic energy. To minimize cornering kinetic energy may be considered to be a reduction of the cornering load on

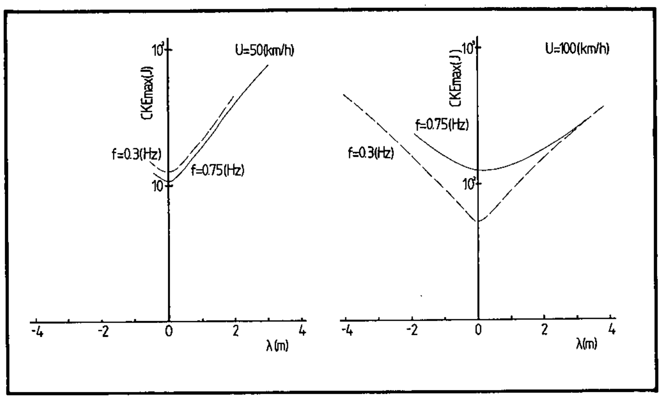

Figure 14. Cornering kinetic energy relative to yawing radius of periodic steering input

tires. And it may make possible to raise the cornering limit.

Conclusion

With the introduction of the concept of yawing center, the effects of a four-wheel steering system were analyzed by computer simulation. The results may be summarized as follows:

(1) Yawing center of two-wheel steering car moves forward as vehicle speed increases.
(2) It is possible to keep yawing center of four-wheel steering car further backwards than that of two-wheel steering car by setting proper steer angle ratio of rear to front k according to vehicle speed.
(3) To equalize phase delay of lateral acceleration response to steering input with that of yaw velocity has nearly the same meaning as to make yawing radius of corresponding periodic input zero.
(4) Selecting the proper steer angle ratio of rear to front k to keep yawing center on the center of gravity of the vehicle may make it possible to minimize cornering kinetic energy.

Discussion

From the point of minimizing the phase difference of lateral acceleration to yaw velocity, and/or from the point of minimizing cornering kinetic energy, the best location of yawing center is on the center of gravity of the vehicle. That is vehicle slip angle β is zero.

However, the normal drivers may not be aware of the location of the center of gravity of the vehicle. So it may not be really significant for the driver that vehicle slip angle β is zero at the center of gravity of the vehicle, i.e. yawing radius λ is zero. What may be very significant for the driver is how far yawing center is from the driver's seat.

As an initial study of these subjects, this paper introduces the concept of yawing center, and reports the results from simulation analysis of four-wheel steering system.

The subject of the best location of yawing center for the driver to control vehicle steering performance is left to future studies.

Effects such as this that take advantage of four-wheel steering system may contribute to the field of active safety for vehicles in the near future.

Appendix A. Mathematical Model

Model of a Vehicle with Four-Wheel Steering System

Equations of Vehicle Motion

Equations of vehicle motion with two-degrees of freedom in a coordinate system fixed to the vehicle center of gravity with the constant forward velocity U and small vehicle slip angle β can be expressed as follows (See Reference (2)):

$$\begin{bmatrix} I\dot{r} \\ m\dot{\beta} \end{bmatrix} = \begin{bmatrix} -\frac{Cfa^2+Crb^2}{U} & Crb-Cfa \\ -m-\frac{Crb-Cfa}{U^2} & -\frac{Cf+Cr}{U} \end{bmatrix} \cdot \begin{bmatrix} r \\ \beta \end{bmatrix} + \begin{bmatrix} Cfa & -Crb \\ \frac{Cf}{U} & \frac{Cr}{U} \end{bmatrix} \begin{bmatrix} \delta f \\ \delta r \end{bmatrix} \quad (1)$$

The definitions of the symbols used in the equation are given in the nomenclature at the end of the appendixes.

Transfer Functions for Steering Response

The right and left members of equation (1) may be Laplace-transformed by using the relations:

$$\delta sw = n\delta f \quad (2)$$
$$\delta r = k\delta f \quad (3)$$
$$\ddot{y} = U(\dot{\beta}+r) \quad (4)$$

Then the transfer functions for yaw velocity, vehicle slip angle and lateral acceleration response to steering angle δ_{sw} can be expressed respectively as follows:

$$\frac{r(s)}{\delta sw(s)} = G^{r}_{\delta sw}(s) = G^{r}_{\delta sw}(0) \cdot \frac{1+Trs}{1+\frac{2\zeta}{wn}s+\frac{1}{wn^2}s^2} \quad (5)$$

$$\frac{\beta(s)}{\delta sw(s)} = G^{\beta}_{\delta sw}(s) = G^{\beta}_{\delta sw}(0) \cdot \frac{1+T\beta s}{1+\frac{2\zeta}{wn}s+\frac{1}{wn^2}s^2} \quad (6)$$

$$\frac{\ddot{y}(s)}{\delta sw(s)} = G^{\ddot{y}}_{\delta sw}(s) = G^{\ddot{y}}_{\delta sw}(0) \cdot \frac{1+\frac{2\zeta'}{wn'}s+\frac{1}{wn'^2}s^2}{1+\frac{2\zeta}{wn}s+\frac{1}{wn^2}s^2} \quad (7)$$

where

$$G^{r}_{\delta sw}(0) = \frac{1-k}{n} \cdot \frac{U}{L(1+KU^2)} \quad (8)$$

$$G^{\beta}_{\delta sw}(0) = \frac{b+ka}{n} \cdot \frac{1+K'U^2}{L(1+KU^2)} \quad (9)$$

$$G^{\ddot{y}}_{\delta sw}(0) = \frac{1-k}{n} \cdot \frac{U^2}{L(1+KU^2)} \quad (10)$$

$$wn^2 = \frac{Cf\ Cr\ L^2(1+KU^2)}{mIU^2} \quad (11)$$

$$\zeta^2 = \frac{\{(Cfa^2+Crb^2)m+(Cf+Cr)I\}^2}{4mI\ Cf\ Cr\ L^2(1+KU^2)} \quad (12)$$

$$Tr = \frac{m(Cfa-kCrb)U}{(1-k)Cf\ Cr\ L} \quad (13)$$

$$T\beta = \frac{I(Cf+kCr)U}{Cf\ Cr\ L(b+ka)(1+K'\ U^2)} \quad (14)$$

$$K = \frac{Crb-Cfa}{Cf\ Cr\ L^2}m \quad (15)$$

$$K' = \frac{kCrb-Cfa}{Cf\ Cr\ L(b+ka)}m \quad (16)$$

$$wn'^2 = \frac{(1-k)Cf\ Cr\ L}{(Cf+kCr)I} \quad (17)$$

$$\zeta'^2 = \frac{(b+ka)^2\ Cf\ Cr\ L}{4(1-k)(Cf+kCr)I} \cdot \frac{1}{U^2} \quad (18)$$

Appendix B. Vehicle Specifications

Basic Specifications

2WS : TWO-WHEEL STEERING SYSTEM
4WS : FOUR-WHEEL STEERING SYSTEM

No	ITEM		UNIT	2WS	4WS
1	VEHICLE MASS	: m	kg	1230	←
2	DISTANCE FROM FRONT AXLE TO C.G.	: a	m	1.01	←
3	DISTANCE FROM REAR AXLE TO C.G.	: b	m	1.50	←
4	YAWING MOMENT OF INERTIA	: I	kg·m²	1820	←
5	WHEEL BASE	:	m	2.51	←
6	STEERING WHEEL ANGLE / STEER ANGLE OF FRONT WHEEL	: n	—	22.0	13.2
7	STEER ANGLE OF REAR WHEEL / STEER ANGLE OF FRONT WHEEL	: k	—	0	0.4
8	FRONT TIRES' CORNERING STIFFNESS	: Cf	N/rad	84200	←
9	REAR TIRES' CORNERING STIFFNESS	: Cr	N/rad	67400	←
	STABILITY FACTOR	: K	sec/m²	0.55×10^{-3}	←

Design Parameters for Equalizing Steady State Response Gain

STEER ANGLE OF REAR WHEEL / STEER ANGLE OF FRONT WHEEL	: k	-0.2	0	0.2	0.4	0.6
STEERING WHEEL ANGLE / STEER ANGLE OF FRONT WHEEL	: n	26.4	22.0	17.6	13.2	8.8

(For Fig. 7,8,9,10,11,12,13,14)

Design Parameters for Changing Stability Factor

STABILITY FACTOR ($\times10^{-3}$)	: K	MORE UNDERSTEER 1.10	UNDERSTEER 0.55	NEUTRAL STEER 0	OVER STEER -0.55
FRONT TIRES' CORNERING STIFFNESS	: Cf	77700	84200	90600	96600
REAR TIRES' CORNERING STIFFNESS	: Cr	73900	67400	61000	55000

(For Fig. 2,3,4,5,6)

Nomenclature

a : Distance from front axle to center of gravity (C.G.) of vehicle
b : Distance from rear axle to C.G.
L : Wheel base (a+b)
m : Vehicle mass
I : Yawing moment of inertia
n : Steer angle ratio of steering wheel to front wheel
k : Steer angle ratio of rear to front wheel
Cf : Front tires' cornering stiffness
Cr : Rear tires' cornering stiffness
U : Vehicle forward velocity
u : Vehicle side slip velocity
ρ : Radius of gyration
δsw : Steering wheel angle
δf : Steer angle of front wheel
δr : Steer angle of rear wheel
K : Stability Factor
r : Yaw velocity
β : Vehicle side slip angle
$\ddot{y}$: Lateral acceleration
λ : Yawing radius
Ric : Instant cornering radius
TKE : Translational kinetic energy
CKE : Cornering kinetic energy

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Operational and Design Features of the Steer Angle Dependent Four Wheel Steering System (Written only paper)

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Abstract

Research on a directional control technique which steers the rear wheels along with the front ones has been reported in a series of papers since the 7th International Technical Conference on Experimental Safety Vehicles.

These fundamental studies have led to the development of a new four wheel steering system that varies the steer direction and angle of the rear wheels, depending on the steering wheel input. Among its features are decreased delay of steering response at high speed and decreased minimum turning radius at low speed.

This paper describes the basic principles of this new four wheel steering system, as well as its design and operation. It also discusses the results of tests of system function conducted under various operating conditions over a range of speeds, to demonstrate the benefits of installing this system in a vehicle.

Introduction

Since the 7th International Technical Conference on Experimental Safety Vehicles, we have reported on a series of studies on a control technique which steers the rear wheels in the same direction as the front ones[1]-[7]. These research efforts found that steering the rear wheels in the same direction as the front ones could reduce the delay in lateral acceleration response of the vehicle to steering input, resulting in more responsive steering characteristics under some conditions.

To achieve a shorter turning radius during very low speed maneuvers, specifically for parking the vehicle, it is desirable that the rear wheels should be steered in the opposite direction to the front ones. In an effort to successfully combine these two requirements for controlling the rear wheels, we proposed a steer angle dependent four wheel steering system, called the "Honda 4WS," that could change the direction and angle in which the rear wheels were steered, depending on how much the steering wheel was turned. Through continued research and development activities, the Honda 4WS has recently been completed technically and introduced into the market. The present report discusses the basic operational mechanism and design feature of this steering system.

Operational Principle

It is known that steering the rear wheels in the same direction as the front ones results in a shorter delay in lateral acceleration response to steering input. This can offer more responsive steering characteristics when the driver is given such tasks as a lane change during highway cruising[1]-[5].

However, always steering the rear wheels this way increases the minimum turning radius of the vehicle. Especially during a sharp turn at low speed, it is preferable to steer the rear wheels in the opposite direction to the front ones. To meet these two conflicting requirements—one at high speed and the other at low speed, we have developed a variable control system that can steer the rear wheels in either the same direction or the opposite direction to the front ones, depending on the operating conditions of the vehicle.

The steer angle dependent Honda 4WS discussed here is a new steering system which can meet the high and low speed control requirements by mechanical devices alone. As is apparent from the typical example given in Fig. 1, this system steers the rear wheels in the same direction as the front ones when the driver turns the steering wheel in a small angle, but if he

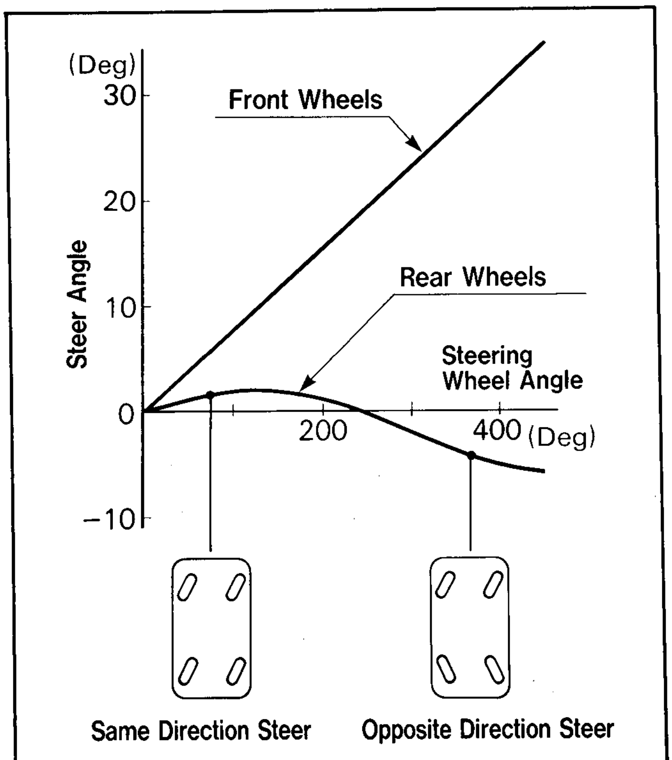

Figure 1. Steering characteristics of Honda 4WS

turns the wheel in a large angle, it steers the rear wheels in the opposite direction.

Major details of how this steering system works in high and low speed ranges are discussed below:

High-Speed Cruise

During a high-speed cruise, the driver usually turns the steering wheel in a relatively small angle. Fig. 2 shows the results of theoretically calculating the relationship between the steering wheel angle and vehicle speed of an automobile equipped with the Honda 4WS when making a steady-state circular turn. The diagram uses lateral acceleration as a parameter, which was selected at 0.2 g, 0.4 g and 0.6 g—typical values which could be used in normal highway cruises.

When the vehicle is going at 50km/h or faster, as indicated in the diagram, the steering wheel angle for making a steady-state circular turn with a lateral acceleration of 0.6 g or less comes within a 100-degree range. In the typical example of front and rear wheel steer angle characteristics given in Fig. 1, the rear wheels are steered in the same direction as the front ones in a speed range above a certain level. This results in a smaller delay in lateral acceleration response of the vehicle to steering input, offering quick, smooth steering response characteristics.

Low-Speed Maneuver

The driver turns the steering wheel in a large angle during short, sharp turns at very low speeds particularly for parking the vehicle, making a U-turn, or turning to the right or left at an intersection of narrow back streets. For such low-speed maneuvers, the Honda 4WS steers the rear wheels in the opposite direction to the front ones, reducing the minimum turning radius of the vehicle.

A steady-state turning radius of a vehicle equipped with the Honda 4WS is virtually equal to that of a

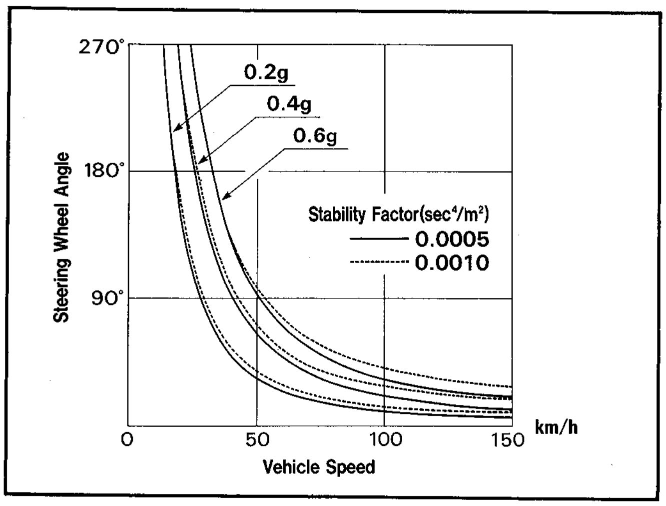

Figure 2. Relation between steering wheel angle and vehicle speed of Honda 4WS car at steady-state cornering

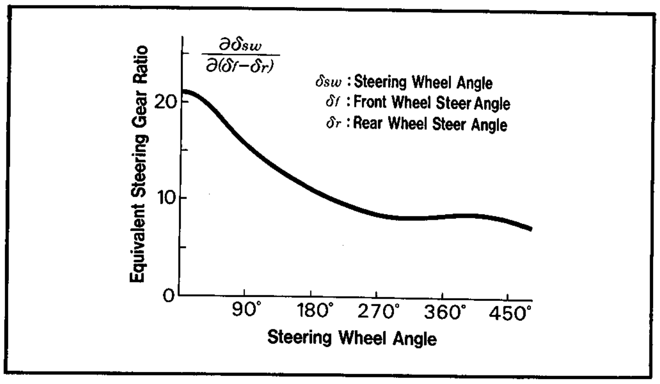

Figure 3. Equivalent steering gear ratio of Honda 4WS

conventional two-wheel steering car, the front wheels of which are steered by the difference between the front and rear steer angles of the Honda 4WS. As shown in Fig. 3, therefore, the equivalent steering gear ratio of the Honda 4WS depends on the relationship—the difference between the front and rear wheel steer angles in response to the steering wheel angle.

In a Honda 4WS vehicle, the direction and angle in which the rear wheels are steered depend on how much the steering wheel is turned. This means that its equivalent steering gear ratio varies with steering wheel angle. In the smaller range of steering wheel angle, the gear ratio becomes slow, while in the larger steering angle range, the ratio becomes more quick. During roughly straight-ahead driving, therefore, the system has a relatively small gain in yaw response to steering input, offering a moderate directional response. As the turning radius of the vehicle becomes shorter, the yaw response gain increases resulting in a larger directional response.

Construction and Operation

System Construction

The Honda 4WS can be constructed by simply adding a mechanical subsystem for steering the rear wheels to the conventional front-wheel steering system. As shown in Fig. 4, this four wheel steering system is essentially comprised of two subsystems, the front and rear steering gear boxes, which are mechanically linked with each other by center steering shaft.

Now let us briefly describe how the system works. When the driver turns the steering wheel, a rack and pinion mechanism in the front steering gear box moves the rack axially. This rack stroke steers the front wheels, and at the same time, turns the output shaft by another rack and pinion mechanism in the front gear box to transmit the steering wheel angle to the rear steering gear box through the center steering shaft. In response to the steering angle thus transmitted, the stroke rod in the rear gear box moves axially to steer the rear wheels through the tie rod.

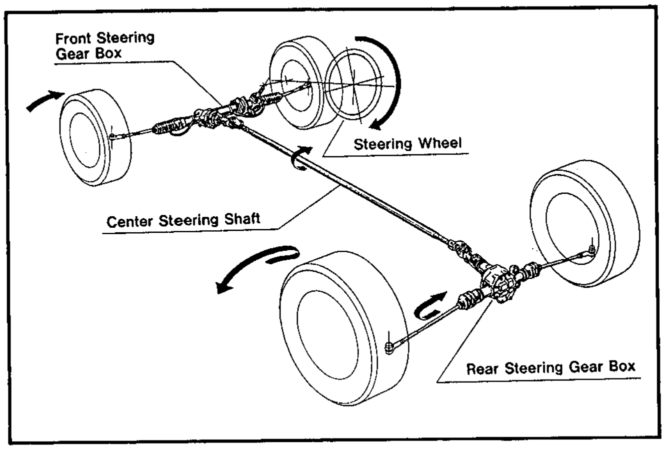

Figure 4. System construction of Honda 4WS

The rear gear box has a built-in variable gear ratio mechanism which changes the direction and ratio of the stroke rod's output stroke to the input, depending on the steering wheel angle. This gives the system steer angle dependent control characteristics. More specifically, when the steering wheel is turned from the straight-ahead position, the rear wheels are steered at first in the same direction as the front ones, but as the steering wheel angle becomes larger than a certain value, the rear wheels are steered in the opposite direction to the front ones.

Construction and Operation of the Rear Steering Gear Box

Fig. 5 shows the construction of the rear steering gear box. A schematic description of its basic mechanism is given in Fig. 6. As shown in these diagrams, the rear gear box uses a combination of two offset shafts, the revolution of which are synchronized by a planetary gear that meshes with a stationary internal gear. Through this mechanism, the orbital motion of axis PP′ around axis OO′ is joined with the QQ′'s movement by axis PP′'s rotation, so that changes in the lateral position of axis QQ′ are transmitted to the stroke rod as its stroke output. Changes in the vertical position of axis OO′ are absorbed by the slider/guide mechanism.

Fig. 7 shows the operation of the planetary gear when the turning angle of the shaft, input into the rear steering gear box, is varied as 90°, 180° and 270°.

Fig. 8 describes the input/output characteristics given to the rear steering gear box by its working mechanism discussed above. This is how the steer angle characteristics of the front and rear wheels in Fig. 1 are obtained.

Test Results of Honda 4WS System

A series of tests were conducted on the Honda 4WS installed in a compact car powered by a 1.8-liter engine to compare its performance with that of a two-wheel steering car of the same basic specifications. Some of the tests findings are discussed below:

Frequency Response Characteristics at High Speed

Fig. 9 shows the lateral acceleration and yaw velocity response of the vehicle to steering input for a steering wheel angle of around ±45°, which is within the smaller steering angle range where the rear wheels are steered in the same direction as the front ones.

The Honda 4WS and the comparable two wheel steering car differed most in phase delay in lateral acceleration: the Honda 4WS vehicle had a shorter phase delay at high steering frequency. Another notable finding is that the change in yaw rate gain with Honda 4WS is less than with two wheel steering up to a high steering frequency.

Lane-Change Test at High-Speed

In this test, the driver was requested to steer the vehicle in such a manner that the position of its center of gravity would follow as closely as possible the

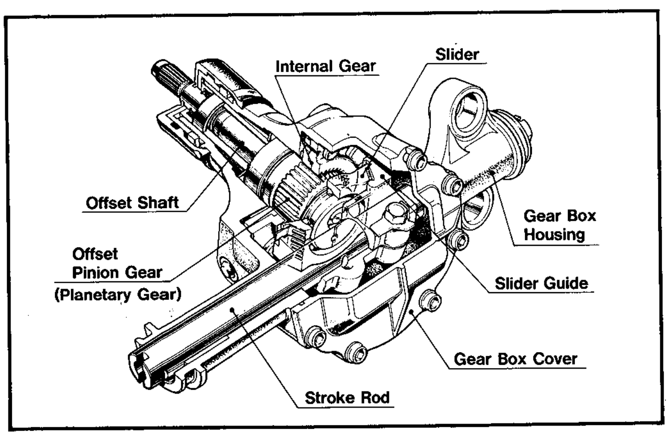

Figure 5. Perspective view of rear steering gearbox

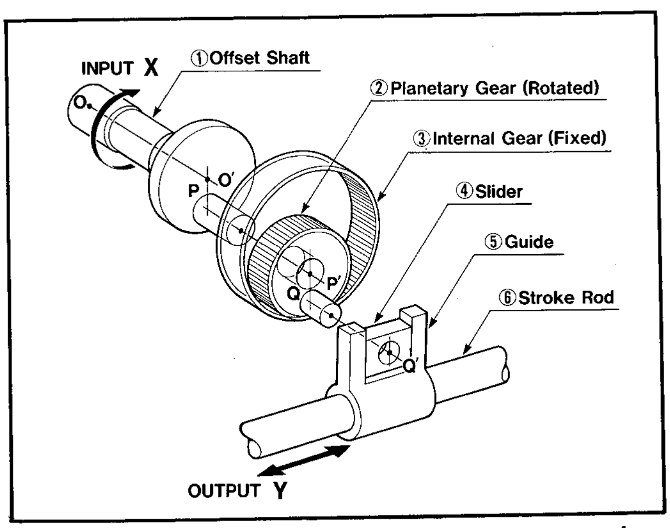

Figure 6. Basic mechanism of rear steering gearbox

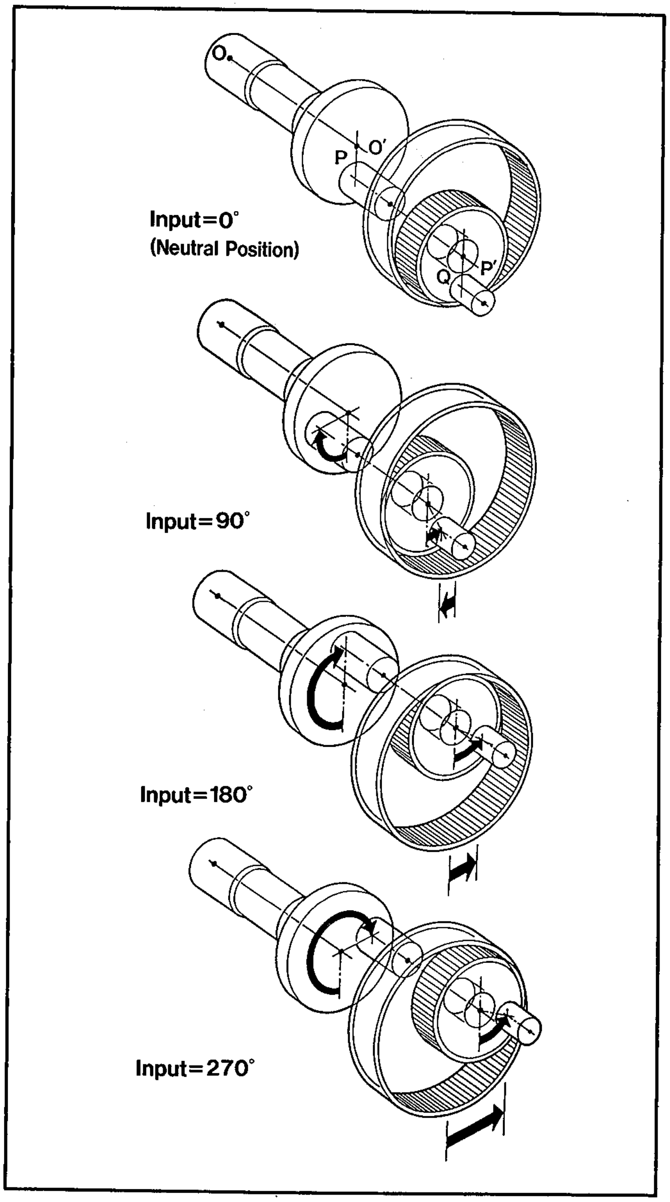

Figure 7. Operation of rear steering gearbox

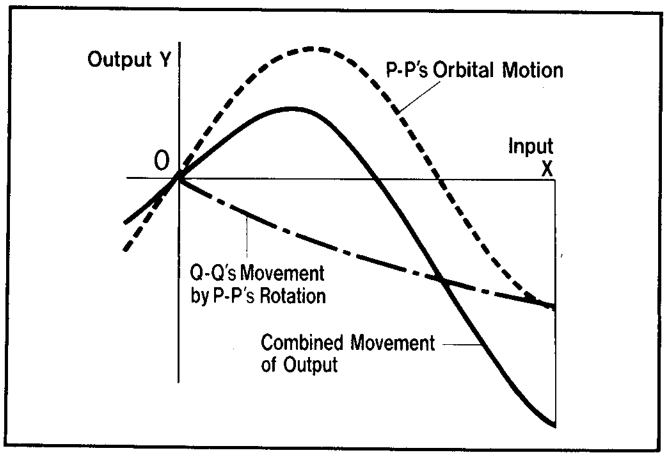

Figure 8. Input/output characteristics of rear steering gearbox

desired path marked with an objective line on the test course surface as described in Fig. 10.

The difference between the two test vehicles in steering response during an example lane change is shown in Fig.11. In this particular example, the Honda 4WS vehicle had smaller amplitudes of lateral acceleration, yaw velocity and roll angle.

Turns at Very Low-Speed

A U-turn test was conducted at very low speed with the steering wheel set at the maximum steering wheel angle. Because of its minimum turning radius, some 0.5 meters shorter than the two-wheel steering car, as shown in Fig. 12, the Honda 4WS vehicle could make U-Turn in a test course 1 meter narrower than the minimum course width required by the other test car. Of course these results will vary slightly with the

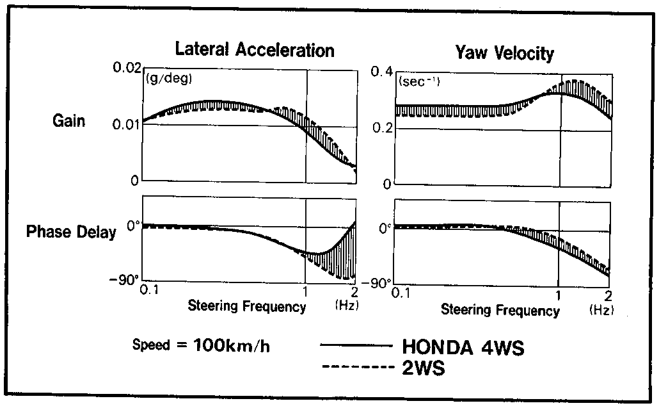

Figure 9. Vehicle steering response characteristics

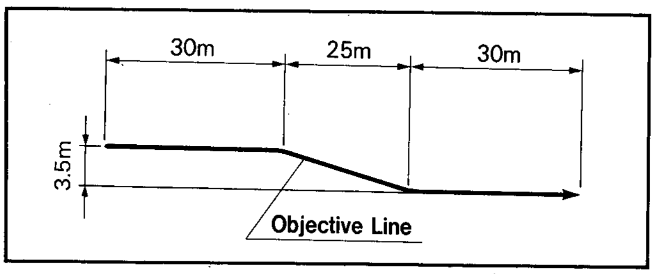

Figure 10. Course layout of lane-change test

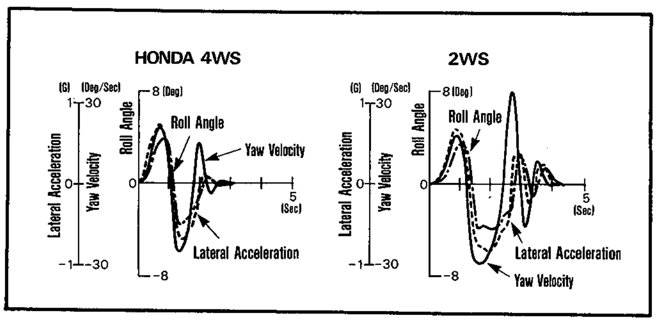

Figure 11. Results of lane change test (vehicle movement)

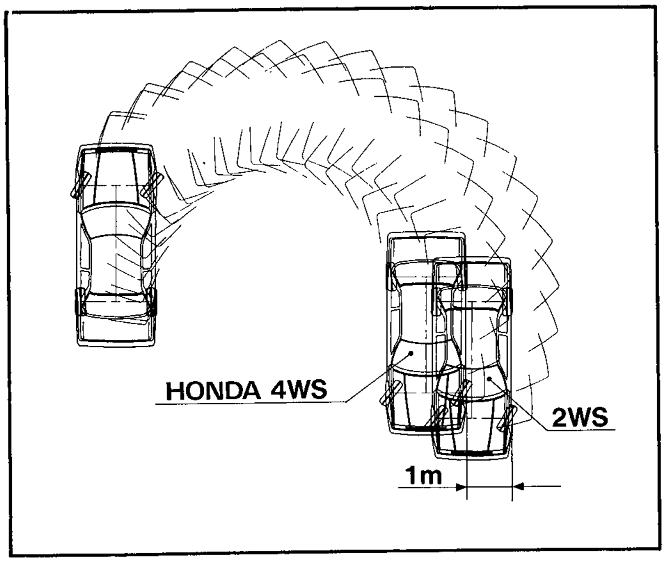

Figure 12. Difference of vehicle path during U-turn

wheelbase, track and other specifications of the vehicle in which the steering system is installed.

Conclusion

In the foregoing sections, we have described the operational and design features of the steer angle dependent Honda 4WS.

The Honda 4WS offers both steering ease at high speed and good maneuverability at low speed by mechanical devices alone. We hope the present report will be helpful for further research and development in this field.

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Vision-Impairing Wear of Windshields (Written only paper)

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Windshields, like other components, are subject to wear during the life of the car; windshield wiper action, scraping of ice and the impact of small pebbles while driving will damage the glass. These defects will cause glare, which may be a contributing factor in night-time accidents. An instrument to measure the scattered light from these imperfections was introduced in a preceding paper at the 10th ESV meeting.

The present paper is a report on measurements of the stray light index of cars on the road in selected regions in Sweden, in the middle of W.- Germany and in southern W.- Germany. The regional differences were found to be significant; values in Sweden being more than twice as high than those in the other two regions. Relationships were found between weather conditions of the regions, parking behavior, mileage and age of the windshield and windshield wear.

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Introduction

Night-time traffic causes a larger number of and more severe accidents than its percentage of total traffic would suggest. The statistics published in 1986 by the Federal Department of Transportation in Bonn, W.- Germany, show that, although night-time traffic represents only 20% of total traffic, accidents at night caused 44% of total fatalities and 50% of pedestrian deaths.

Work at ASSeV has shown that windshield wear may be one of the contributing factors. Windshields deteriorate during the life of the car and reduce visibility at night as a result of glare. One factor is, of course, windshield wiper action: the wiper blades in conjunction with street dust abrade the windshield. In cold climates scratches also occur when ice is removed from the glass. A third factor is less obvious: the glass surface is pitted by the impact of small particles of sand or other materials harder than glass in a form of sandblasting when the car is moving on the road.

The pits (C on figure 1) and scratches (A and B on figure 1) scatter the lights of an oncoming car, which results in a haze and various "tails". This has the effect that the contrast for objects near the light source is reduced and that means that a pedestrian nearly disappears behind the "tails" (Fig. 2).

Stray light is caused by pits half a thousandth of an inch in diameter. Pits of this small size will be hard to detect by inspection of the windshield with the unaided eye. Straylight is also caused by scratches and wiper damage. The light is scattered into "tails" of high intensity which may strongly affect the driver's ability to see objects on the road. The multitude of light sources in city driving and lightened highways may compound the problem since these "tails" flicker, depending on their position distributed on the whole windshield. For most cars, wear may not be as obvious. However mesoptometer tests indicate that a lesser degree of wear can lead to a considerable extension of adaptation, perception and reaction time. It can also be said that twinkling stray light costs energy of concentration. This time and energy is lost for the driver's application of his brakes and the avoidance of crash.

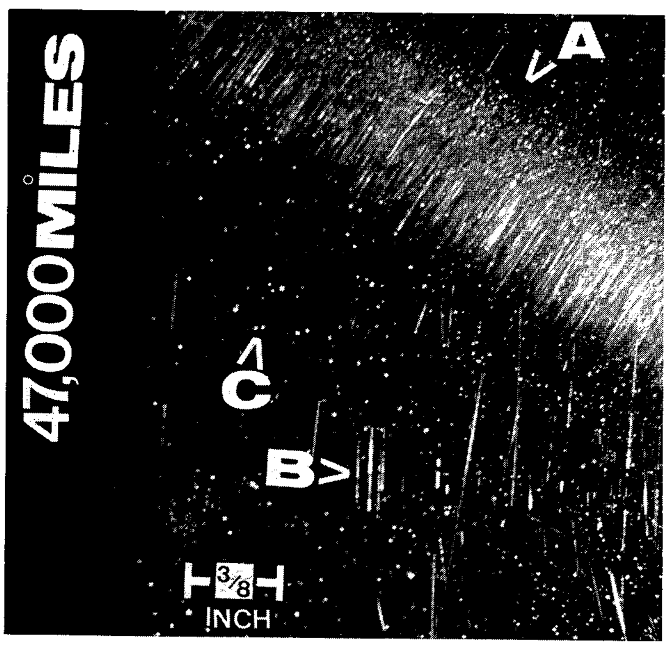

Figure 1. The windshield surface shows three types of wear in dark field illumination

Measurement of Windshield Wear

In order to obtain an objective measure of windshield wear, an instrument called Stray Light Analyzer was developed at ASSeV and presented by A. Timmermann at the 10th ESV conference. Since then the instrument has been made more compact and lighter so that it can be held by hand (Fig. 3). It allows determining windshield wear in the form of a "Stray Light Index" (SLI) and can be used to inspect a large number of vehicles per day. The actual measurement now takes under a second. The unit of the stray light index is, for practical purposes, equal to candela per square meter and lux.

The stray light diagram is reduced to two indices: (1) a "mean" value, which is an average over all intensities and in which the effect of pits predominate, and (2) a "peak" value, which is the maximum value found in the diagram and essentially indicates the amount of wiper damage and ice scratches.

To get an idea as to typical SLI values of cars on the road, measurements were made on a voluntary basis at motor vehicle stations in the local area around Cologne (Fig. 5, points marked by triangles).

The question at which SLI-value a worn windshield actually represents a safety hazard will have to be answered by further physiological tests. Nevertheless measurements and analyses lead to the conclusion that already SLI-values as low as 1 for the average and 5

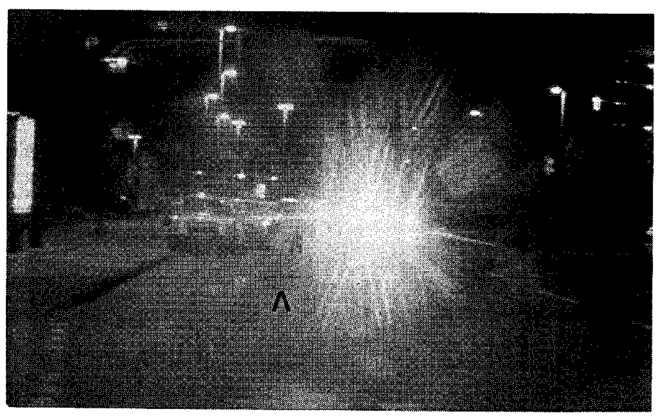

Figure 2. Sight through a worn visor. Both windshields and visors are exposed to the impact of small particles of sand and dust as well as abrasive effects of cleaning efforts. A pedestrian (marked with ∧) nearly disappears behind the "tails"

Figure 3. Stray light analyzer for measuring windshield wear in terms of a stray light index.

for the peak value affect safety. Most convincing, of course, would be an analysis of crash data, because this would represent the ultimate proof of the public benefit of any measures taken such as including windshields in motor vehicle inspections.

Results of windshield wear measurements in different regions

A number of measurements were made on windshield samples mailed in from Sweden (Fig. 5, open circles). The index was twice as high for all kilometer values as was established for the Cologne sample of 404. In order to determine whether test results differ from region to region, an additional sample of 162 cars was measured by an ASSeV team in cooperation with Swedish vehicle inspection stations, which confirmed the result.

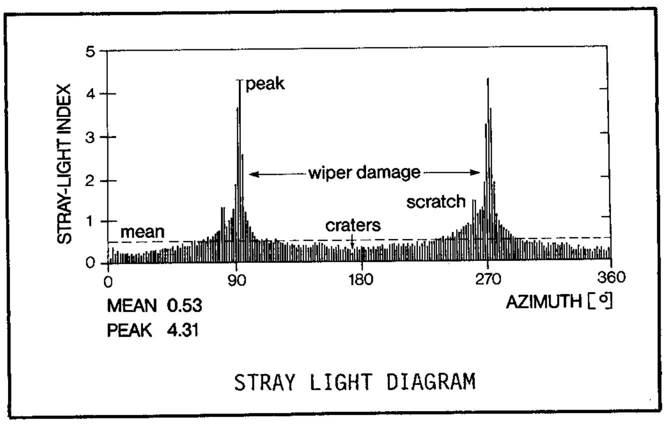

Figure 4. A typical stray light diagram: there is a low intensity level which is almost independent of the angle in the glare azimuth; this is due to pits. The other peaks show up in pairs 180 grades apart. The smaller pair is due to a scratch, the larger pair is due to wiper damage. These peaks can be very intense and then have a blinding effect (see figure 2) even in a relatively well lit city street

With the discovery that there are significant differences between the results for Cologne and for Sweden, the question arose as to whether these are a product of various factors. The most important factors are weather conditions, geographical differences and road conditions all of which increase windshield wear.

Sweden is a snowbelt region with dust-laden winds and often unsurfaced roads. In the middle of Germany you find seldom snow and mostly highways. Ice scraping in snowbelt regions cause scratches on windshield surface; dust-laden winds can act as a powerful abrasive producing tiny craters and, combined with wiper action, fine scratches.

For testing these factors, a sample of 476 windshields was measured in a third region, the area of southern Germany, which has a snow period shorter than Sweden but longer than the area around Cologne. Road conditions are nearly similar.

Significant differences between these three areas can be noticed: Sweden always shows the highest peak and mean values at minimum twice as high than those in the other two regions and a higher increase of stray light over all mileage ranges. This can be interpreted as a combined effect of the above mentioned factors. As expected the area of southern Germany shows higher peak values than the Cologne region, caused by scratches of ice scraping. Histogram b shows higher mean SLI-values for Cologne; this indicates damages by tiny craters.

In order to find other explanations for the difference between the regions, additional items were collected in the area of Cologne and the region of southern Germany. A multiple regression analysis for both regions has been made with the result, that the combination of mainly 3 variables influenced the SLI (mean and peak values): mileage, age of windshield, and parking behavior. Damage to the surface of glass increases linearly with mileage and age of the windshield. Windshields of cars parked in the streets had higher SLI-values than garaged cars. Dust and dirt will collect in the corner between glass and wiper blade and will adhere to the glass and blade when wet. When it begins to rain and wipers are activated abrasive material will be moved over the glass until the layers of dirt are dissolved or rinsed off. Garaged vehicles had also fewer scratches caused by ice scraping than cars parked in the street.

Mileage and age of the windshield are of equal importance to the SLI, while the correlation of SLI with parking behavior is not so strong but nevertheless significant.

Other variables—windshield cleaning habits, roads preferably used by the drivers (city streets, country roads, highways), smoking etc.—correlated less (n.s.).

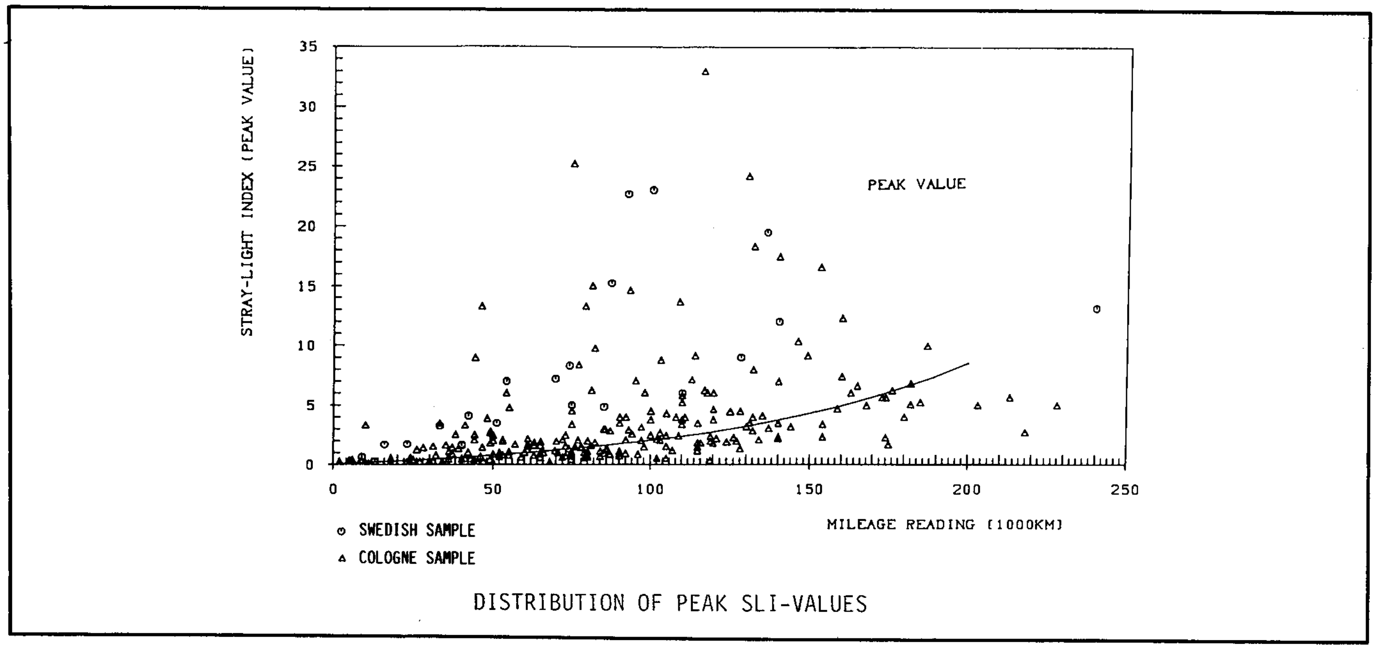

Figure 5. Stray light index as a function of mileage for cars in Cologne (points marked by triangles) and in Sweden (open circles)

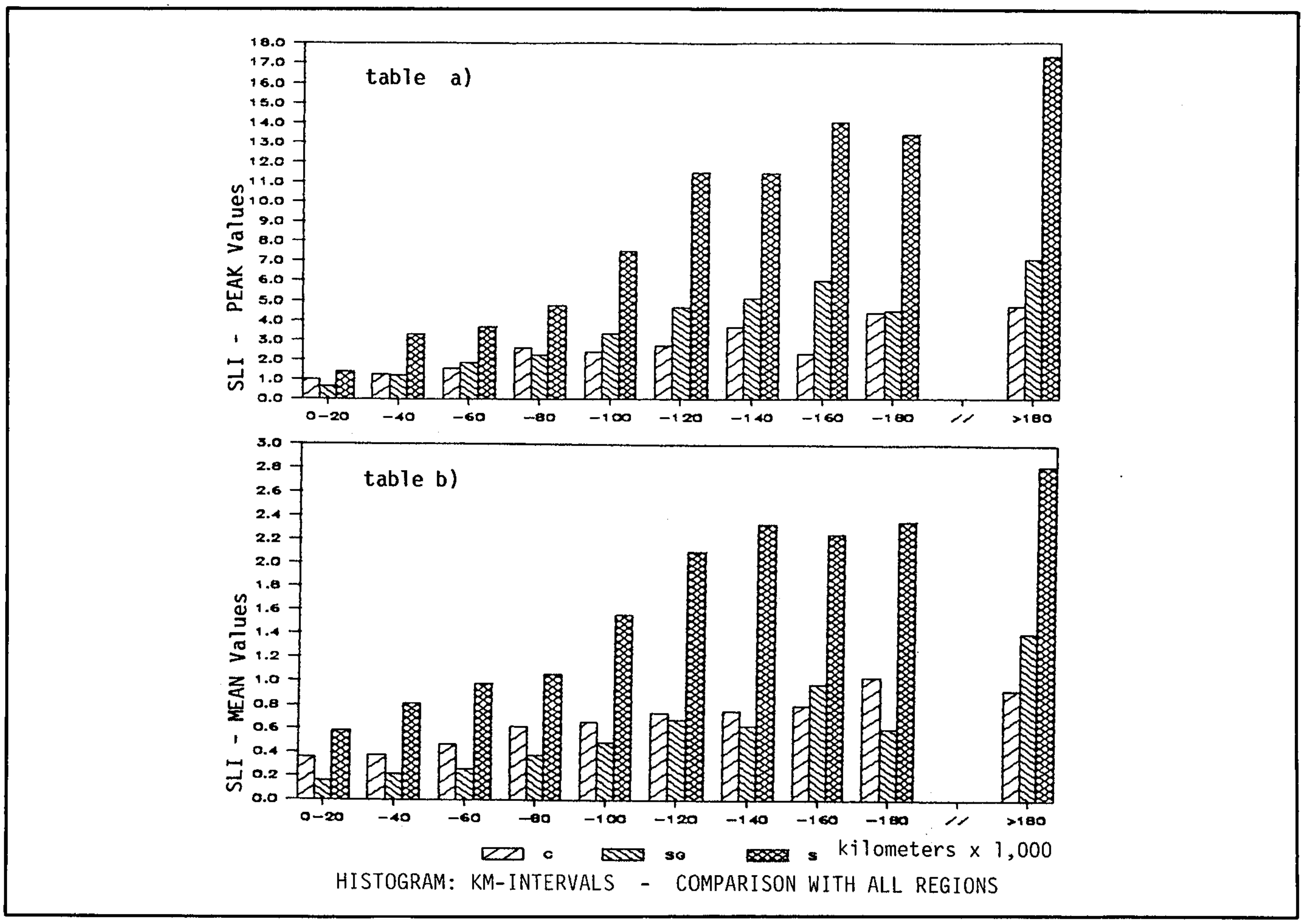

Figure 6. The histograms (a = peak, b = mean) show the comparison of the 3 regions (C = Cologne area in the middle of Germany, SG = area of southern Germany, S = Sweden) in intervals of 20,000 kilometers versus average of SLI. (Note the different scale of SLI!)

One surprising result was a correlation of SLI value with the age of the driver, which was found in a section of southern Germany; younger drivers had a higher SLI value. The explanation may be, that younger drivers do not keep big distances to preceding cars; but the difference may also be that younger drivers are not able to afford more recent models and tend to drive older cars. This sample included several measurements in barracks.

Summary

The worst damages to windshield surfaces, tiny craters as well as scratches, were found in Sweden. Windshields of the Cologne area had fewer scratches but more tiny craters and pits in contrast to the southern region. Regional differences are significant.

Besides mileage and age of the windshield there is another variable which evidently affects wear. Other components are obviously wiper action. Pilot studies indicated a trend of wear influence; and further long term studies are planned on fitting positions of windshields and different types of roads (city streets, highways etc.) preferably used by the drivers.

Acknowledgement: The authors wish to thank Dr. Alwin Timmermann for informative discussions.

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Technical Session Five

Occupant Protection for Frontal Impact

Chairman: Dr. Kennerly H. Digges, United States

Comparison Between the Three-Point Belt and the Air Cushion Evaluation and Discussion of Their Cost-Efficiency Ratio

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Summary

Based on a detailed examination of accidents, the author compares the measured efficiency of the three-point seatbelt with that for a case-by-case evaluation of the air cushion; the result obtained is discussed, firstly as a function of the accidental or technical parameters which govern it and of the perspectives on the wearing of the seatbelt; then taking economic factors into account, as a function of the "cost-efficiency" ratio of these two devices.

Introduction

The aim of this presentation is to establish the advantages and disadvantages of today's two most effective devices for protecting car occupants in the event of an accident over the coming decades; these devices are the air cushion and the three-point fixing seatbelt, and we discuss the results which may be expected.

This subject has been treated several times over the last ten years. However, we believe it is not inopportune to look at the situation again at this point in time where the air cushion technique has passed its first stage of production fitment; meanwhile in many countries the decision to apply this restraining device has not yet been taken.

Our initial approach will be to determine as precisely as possible the protection potential of the seatbelt and of the inflatable bag.

- Then we will discuss the results obtained as a function of the accidental and technical parameters.
- Particular attention will be accorded to the problem of the frequency of wearing of the seatbelt.
- Finally we will treat the economic aspects and will evaluate the cost-effectiveness ratio of each device, either alone or in combination together.

Evaluation of the Protection Potential of the Seatbelt and of the Air Cushion

Accident parameters

We will use those from the enquiry of the "Physiology" Laboratory of the PSA-RENAULT Association. This multidiscipline investigation is based on more than six thousand accidents studied, with the cooperation of the Police, the doctors of the Orthopedic Research Institute from the hospital of GARCHES, and the engineers of the automobile companies PEUGEOT, CITROEN and RENAULT. The geographic zone concerned is the west Paris region; it comprises different types of traffic, town, main road and highway, in proportions which represent a typical average European situation.

The different types of impact are detailed in annex 1, page 2 and summarised in the following table:

	FRONTAL IMPACT	SIDE IMPACT	OTHERS	TOTAL
Victims Front Seat	2951	823	989	4763
Frequency	62%	17.3%	20.7%	100%
Serious injuries non-belted occupants	64%	21.7%	14.2%	100%

One can note the great preponderance of frontal impacts in France, while a mostly highway traffic situation would have given a less highly contrasted distribution (e.g. USA, source FARS):

	FRONT IMPACT	SIDE IMPACT	OTHERS	TOTAL
Deaths	50%	29%	21%	100%

Calculation of the efficiency of the seatbelt

The device concerned is the three-point fixing seat belt with retracting reel. The efficiency is determined by comparing the proportion of "deaths and serious injuries" of the victims according to whether or not the seatbelt was being worn. The calculations are detailed in annex 1.

The following are the results:

	FRONTAL IMPACT	SIDE IMPACT	OTHERS	TOTAL
Seat-belt efficiency	52.6%	45.6%	71.9%	
Distribution of Victims	641	217	142	1000
Protected by Seat-belt	337	99	102	538

Overall efficiency of the seatbelt for this impact distribution is therefore 53.8%

The good performance (52.6%) of the seatbelt in frontal impact is not surprising; the seatbelt was designed for that; its efficiency in severe frontal impact is due to its excellent accommodation to the seated human body with its tension assured by the retractor.

Moreover, the performance of the seatbelt remains good in side impact (45.5%) and better still in the other impacts which comprise rear impacts, rollovers and complex impacts (71.9%).

This high efficiency in non-frontal impacts has surprised the specialists; it is due in part to the fact that the seatbelt eliminates large movement within the occupant space and thus prevents many of the impact injuries against the rigid zones, resulting sometimes from otherwise light impacts. Additionally the seat belt avoids ejection which continues to occur, in spite of "anti-burst" doorlocks, and of which we know the statistically disastrous consequences.

Evaluation of the air cushion efficiency

This is the passive retention system comprising both the air cushion and a sheet-metal knee bar which guarantees the correct location of the occupant's lower body.

We have not found in the literature facts which allow us to determine in a pertinent manner the efficiency of the air cushion as a function of the parameters characterising the geometry of the impa, and their severity. We have therefore evaluated it on the basis of the APR enquiry by examining case by case, for the non-belted occupants, if the driver's or passenger's bag appeared on the anticipated trajectory of the occupant, taking into account the known characteristics of each impact.

If the occupant meets the airbag, we have arbitrarily decided that the protection afforded is 100% in frontal impact whatever its severity without taking into account the limits linked to intrusions into the occupant space or of the air cushion's own protection capacity; the only exception made was for extremely severe impacts where the speed difference exceeded 44 M.P.H.

In the case of side impact, the risks of ejection, intrusion or sliding on the bag are much greater than for frontal impact: we have therefore evaluated the seatbelt efficiency for the occupants in the car being struck with respect to the various positions of the striking vehicle; we have also attributed this same efficiency to the bag each time it is impacted by the occupant.

We should note that the efficiency of the cushion as defined in this case-by-case evaluation is comparable to the values established in the Literature (notably BLIN and ROMEO, ref. No. 7) as well as with the results of our own tests (annex 3).

This evaluation is detailed in annex 2. The results are as follows:

	FRONTAL IMPACT	LATERAL IMPACT	OTHERS	TOTAL
Efficiency of airbag	76%	17%	14%	
Distribution of victims	641	217	142	1000
Protected by the airbag	487	37	20	544

The overall efficiency of the inflatable bag is therefore evaluated at 54.5% in this optimistic hypothesis.

Comparison "seatbelt—air cushion"

We have found an efficiency of 53.8% for the seatbelt and 54.4% for the air cushion. In other words the efficiency potential of the two types of restraint appears equivalent in the accident conditions of the APR enquiry and that they are each capable of preventing at least half of the death or serious injury victims provided that they are each deployed 100%.

Discussion of These Results

Comparison with previous other results

D. Huelke has made the same type of analysis in the USE in 1981: those results were of the same order; they corresponded, for the sum of serious injuries and deaths, to an efficiency of 55.2% (in place of 53.8% for the same three point seatbelt, and 48.9% (in place of 54.4%) for the air cushion.

Effect of accident distribution by type of impact

We note that the airbag is superior to the seatbelt in frontal impact, due to its 76% potential instead of 52.6%: our hypothesis of total efficiency in case of contact with the air bag impact expresses the high performance which we expect of the inflatable bag due to its physical characteristics; i.e. considerable contact surface and high energy absorption. But vice versa the seatbelt is considerably ahead for side and other type of impact (45.5% and 71.9% against 17% and 14%) because, in contrast to the airbag without seatbelt, it guarantees restraint in all senses.

It is therefore normal that in the context of traffic in which the highway dominates the comparison between the two restraint methods is modified in favour of the seatbelt.

For the efficiencies evaluated in this investigation, and for an impact distribution of the USA type, the overall efficiency potentials become,

—54.6% for the three-point seatbelt

—45.8% for the air cushion,

which is close to the HUELKE study.

Prospects for progress

As we have stated, the bag and the seatbelt have a potential for protection which at best spares just over half of the serious non-restrained victims.

This result is not to be dismissed; it is, however, not enough. Is further progress possible? We know that the vehicle manufacturers are researching the subject: protection in side impacts; greater structure resistance in the case of more severe impacts, allowing a better exploitation of a high-performance restraint, be it bag or seatbelt. These research themes are, however, outside the scope of this presentation.

Combining the air cushion with the seatbelt

Remaining within the domain of restraint methods as such, we will obtain a maximum evaluation of their possible efficiency by combining their effects:

	FRONT IMPACT	SIDE IMPACT	OTHERS	TOTAL
Maximum efficiency %	76	45.5	71.9	
Population concerned	641	217	142	1000
Protected	487	99	102	688

The fact of combining the two methods of restraint allows the addition of their basic potential, close to 54%, a very significant supplement which lifts their overall efficiency to 69%

Reservations concerning the air cushion

We cite:

- a risk of inefficiency linked to the severity of the impact: if due to reasons of intrusion the upper limit of protection offered by the air bag would be situated at a speed differential of 41 M.P.H. instead of the 44 M.P.H. we concluded, the frontal impact efficiency of the bag would be four points reduced; also its global efficiency would drop to 52% for the case of the APR investigation, and to 43% in the case of a highway-dominant type circulation of the US type.
- Risks linked to the loss of reliability of the device after several years; to the non-conventional positions of the occupants; to multiple impacts.

Reservations concerning the seatbelt

Injuries can be caused by the seatbelt; generally light, they have been taken into account in the efficiencies measured by the enquiry; the principal risk, submarining is eliminated in the front seats of recent cars by a correct installation in the vertical sense, and better still by the mounting of the anchorage on the seat.

The Rate of Wearing of Seatbelts

This is the parameter which finally governs the efficiency of the seatbelt.

The rates of wearing

They vary very much from one country to another, from one type of road to another, from one year to the next, as a function of three principal parameters:

- **the legislation:** voluntary or obligatory wearing. The decision to make belt-wearing mandatory systematically increases the rate of belt wearing in a very significant manner, but the results are only durable if the other factors are taken into account.
- **checking,** if the wearing is mandatory, is efficient insofar as the user perceives it is probable and not only exceptional; a high level of financial penalties does not compensate for too infrequent checks.
  High rates of wearing are obtained in countries where checking is efficient.
- **user information,** on the usefulness of seatbelt wearing. The high and durable rates of belt wearing are also obtained in the countries where the stages of implementation of legislation have been accompanied by information directed at the users.
  Northern European countries achieve wearing rates superior to 90% because they have acted simultaneously on the three parameters which govern belt wearing rate.
  The fig. 1, p. 14 taken from TORE VAAJE (ref. 5) clearly illustrates this efficiency achievement by taking the example of the F.R.G. and Britain.

The importance of user information

Convincing the users is the principal parameter. Many users believe however that the seatbelt aggravates injuries when there is crushing of the occupant space, by downing or by fire: the statistics prove that this is false. The belief is that ejection will save them in case of accident, where in reality it multiplies by five the risk of death. They do not know the role and the usefulness of retention which, we have seen, reduces today by a factor of more than two the risks of injuries or death.

In order that the users wear, and continue to wear the seatbelt, it is necessary that far from fearing it, they should know it is their best safeguard. Such information can be understood and accepted even by those who don't much like being "tied in".

It is certain that an information campaign over a few weeks is not sufficient to transmit such a message; the efficiency comes from an action which is explanatory, clear, persistant and concerted with the regulatory actions which express the commitment of the Public Authorities.

The proof of this is that the rate of belt wearing today reaches 95% in the countries where the message has been understood, and where the Public Authorities and their representatives on the road have given the example of wearing and making worn the seatbelt.

Passive seatbelts

The passive seatbelts appear as a variant almost as efficient of the "three point", and allows the establishment of a higher wearing rate without obligation or sanction (RABBIT/VAN DYKE 1982).

This device has the advantage of eliminating the voluntary act of fixing the seatbelt; but it in no way eliminates the constraint and the annoyance linked to the presence and the movement of the belts; we do not see how the users who complain of this aspect of the three-point seatbelt would more readily accept it in the case of the passive seatbelt; it is however just these people who must be convinced.

Improving the comfort of the use of the three-point seatbelt

This appears to use a more interesting approach. The retractor reel represented a considerable step forward in the comfort and the quality of seatbelt wearing; it increased in a decisive fashion the wearing rate and the efficiency levels (from 45% to 54% in the APR enquiry). The seatbelt attachment linked to the seat which is widespread today is a new step forward by facilitating the connection and increasing efficiency.

Other improvements are being studied or are available but not yet in general use; we would mention retractors which control the tension, the upper diagonal attachment variable in height which improves comfort; the blockers or retractors which noticeably increase the performance, particularly in frontal impact.

The potential for improvement of three point seatbelts is far from being exhausted at present.

The Economic Factors

Cost of seatbelts

The cost of front seat equipment comprising three-point seatbelts with retractors, the lap belt fixings on the seat and an adjustable upper fixing is stabilised in France at about $60 customer price.

Cost of the inflatable cushion

The "air cushion" in the steering wheel is available as an option in the USA; it is offered presently at prices varying from $800 (driver only) to $3600 (driver and passenger).

The manufacturers have announced that the price of a steering-wheel bag would be reduced to around $300 if they were mounted in full-scale production.

A complete front seat passive equipment of air cushion with a knee-retaining panel is evaluated by our analysts at $2200 approximately, taking into account the adaptation of the vehicle.

We estimate that eventually for high-volume production such equipment could at best stabilise at around $800.

An air-cushion device therefore costs today from twenty to thirty times more than a three-point seatbelt; eventually this ratio will come down to ten to fifteen times approximately.

Evaluation of the social cost of the victims of traffic accidents

The statistics for France in 1986 are as follows:

	Unit cost	No. of front seat victims	Total
Lightly injured	9 500 F	104 379	1.0 thousand million F
Seriously injured	145 000 F	30 946	4.5 " million F
Deaths	1 600 000 F	5 975	9.5 " million F
Total			15 " million F

or again relating the total cost to the number of serious victims (serious injuries and deaths):

Serious injuries	$ 67877	36 921	$ 10^6	2500

The figures on the number of victims were supplied by the French police. The source for the costs is the SETRA "Circular relative to the calculations on the effectiveness of road investments" of the Ministry of Equipment—March 1986.

Reduction of overall accident costs attributable to the seatbelt or to the airbag

The preceding economic statistics correspond to a wearing rate observed in the APR enquiry (α = 55%) and to the efficiency which we have just calculated (λ = 53.8%).

Without a means of retention (zero wear rate), the number of victims and the calculated prejudice would have been increased in the ratio $1 / (1 - \lambda\alpha)$, or 1.42.

This results in the following table for the reduction in financial costs obtained by means of the seatbelt:

Rate of wear	0	55%	100%
No. of victims	52436	36921	24226
No. saved by the seatbelt	0	15515	28210
Reduction of financial costs (millions of dollars)	0	1048	1908

In the conditions of the APR enquiry the seatbelt has therefore saved about 15,500 serious injuries or deaths, but it would have saved 13,325 others if it had been worn by all vehicle occupants. The costs for the society has been reduced by $1050 millions; but a 100% wear rate would have given a further cost reduction of $860 millions, making a total of $1910 millions.

Reduction of overall accident costs attributable to the inflatable airbag

Starting from the hypothesis of the zero seatbelt wearing rate, the number of serious victims would have been 52,436 resulting in a cost of $3550 millions. Equipping the total vehicle park with inflatable airbags would have reduced by 54.4% the number of these victims (i.e. 28,525 victims saved) as well as the corresponding costs to society (i.e. $1930 millions).

Economic assessment of the restraining methods

Against these financial penalties and their possible reductions is set the following annual investment for fitting new cars.

	Cost of one equipment ($)	No. of vehicles (millions)	Total
Seatbelt	$60	1.91	$ 115 millions
Air cushion in 1987.	$2200	1.91	$4200 millions
Air cushion: full scale production	$800	1.91	$1530 millions

The overall economic assessment for these methods of restraint is therefore the following:

	Cost of one equipment	Reduction of costs to society
Seatbelt:		
(55% wear rate)	$ 115 millions	$1048 millions
(!00% wear rate)	$ 115 millions	$1908 millions
Inflatable airbag:		
(1987 cost)	$4200 millions	$1930 millions
(eventual cost)	$1530 millions	$1930 millions

The case of the seatbelt is extremely favourable. The seatbelt at the 55% wearing rate noted in France in the APR enquiry saves seven times its own cost; it is socially positive starting from 8% wear rate, and we must not forget that the essential damage resulting from accidents, that is, the pain and death, is not of an economic nature and has not been taken into account in these evaluations.

The situation is less favourable for the inflatable bag; it does not repay its cost at today's prices.

Additionally, while the seatbelt is fitted to the world vehicle part and each increase in its wearing rate is immediately and in direct proportion translated into a number of victims saved, the same is not true of the airbag; its action will be different in function of its rate of application to the part; we will have to wait at least four years to obtain only half of the efficiency which is eventually expected of it.

Effectiveness of a Restraining Device Considered as Complementary to the Other

It is clear that the seatbelt is the desirable complement to the cushion: the non-belted occupant of an inflatable bag equipped car has no effective restraining means in the case of anything other than a frontal impact.

Reciprocally, it is not without interest to fit a bag to a vehicle where the occupant is belted. We have seen in passing that the protection potential of a single means of restraint is of the order of 54%, but the combination of the two rises to 68.8%.

We are therefore going to evaluate the economic cost of adding the seatbelt to the cushion as equipment for the whole vehicle part; then that of the inflatable bag added (for example, as an option) to a park equipped with seatbelts with a given wear rate.

Economic assessment of the seatbelt. used as a complement of protection in a part supposed as completely equipped by the inflatable bag.

This assessment is made as a function of the rate of wear of the seatbelt.

Rate of wear	0	25	50	75	100
Persons saved by the bag in non-frontal impact	57	57	57	57	57
Persons saved by the seatbelt if worn alone, in non-frontal impact	0	50	100	150	201
Gain by the seatbelt (non-frontal) in %	0	38	77	115	144
Overall efficiency %	54.4	58.2	62.1	65.9	68.8
Benefit due to the seatbelt (millions of dollars)	0	128	255	366	512

The seatbelt, used as a complement to the inflatable bag, increases the overall efficiency from 54.4% to more than 60% for modest wearing rates of the order of 50%; in these conditions the cost of the seatbelt equipment ($115 millions for 3 points, $60 millions for lapbelt) is already more than twice recovered.

Economic assessment of the inflatable bag used as a complementary protection to the three-point seatbelt.

We will suppose the bag is fitted 100% to facilitate the calculation because benefit of the bag and its cost are one and the other proportional to its equipment level such that its cost/efficiency ratio is independent of it.

Rate of wear	0	25	50	75	100
Gain seatbelt/bag in non-frontal	0	38	77	115	144
Bag efficiency %0	544	544	544	544	544
Total bag + belt	544	582	621	659	688
Total belt alone	0	134	269	403	538
Gain of bag %o	544	448	352	256	150
Benefit of bag (millions of dollars)	1930	1590	1250	909	532

We conclude that the contribution of the bag on the level of protection compared to the seatbelt is by no means negligible; for a belt wearing rate of 75% for example, the seatbelt efficiency alone is 40%, while the overall efficiency with the bag as complement increases to 65%.

However, the cost-efficiency ratio is less good: at its present price (4200 million dollars annual equipment cost) the bag does not amortise a quarter of its cost, for a mediocre 75% rate of belt wearing even for the eventually optimistic price which we estimated ($1530 million) the assessment would only be favourable if the seatbelt was not worn at all.

It remains to be said that the economic assessment is not everything and that the inflatable airbag is without doubt, after the seatbelt, one of the most capable devices for saving human lives.

Conclusions

The three-point seatbelt is today the best-performing restraint method; its simplicity and low cost provide it with an exceptional efficiency.

With a view to improving automobile security, all which is humanly possible must be done to increase the wearing rate. Several European countries have understood this and have shown an example by applying effective methods:

- obligatory wearing: commitment of Public Authority
- obvious check-in-use: appeals to civic duty—and fear of the police.
- persistent information on the effectiveness of wearing it.

In these countries—Germany, Great Britain, Sweden—the wearing rate exceeds 90%. Other countries can do it. Given a 59% efficiency in the APR enquiry where deaths are concerned, and evaluating 100,000 the number of front-seat deaths in the world, while supposing a 50% belt wear rate, we note (fig. 2) that the seatbelt effectively saves 42,000 people each year and would save another 42,000 if it was worn 100%.

The inflatable bag is an effective device of which the protection potential is high in frontal impact. It has low performance in impacts other than frontal, it cannot in any fashion really replace the seatbelt.

It constitutes at present a complementary restraint device which is being developed very naturally as an option, particularly for the driver's position.

Used however as the sole means of restraint for the two front occupants it will be less efficient than the three-point seatbelt and five to ten times more expensive for the driver alone, ten to twenty times more expensive for the two front occupants.

Equipping the vehicles registered in France with inflatable bags would represent today a supplementary annual investment of four thousand two hundred million dollars, or to put it in another way, about the cost of nine hundred kilometres of motorway; it would still be one thousand five hundred thirty million dollars at the large-scale production rate, equivalent to four hundred and fifty kilometres of motorway.

Knowing that choices must be made between priorities, we will conclude by saying that the principal urgency appears to be to make known and understood the usefulness of the seatbelt in order to bring about a high use rate everywhere.

Efficiency of a Restraining Device

(Seatbelt or Air Cushion)

Legend : V Number of victims observed annually in a country

λ Efficiency of restraining device

α Rate of wearing (seatbelt) or of equipment (air-bag)

e Number of victims really saved annually by restraining device: $e = \dfrac{V\alpha\lambda}{1 - \alpha\lambda}$

E Number of victims who really would have been saved annually for a wearing rate or equipment rate of 100%

$$E = \frac{V\lambda(1 - \alpha)}{(1 - \alpha\lambda)}$$

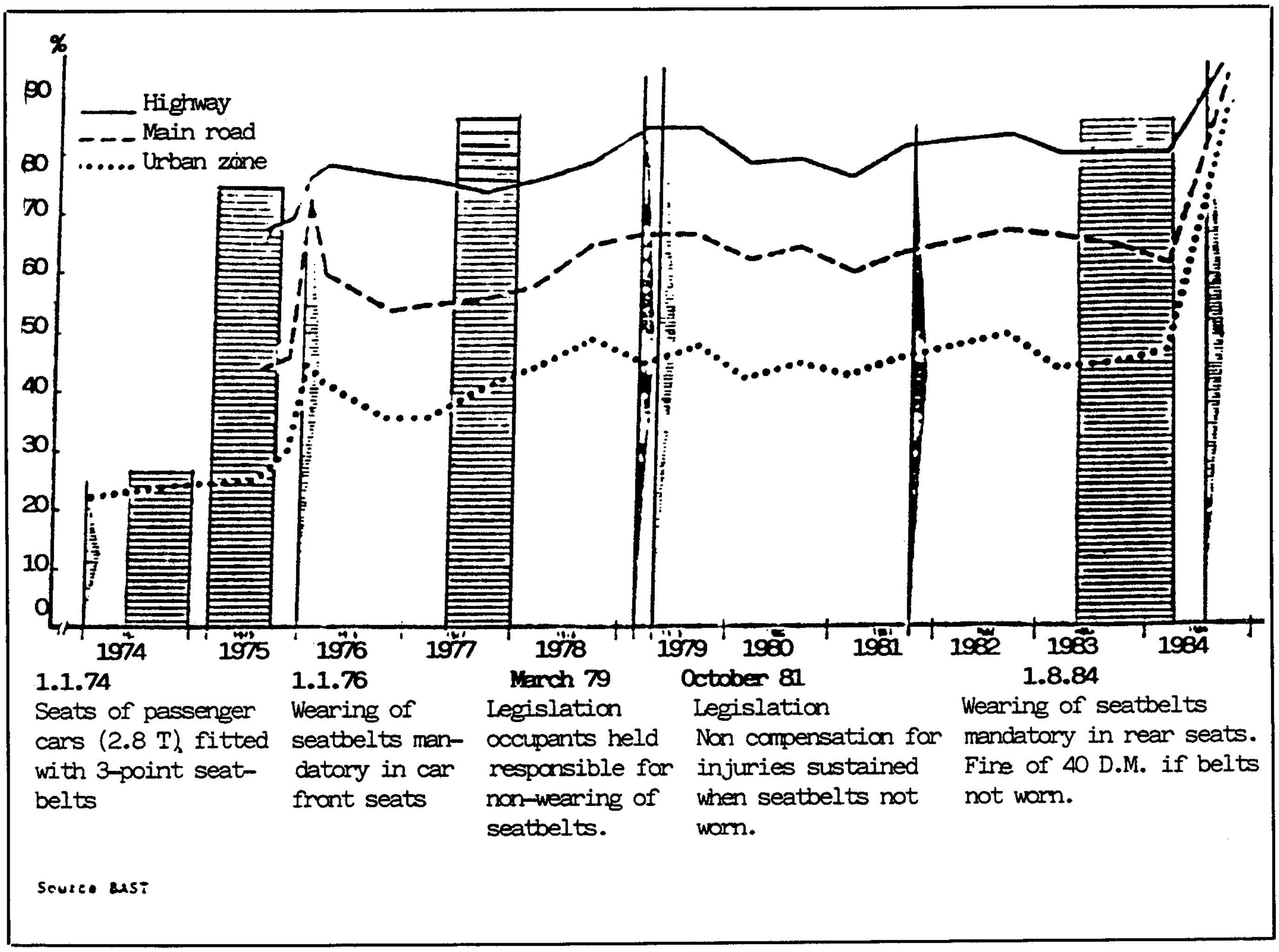


Figure 1. Development of seatbelt wearing in F.R.G.

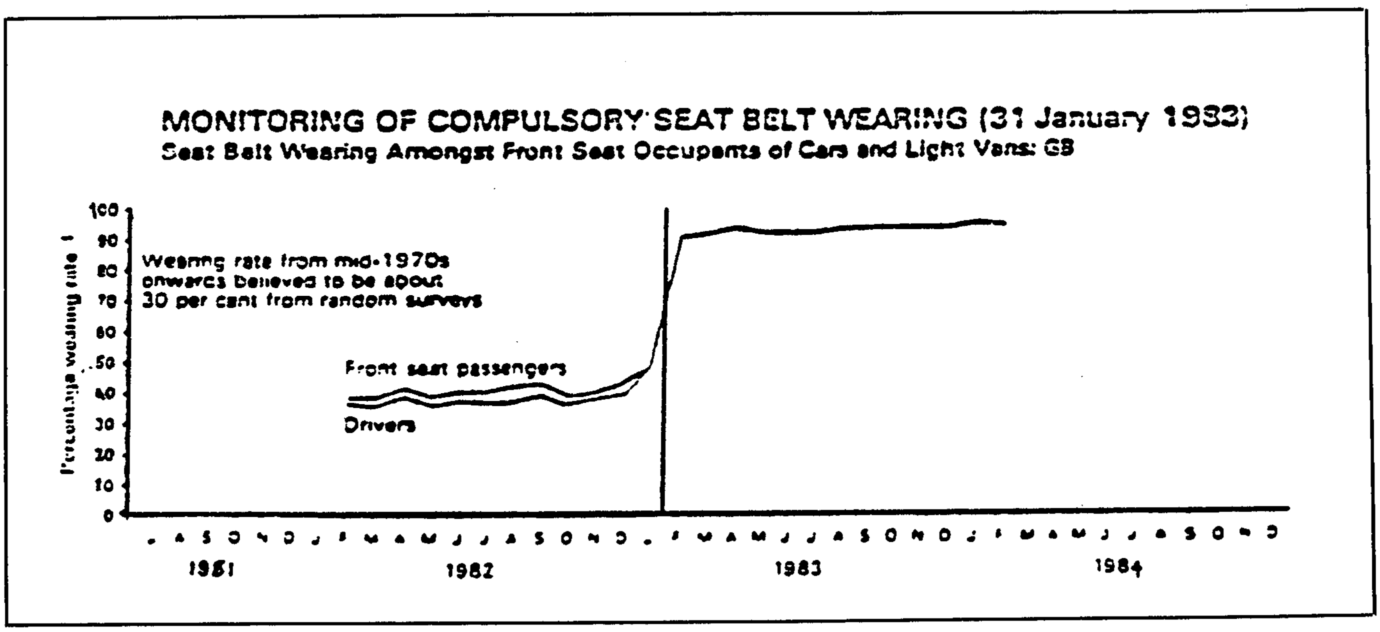
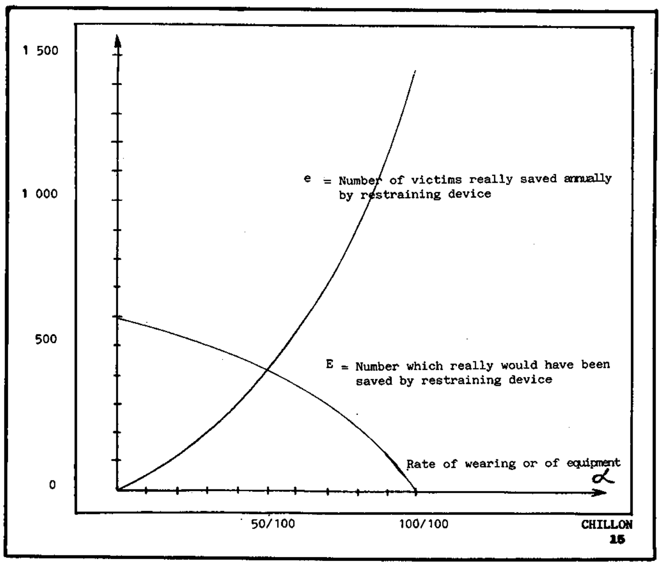


Figure 2. Development of rate of seatbelt wearing in Great Britain

In the graph below, the efficiency of the restraining device is that observed for the seatbelt with respect to deaths in the APR investigation $\lambda = 0.59$

The starting point is a single group of $\triangle V = 1000$ victims annually sampled in a country.



Calculation of the efficiency of the seatbelt.

The subject is the three-point seatbelt with retractor. The statistics are those of the "PSA—RNUR Association" taken in the PARIS Region with the assistance of:

- the police,
- the doctors of the Orthopedic Research Institute of GARCHES Hospital.
- the engineers of the automobile companies PEUGEOT, CITROEN and RENAULT.

Details of more than 6700 accidents are taken into account.

Since the advent of the seatbelt with retractor reel has had a significant influence on the efficiency of the seatbelt in frontal impact but not in lateral or other impacts, we have restricted our sample to wearers of retractor type seatbelts only in the case of frontal impact.

We have to preserve simplicity, distinguished only between three types of impact: frontal, lateral and others; the details are presented in tables 2 and 3.

We take into account the totality of "serious injuries + deaths" or "serious victims".

The efficiency is calculated as $e = \dfrac{Rnb - Rb}{Rnb}$

Where Rb and Rnb are the rates of "serious victims" versus those concerned, belted or non-belted (tables 1, 2 and 3).

Statistics of the A.P.R. Enquiry Front Seat Occupants.

Frontal impact

	Belted	With retractors	Non-belted
No. concerned	1561	1188	1390
Non-injured	534	—	140
Light injuries	826	—	919
Serious injuries	147	80	231
Deaths	54	32	100
Serious injuries + deaths	201	112	331

Side impact

	Left side		Right side		Total	
	belt	no-belt	belt	no-belt	belt	no-belt
No. concerned	214	210	204	195	418	405
Non-injured	56	23	77	37	133	60
Light injuries	128	25	94	108	222	233
Serious injuries	25	38	17	29	42	67
Deaths	5	24	16	21	21	45
Serious injuries + deaths	30	62	33	50	63	112

"Other" impacts

	Rear impacts:		Roll over		Complex		Total	
	belt	no-belt	belt	no-belt	belt	no belt	belt	no belt
No. concerned	268	63	339	220	52	47	659	330
Non-injured	111	17	119	41	7	7	137	65
Light injuries	149	37	203	134	29	21	381	192
Serious injuries	7	4	14	29	10	11	31	44
Deaths	1	5	3	16	6	8	10	29
Serious injuries + deaths	8	9	17	45	16	19	41	73

Sample of Accident Investigation by Peugeot SA/Renault Association

(Assessment based on front seat occupants of vehicles of model years after 1972)

Table 1. Belted front occupants.

	TYPES OF IMPACT				
	FRONTAL (1)	(2)	LATERAL	OTHERS	TOTAL
Deaths	54	32	21	10	85
Serious injury	147	80	42	31	220
No.concerned	1 561	993	418	669	2638

(1) All belted (static + retractor)
(2) Retractor seatbelts

Table 2. Non-belted front occupants.

	TYPES OF IMPACT			
	FRONTAL	LATERAL	OTHERS	TOTAL
Deaths	100	45	29	174
Serious injury	231	67	44	342
Total deaths + Ser.inju.	331	112	73	516
Non-belted	(64,1 %)	(21,7 %)	(14,2 %)	(100 %)
No. concerned	1390	405	330	2125

Table 3. All front occupants (I + II).

	TYPES OF IMPACT			
	FRONTAL	LATERAL	OTHERS	TOTAL
Deaths	154	66	39	259
Serious injury	378	109	75	562
No. concerned	2951	823	989	4763

Efficiency in frontal impact.

	Belted with retractors	Non-belted
No. concerned	993	1390
Serious injuries + deaths.....	112	331
Rates	11.28%	23.81%

Efficiency e = (23.81 − 11.28)/23.81 = 52.6%

Efficiency in side impact.

	Belted	Non-belted
No. concerned	418	405
Serious injuries + deaths.....	63	112
Rates	15.07%	27.65%

Efficiency e = (27,65 − 15.07)/27.65 = 45.5%

Efficiency in other impacts

	Belted	Non-belted
No. concerned	659	330
Serious injuries + deaths	41	73
Rates	6.22%	22.12%

Efficiency e = (22.12 − 6.22)/22.12 = 71.9%

Note: the same method, used for deaths only would have given the following efficiencies: frontal impact 55.6%—side impact 54.5%—other 82.1%—overall efficiency 59.1%.

Evaluation of the Efficiency of the Air Cushion

In the absence of accident details which are sufficiently precise and extensive, we will evaluate the efficiency of the inflatable bag by a case-by-case examination, based on the information of the APR enquiry.

In this evaluation, we will distinguish between frontal, lateral and other types of impact.

Evaluation of the Efficiency in Frontal Impact

The case-by-case study is made for the impacts for which the characteristics are precisely established, i.e. those where the speed differential $\triangle V$ is known; the study is limited to $\triangle V$ levels which are significant for these types of impact (table 1).

We examine in each case if the predicted trajectory of the occupant meets the bag. A geometric account is taken of the intrusion; we do not take into account the rising of the steering wheel which would be eliminated in the case where an air cushion is fitted.

Table 1. Front impact—non-belted occupants severity ($\triangle V$) of front impacts (front seat occupants severely injured or killed).

	△ V (km/h)							
	25	26–35	36–45	46–55	56–65	66–75	75	TOTAL
Non-belted Seriously injured	6 3,5	23 13,6	40 23,7	55 32,5	33 19,5	10 5,9	2 1,2	169 100%
Killed	0	3 5,4	6 10,7	19 33,9	10 17,9	16 28,6	2 3,6	56 100%

Seriously injured : average of △ V : 48 km/h
Standard deviation : 12 km/h
Representative class of △ V : 36 to 60 km/h

Killed : average of △ V : 57 km/h
Standard deviation : 12 km/h
Representative class of △ V : 45 to 69 km/h

Table 2. Serious injuries.

OCCUPANT TRAJECTORIES	△V (km/h)						
	UNKNOWN	≤ 35	36–45	46–55	56–65	66–75	TOTAL
11 o'clock	12	3	5 (0à2)	11 (4à5)	1 (–)	2 (1à2)	34
12 o'clock	42	23	31 (–)	41 (0à1)	28 (2à4)	7 (1à2)	172
01 o'clock	7	3	4 (1)	3 (2)	4 (0à1)	1 (1)	22
TOTAL	61	29	40 (1à3)	55 (6à8)	33 (2à5)	10 (3à5)	228

Values ( ) : case where occupants do not have the protection of the bag.

Table 3. Deaths.

OCCUPANT TRAJECTORIES	Δ V (km/h)						
	UNKNOWN	$\leqslant$ 35	36-45	46-55	56-55	66-75	TOTAL
11 o'clock	6	1	0	6 (2à3)	1 (-)	0 (-)	14
12 o'clock	33	2	6 (-)	13 (-)	8 (-)	14 (2)	76
01 o' clock	5	-	-	-	1 (-)	2 (1à2)	8
TOTAL	44	3	6 (-)	19 (2à3)	10 (1)	16 (3à4)	98

The results of the case-by-case analysis are as follows (see tables 2 and 3):

- for the seriously injured (non-belted)
  Efficiency $\frac{128 - 12.5}{128} = 90\%$ for 228 injured, so 205 saved.
- For deaths
  Efficiency $\frac{45 - 7}{45} = 84\%$ for 98 deaths, so 82 saved.
- The overall efficiency for serious victims is

therefore:

$$e = \frac{205 + 82}{228 + 98} = 88\%$$

It is necessary to subtract from this value 3% of persons ejected in frontal impacts through the doors after impact; in addition we will suppose that the efficiency of the bag is 100% whenever it is impacted while limiting the scope of protection to impacts of ΔV less than 44 M.P./H, which is very high; this limitation linked to ΔV causes a 5 point loss.

We finish with a final evaluation of e = 80% for frontal impacts where ΔV is known (i.e. car to car or fixed obstacles).

We have not tried to evaluate the losses of efficiency due to reliability faults linked to ageing, to displacement of the occupants, etc. . . ., which we suppose are negligible.

Case of other frontal impact

Those impacts of unknown ΔV are for the most part impacts against trucks; these are varied types of accidents, often severe with considerable intrusion; we have assumed an efficiency equal to that of the seatbelt for the same type of impact, which is 52.6%; this class represents 17% of serious victims in frontal impacts.

Overall, for all frontal impacts, the efficiency of the inflatable bag is $e = (0.83 \times 0.80) + (0.17 \times 0.53) \cong 7.6\%$.

Evaluation of the Inflatable Bag Efficiency in Side Impacts

Because of the diversity of situations in side impacts, the analysis has distinguished eight configurations which take account of the places occupied, the position of the occupants related to the impact and to the intrusions, and of the principal direction of the forces applied.

We have examined case-by-case if the occupants have or not impacted one or other of the two bags and could because of that benefit from an effect of protection. We admit that in this type of impact the bag deploys systematically, which in fact is a hypothesis.

In the case where the occupant contacts the cushion, we consider that the protection afforded equals that of the belt in the same accident configuration: this is probably also an optimistic hypothesis as there are risks of intrusion, of ejection or of sliding on the bag.

The efficiency of the air bag in side impacts is 17%.

Efficiency of the Inflatable Bag in Impacts Other Than Frontal and Lateral

This category included rear impacts, rollovers and those which cannot be classified.

The inflatable bag does not have an intrinsic efficiency in these configurations where ejection is predominant and where the impacts are often multiple.

However, the presence of the inflatable bag requires the application of the technology of the glued-in laminated windshield. This provides a "safety net" effect preventing total or partial ejection in that direction; the result is a gain in protection which is attributed to the inflatable bag.

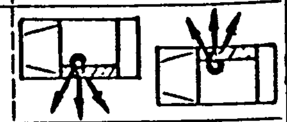
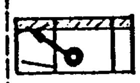
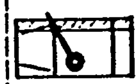
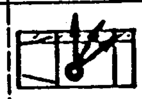
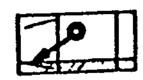
Distributing of the severity according to belt wearing and ejection modes:

	non - belted				belted
	not ejected	ejected by other apertures	ejected by windshield	total non-belted	
KILLED	13	11	6	30	10
SERIOUSLY INJURED	24	12	7	43	31
NO. CONCERNED	265	46	19	330	669

For 19 occupants ejected by the windshield there are 13 killed or seriously injured; with the bag and the glued-in windshield, these 19 occupants would have avoided ejection and would instead have run the same risks as the non-ejected occupants, i.e. $\frac{37}{265}$.

There would have therefore been $\frac{19 \times 37}{265}$, or about 3 victims only instead of 13; so 10 occupants would have been saved and the efficiency of the bag (with glued-in windshield) offers finally in this case an efficiency of $\frac{10}{73}$, or 14%.

Measured Efficiency of the 3-Point Seatbelt and the Expected Benefit of the Inflatable Airbag for Front Seat Occupants in a Lateral Impact.
Legend: B = 3-point belt NB = non-belted AB = airbag

LINE	Column PLACE CONFIGURATION DIRECTION OF FORCES	(a) No. OF SERIOUS INJURIES AND DEATHS B	(b) NB	(c) % SERIOUS INJURIES AND DEATHS (b) NB	(d) No. CONCERNED B	(e) NB	(f) :: (g) SEVERITY RATE (a)/(d) B	 (b)/(e) NB	(h) CALCULATED EFFICIENCY OF SEATBELT	(i) EFFICIENCY OF RETENTION METHOD B	(j) AB	(k) BENEFIT FOR 100 KILLED + SERIOUS INJUR. B	(l) AB	DIAGRAMS DRIVER FRONT PASSENGER
1	DRIVER + FRONT PASSENGER IMPACT SIDE WITH INTRUSION	28	49	43,8 %	89	112	31,5	43,8	- 28 %	Yes	No	- 12	0	
2	DRIVER + FRONT PASSENGER IMPACT SIDE WITHOUT INTRUSION	11	17	15,2 %	122	98	9,0	17,3	- 48 %	Yes	No	- 7	0	
3	DRIVER OPPOSITE SIDE TO IMPACT "AT ONE O'CLOCK"	0	3	2,6 %	15	23	0	13,0	(- 95 %)	Yes	Yes	- 2	- 2	
4	DRIVER OPPOSITE SIDE TO IMPACT "AT TWO O'CLOCK"	5	17	15,2 %	57	61	8,8	27,9	- 68 %	Yes	Yes	- 10	- 10	
5	DRIVER OPPOSITE SIDE TO IMPACT "NOT AT ONE OR TWO O'CLOCK"	14	13	11,6 %	66	56	21,2	23,2	- 8 %	Yes	No	- 1	0	
6	FRONT PASSENGER OPPOSITE SIDE TO IMPACT	0	1	0,9 %	8	11	0	9,0	(- 95 %)	Yes	No	- 1	0	
7	FRONT PASSENGER OPPOSITE SIDE TO IMPACT "AT TEN O'CLOCK"	2	7	6,2 %	28	22	7,7	31,8	- 78 %	Yes	Yes	- 5	- 5	
8	FRONT PASSENGER OPPOSITE SIDE TO IMPACT "NOT AT TEN OR ELEVEN O'CLOCK"	3	5	4,5 %	33	22	9,1	22,7	- 60 %	Yes	No			
9		63	112	100 %	418	405						- 41	- 17	

Overall Efficiency of the Air Cushion

For the total of the three categories of impact, we find:

$$\frac{.76 \times 641 + 0.17 \times 217 + 0.14 \times 142}{1000} = 54.4\%$$

Experimental Confirmation of the Compared Efficiency of Occupant Retention by Three-Point Seatbelt or Inflatable Bag in the Steering Wheel During Severe Oblique or Lateral Impacts.

Test Definitions

1) Evaluation of the performance of the steering wheel-mounted inflatable bag, measured on the HYBRID II dummy, during severe collisions comprising a deformation of the occupant space.
   a) Intrusion simulation of 150 mm. at dashboard and steering wheel level; test conducted on a bogey-mounted car body $\triangle V$ = 50 km/h.
   b) Destructive test of a PEUGEOT 505 vehicle against 30° oblique wall $\triangle V$ = 58 km/h.
2) Performance evaluation of the steering wheel-mounted air bag and of the three-point seatbelt, measured in biomechanical terms on the APROD dummy, in lateral impacts. These tests have been conducted with a deformable mobile barrier of 100 kg impacting a PEUGEOT 505 vehicle at 45°. The axis of the barrier was centered on the H-Point of the dummy.

Results

Tests with vehicle body intrusion

During the dummy's deceleration period in the bag:

—H I C head	248
—Thorax	48 g in 3 ms module,

Dynamic: the head and the thorax are decelerated on the side of the bag. Risk of hitting the A-post on some vehicles.

Destructive vehicle tests on an oblique wall $\triangle V$ = 58 km/h.

—H I C head	845
Thorax	56 g. in 3 ms module,

Dynamic: the inflatable air bag protection is assured, the head and thorax impact practically the centre of the steering wheel bag.

Destructive vehicle tests in lateral impact by a deformable mobile barrier (45° angle)

a) Dummy restrained by three-point seatbelt barrier impact speed:

Dynamic: no head impact.

b) Vehicle equipped with a wheel-mounted inflatable bag.

Barrier impact speed: 52 km/h.

—H I C head	137
—Thorax	28 g in α module at 3 ms
—Pelvis	48 g in α module at 3 ms

Conclusion

The inflatable bag offers an efficient protection in severe frontal impacts at 0° and 30° ($\triangle V$ = 60 km/h), including the case where there is limited cabin deformation.

This protection is no longer guaranteed during side impact when the shock angle reaches 45°. In this scenario the seatbelt plays a role only against the eventual risk of ejection.

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Study on the Relationship Between Seat Belt Anchorage Location and Occupant Injury

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Abstract

In order to accomplish effective protection of vehicle occupants during collisions, much care has been paid to vehicle safety parameters such as crashworthiness, the performance of occupant restraints, the energy absorption characteristics of the interiors and so on.

The seat belt is the most popular restraint system and its effectiveness has been demonstrated through accident investigations and many kinds of tests. However, it is now always easy to evaluate seat belt anchorage location without conducting crash tests or math-model simulations.

We have developed a new method to estimate this effect, which can easily relate the seat belt anchorage location to occupant injury level by adopting the "Anchorage Coefficient". This coefficient can be easily calculated by using a simple formula based on the geometric location of seat belt anchorages relative to the occupant.

Validity of the method has been confirmed through various series of sled tests, and some applications have also been available with satisfactory results.

Introduction

The seat belt is the most popular restraint system and has proved its effect through many kinds of accident investigations and various impact tests.(1)(2) However, it should be carefully designed in the development stage of new vehicles.

The seat belt anchorage location is one of the important factors which affect the occupant injury level, and much investigation has been conducted on this point. However, such investigations were based upon sled testing and/or mathematical simulation, and the results were relative ones or unique to the test vehicle. Therefore, it is not always easy to apply such result to newly developed vehicles.

On the other hand, the seat belt anchorage locations have to be fixed in the early stage of development because

(1) seat belt anchorage locations will affect the fit and the accessibility of the seat belt, and
(2) change of the upper torso anchorage location will often result in change to styling, since the achorage is usually mounted on the pillar.

Therefore, if the effect of seat belt anchorage location on occupant injury level could be predicted, it would become a useful tool during early design stage.

From above viewpoint, we have developed a new method to assess the effect of seat belt anchorage location from the geometrical layout. This method is based upon the ride-down effect.

The validity of the method has been verified through various kinds of sled tests.

Method

Background

In order to reduce the occupant injury level during impact, it is essential to increase the ride-down effect. The ride-down effect is the phenomenon where the occupant kinetic energy is converted into vehicle deformation through the restraint system during impact.

The ride-down effect is usually specified quantitatively by the ride-down efficiency (η) defined by formulas (1).

$$\eta = \frac{\int F\,dx}{1/2M \cdot V^2} \qquad (1)$$

here, M : occupant mass
V : impact speed
F : force applied on occupant by seat belt
x : vehicle displacement

Figure 1[(3)] shows the relationship between the occupant injurylevel (chest-G) and ride-down efficiency. It can be clearly seen that higher ride-down efficiency creates lower injury level.

Judging from formula (1), we can conclude that higher ride-down efficiency will be achieved if the seat belt is tensed in the early stage of an impact. This fact requires that the seat belt anchorage must be located

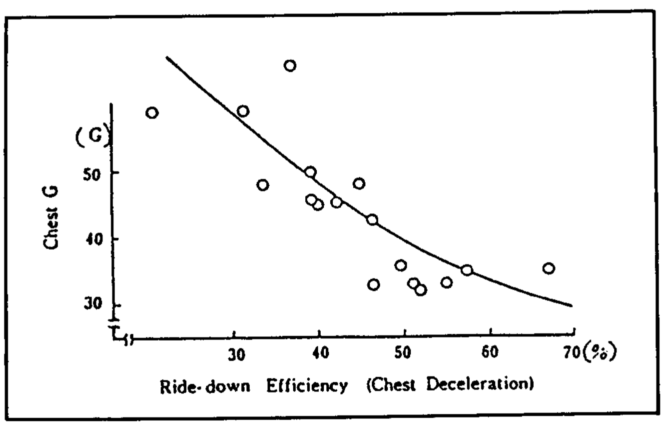

Figure 1. Chest G vs. ride-down

so that it can produce the belt force as early as possible during an impact. In an effort to assess it, we have introduced a new criteria, that is, "Anchorage Coefficient", which is explained in the following sections.

Definition of Anchorage Coefficient

Figure 2 specifies the geometric relationship of a seat belt between an occupant and the vehicle compartment. Here,

P	:	belt force applied point on occupant
0	:	seat belt anchorage point on vehicle
α	:	angle between belt and vehicle longitudinal line (X-direction)
l_b	:	belt length (between P and O)
x	:	X-component of length between P and O

We suppose now that point P is travelling forward by small increments ($\triangle x$) relative to the compartment. Also we assume that there is no slip between the belt and the occupant, that is, the belt is fixed to the occupant at the point P.

From simple geometric considerations, the increment of the X-component of seat belt force is written in formula (2), neglecting terms of $\triangle x$ with second or higher order.

$$\triangle Fx = Ko\ (x^2/1_b^3)\triangle X \tag{2}$$

Ko: spring constant of seat belt per unit length

The term $(x^2/1b^3)$ of formulas (2) depends only upon the seat belt layout, and we defined it as "Anchorage Coefficient".

$$C = x^2/1_b{}^3 \tag{3}$$

The meaning of formula (3) is interpreted as follows. When C is larger, $\triangle Fx$ will be increased, given the same $\triangle x$ in formula (2), and, consequently, the seat belt will restrain the occupant earlier in the impact. At the same time the larger ride-down effect will be produced and it will result in a lower occupant injury level. Therefore, it can be seen that a seat belt anchorage with a larger C has an advantage to reduce the injury level.

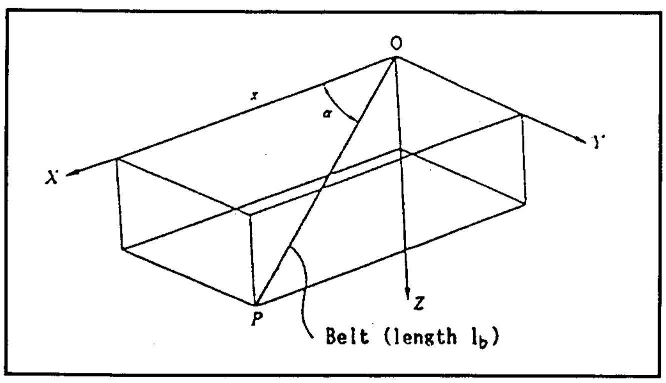

Figure 2. Scheme of seat belt

If point P is assumed to be coincident with point 0, as an extreme example, C will be infinite because of zero value of 1_b, and the highest efficiency will be achieved. In that case, the occupant is rigidly connected to the vehicle and occupant deceleration is the same as vehicle deceleration, the ride-down efficiency becomes 100 percent.

Formula (3) can be represented as follows, because $x = 1_b \cos\alpha$.

$$C = \cos^2\alpha/1_b \tag{4}$$

Upper Anchorage Coefficient

The Upper Anchorage Coefficient (C_S is calculated by formula (4), where the upper torso point near the shoulder shown in figure 3 is regarded as the belt attached point (P).

Lower Anchorage Coefficient

The Lower Anchorage Coefficient (C_L) is calculated by formulas (4), where the outside point of the pelvis shown in figure 4 is regarded as the belt attached point (P).

Sled Test

Sled tests have been conducted to evaluate the validity of the newly-introduced method. Both "lap and torso belt (3-point belt)" and "torso belt and knee bolster (2-point belt)" systems were tested. The impact condition of the tests correspond to the 35 mph frontal barrier crash test. The PART 572 HYBRID-II dummy was used, and the injury scores specified in FMVSS 208 were measured.

3-Point Belt

(1) Anchorage Location Range of Variation

Table 1 shows the anchorage location areas which were tested. In order to identify influential parameters, either upper torso

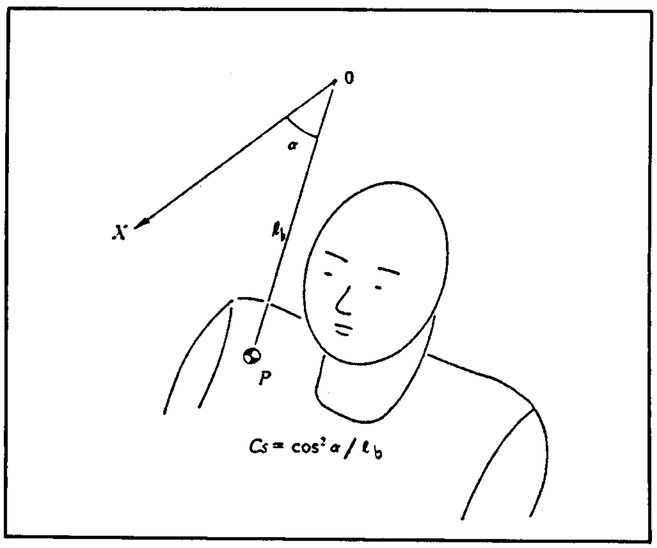

Figure 3. Definition of upper anchorage coefficient

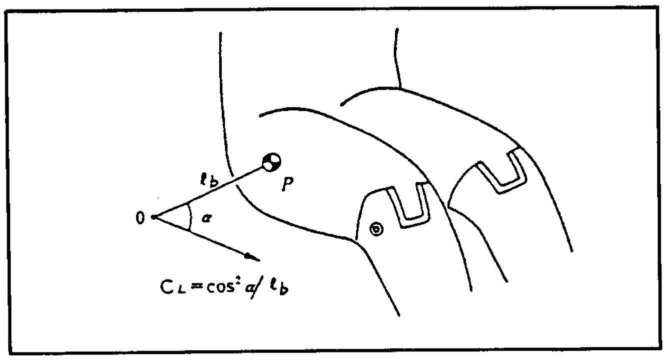

Figure 4. Definition of lower anchorage coefficient

anchorage location or lap anchorage location was varied, with the pitch of 50mm to 100mm, while the other was fixed. All other conditions were identical in each test.

(2) Test Result

Figure 5 shows the relationship between HIC and Upper Anchorage Coefficient (C_S). There seems to be good correlation between them, although there is some variation. The higher C_S seems to produce lower HIC. For example, when C_S is increased from 1.0 to 2.0, HIC decreases by 35%.

Figure 6 denotes the relationship between head forward displacement and C_S. A higher C_S will result in a smaller forward displacement.

Judging from these two results, it can be concluded that the upper anchorage coefficient can predict the restraint performance of a proposed upper anchorage location.

Figure 7 shows the relationship between HIC and Lower Anchorage Coefficient (C_L) which is obtained in the series of tests with various lower anchorage locations. The variation of HIC is different from the one in the case of C_S. HIC has minimum value at $C_L = 1.0$ and increases with C_L higher than 1.0. The reason is estimated that higher C_L than 1.0 will result in too stiff restraint at the pelvis and will increase the head rotation creating the more centrifugal force.

Table 1. Test condition—range of variation. (3-point belt)

(a) Upper Anchorage		(b) Lap Anchorage	
Area of Upper Anchorage		Area of Lower Anchorage	
longitudinal	250 mm	longitudinal	500 mm
vertical	100 mm	vertical	100 mm
lateral	150 mm	lateral	0 mm

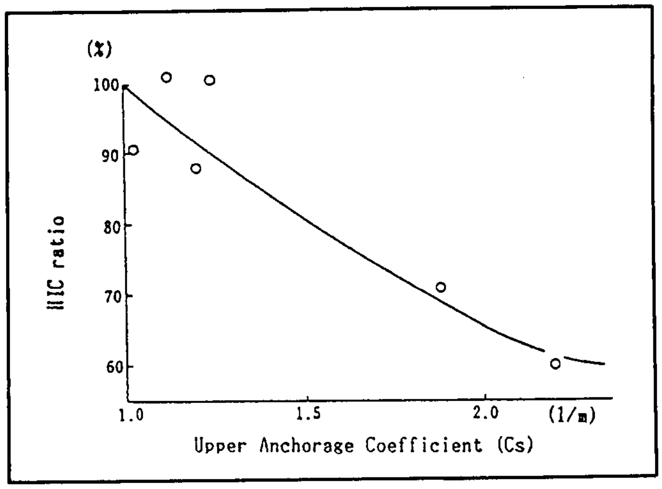

Figure 5. HIC vs C_S

This series of these tests was conducted with $C_S = 2.0$. This fact and other considerations seem to suggest that $C_S/C_L = 2.0$ is one of the optimum balance. However, much more study will be conducted to identify the optimum condition within practical accuracy.

Anyway, the restraint ability of the lower anchorage could be induced from Lower Anchorage Coefficient.

2-Point Belt

2-point belt and knee bolster systems have been introduced in the U.S. market by many automobile manufacturers. Figure 8[4] shows one example which was intruced by Volkswagenwerk. Hereunder, we call it "2-point belt" for simplification.

(1) Anchorage Location Range of Variation

Table 2 shows the anchorage location areas which were tested. In order to identify influential parameters, either upper torso anchorage location or lower anchorage location was varied, with the pitch of 50mm to 100mm, while the other was fixed.

All other conditions were identical in each test.

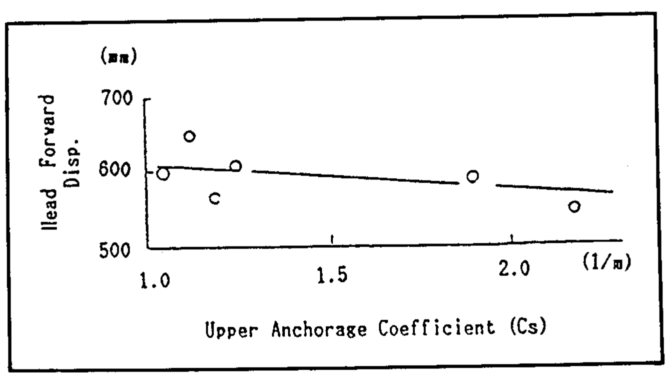

Figure 6. Head disp. vs C_S

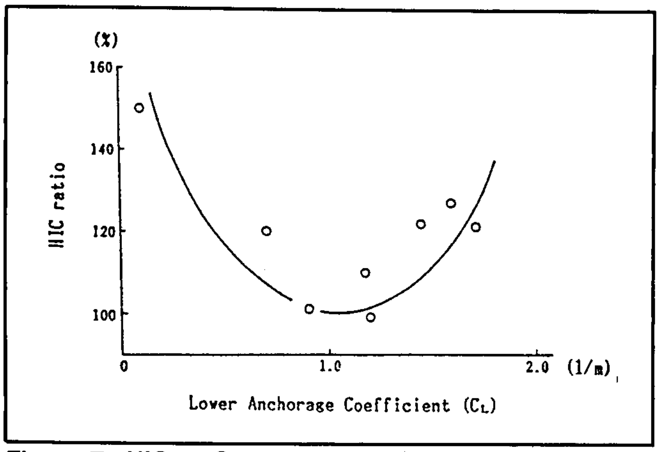

Figure 7. HIC vs C_L

(2) Test Result

Figure 9 shows the relationship between HIC and Upper Anchorage Coefficient (C_S). The higher C_S decreases HIC as in the case of lap and torso belt. However, the decreasing rate of 2-point belt is lesser than the one of 3-point belt. For example, when C_S increases to 2.0 from 1.0, HIC decreases by approximately 10%, while it decreases by 35% in case of a 3-point belt as described above.

Figure 10 denotes the relationship between head forward displacement and C_S. A higher C_S will create a smaller head displacement. The restraint performance of the upper anchorage of 2-point belts also seems to be predicted by Upper Anchorage Coefficient, although the decreasing rate is different from 3-point belts.

Figure 11 specifies the relationship between HIC and Lower Anchorage Coefficient (C_L). In this case, the higher C_L increases HIC, different from the other three cases mentioned above. The reason is estimated as follows. In the actual sled test, the torso belt

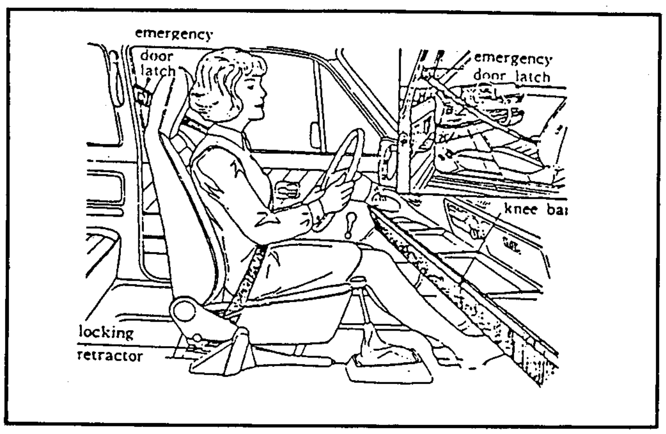

Figure 8. 2-point belt

Table 2. Test condition—range of variation. (2-point belt)

(a) Upper Anchorage

Area of Upper Anchorage	
longitudinal	350 mm
vertical	150 mm
lateral	0 mm

(b) Lower Anchorage

Area of Lower Anchorage	
longitudinal	300 mm
vertical	50 mm
lateral	0 mm

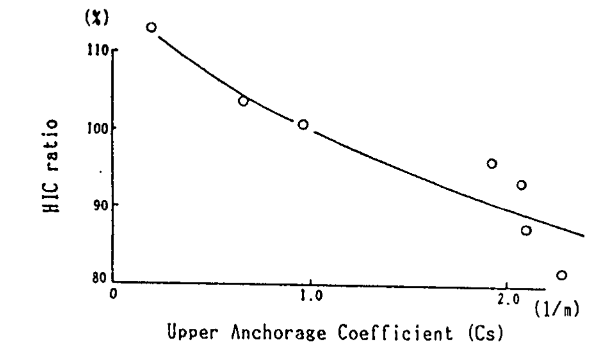

Figure 9. HIC vs C_S

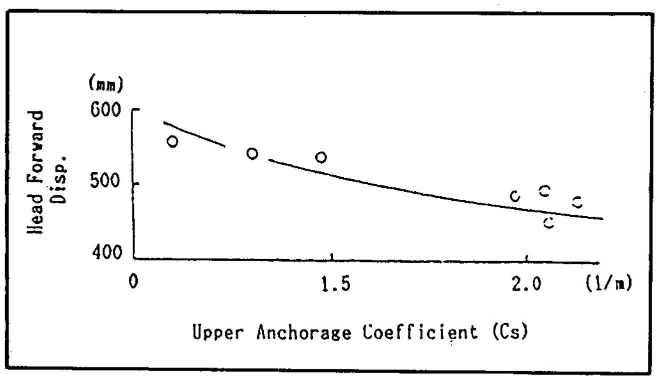

Figure 10. Head disp. vs C_S

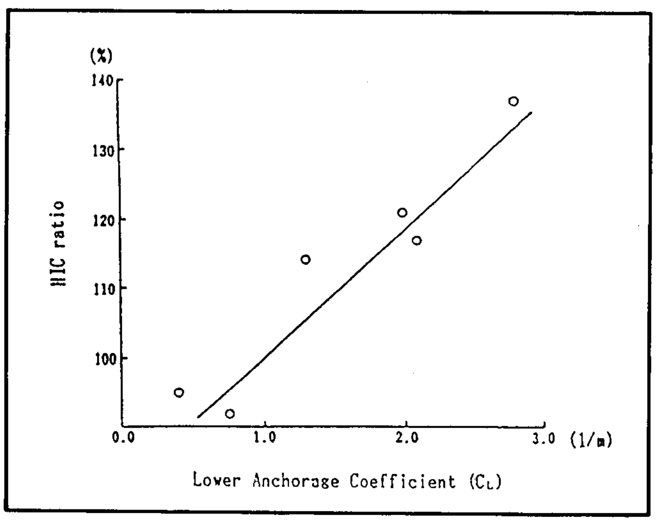

Figure 11. HIC vs C_L

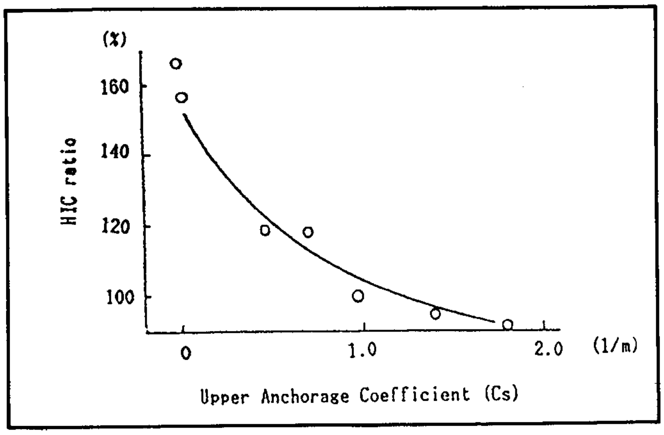

Figure 12. HIC vs C_S (AF5%ile)

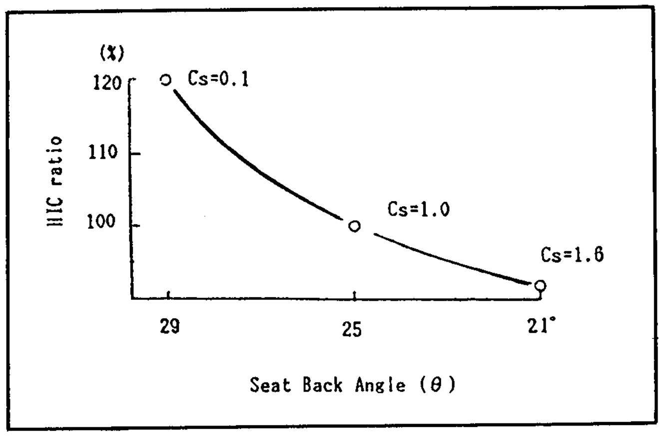

Figure 14. HIC vs seat back angle

was observed to slide up over the upper torso of the dummy, which means that the belt applied point on the dummy changes during crash.

The initial assumption that the belt applied point is fixed to the dummy in the Anchorage Coefficient could cause deviation. Therefore, it will be improved to take account of the slip of the belt.

Summary of Confirmation Test

Judging from the above-mentioned results, it can be said that the restraint performance can be approximately predicted by the Anchorage Coefficient, when the belt system ("3-point belt" or "2-point belt") and anchorage locations are known.

Obviously, the absolute value of injury level cannot be estimated, because the injury level depends upon the vehicle crash characteristics and many factors other than seat belt characteristics. However, the Anchorage Coefficient will suggest important design information especially in the early stages of design.

Application

Different Size Dummy

The Anchorage Coefficient was applied to the AF5%ile dummy to confirm the applicability of the method. In case of AF5%ile, the belt applied point on the dummy (Point P in the Figure 2) is selected to fit the upper torso of the dummy. The impact velocity was also 35 mph.

Figure 12 shows the relationship between HIC and Upper Anchorage Coefficient (C_S). It can be seen that HIC decreases more than in the case of AM50%ile, as C_S increases

Figure 13 shows the relationship between head forward displacement and C_S. The figure denotes higher C_S will produce a smaller head displacement.

Seat Back Angle[5]

It has been demonstrated that the seat back angle affects the injury level. We have tried to confirm whether the Anchorage Coefficient can explain the phenomenon or not.

Figure 14 shows the relationship between HIC and seat back angle in the 2-point belt system. HIC increases as seat back angle increases.

On the other hand, as seat back angle increases, the Upper Anchorage Coefficient (C_S) decreases. In figure 14, the value of C_S corresponding to the seat back angle is also presented. When we use figure 9, we can explain the effect of seat back angle by the variation of C_S. Figure 15 shows the result.

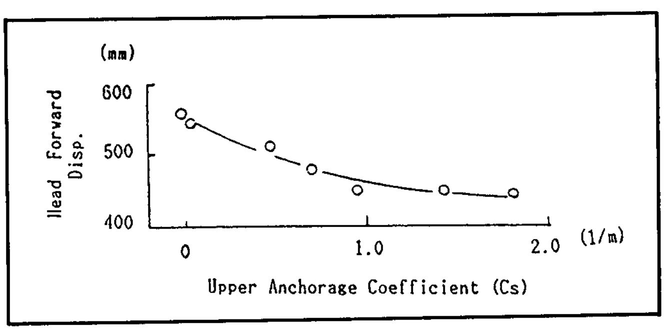

Figure 13. Head disp. vs C_S (AF5%ile)

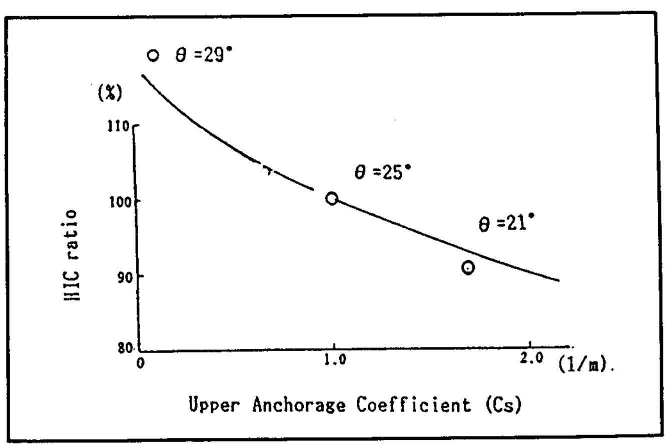

Figure 15. HIC vs C_S

Conclusion

1. The newly-developed Anchorage Coefficient can predict the occupant restraint performance approximately without any computer simulation or crash test, and is a useful method especially in the early stage of development.
2. The Anchorage Coefficient can be easily calculated by a simple formula based upon the geometrical layout of seat belt anchorages.
3. The method can be applied to the different-size dummy such as AF 5 percentile dummy.
4. The method can also explain the effect of seat back angle.

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Relationship Between Vehicle Front-End Stiffness and Dummy Injury During Collisions

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Abstract

Vehicles must possess adequate front-end stiffness to protect the occupants of the striking vehicle in frontal collisions. At the same time, it is also necessary to reduce the striking vehicle's aggressivity toward other vehicles in rear-end and side impacts. It is important, therefore, to find a suitable level of front-end stiffness that will satisfy both requirements.

As one approach to finding an appropriate front-end stiffness, analyses were made of the NCAP test reports issued by NHTSA. Investigations were carried out concerning the load generated by the barrier at the time of impact, vehicle displacement relative to time, and dummy injury levels.

Based on the target stiffness obtained from those analyses, improvements were made to the impacter of the deformable moving barrier, and methods for varying the stiffness were investigated. On the basis of the results obtained, this paper proposes a suitable stiffness for the DMB impacter used in vehicle collision testing.

Introduction

In many countries around the world research is under way into collision testing methodologies involving the use of a deformable moving barrier (DMB). Evaluations of vehicle safety must take into account crashworthiness as well as the striking vehicle's aggressivity against the vehicle being struck.

Therefore, in developing a DMB to simulate vehicle front-end stiffness, one very important factor is to select an appropriate target stiffness. There is a large body of literature dealing with how vehicle crash characteristics should be designed in order to mitigate the crash impact and protect the occupants. This is an extremely difficult issue which defines simple solutions.

In this work, experiments were first carried out to examine what effect the material strength of the impacter at the front of the moving barrier would have on dummy injury levels. These experiments were carried out under typical conditions for side impacts and frontal collisions.

Using NCAP test reports on 1983-1986 models, analyses were also carried out to identify the actual crash characteristics of vehicles currently on the road.

Relationship between Material Strength of DMB Impacter and Dummy Injury Levels

In side impacts

Two accident patterns were selected for side impacts. For each pattern, investigations were made into the relationship between the material strength of the DMB impacter used to represent the striking vehicle and the injury levels sustained by the dummy in the struck vehicle.

One pattern involved a 90-degree side impact, as illustrated in Figure 1. The struck vehicle was a four-door compact sedan, weighing 1100 kg. This test car was a slightly modified version of the baseline vehicle. The material strength of the DMB impacter was about 1.3 kgf/cm^2 in one series of tests and about 3.2 kgf/cm^2 in another.

The second pattern was created using a crabbed DMB having a 27-degree angle, as shown in Figure 2. The initial angle of impact between the striking and struck vehicles was set at 90 degrees. A four-door compact sedan weighing 1480 kg was used as the struck vehicle. The DMB impacter had a material strength of approximately 1.7 kgf/cm^2 in one set of experiments and about 3.2 kgf/cm^2 in another.

From the results obtained in these two side impact patterns, comparisons were made of the effects of the DMB impacter material strength and dummy injury levels (chest G). With the first pattern, it was found that an increase in material strength of roughly 2.5 times resulted in approximately a 1.5-fold increase in injury levels. In the case of the second pattern, a 1.8-fold increase in material strength caused the injury levels to rise by a factor of about 1.2.

These results confirmed that the material strength of the DMB impacter has a large effect on the injury level sustained by the dummy in the struck car in side impacts.

In frontal collisions

The most commonly employed approach to evaluating occupant safety in frontal collisions is to conduct collision tests using a fixed barrier. In this work, however, the effects of the material strength of the DMB impacter were examined in frontal collisions in which it was assumed that both vehicles were moving.

The test results are shown in Figure 3. In test (a), the baseline vehicle was a four-door compact sedan weighing 1100 kg. In one experiment the DMB impacter was given the same stiffness ratio as the struck vehicle, while in another experiment it was set at about 1.8 times that of the impacted vehicle. The vehicle and DMB were caused to collide at a relative velocity of 70 mph.

Two DMBs, each weighing 1100 kg, were used in test (b). A dummy restrained by a three-point seat belt system was placed in a fixed seat installed on one DMB. The test was carried out under the conditions of a high-speed collision. The relative collision velocity was set at 38 mph so as to avoid destruction of the impacter material.

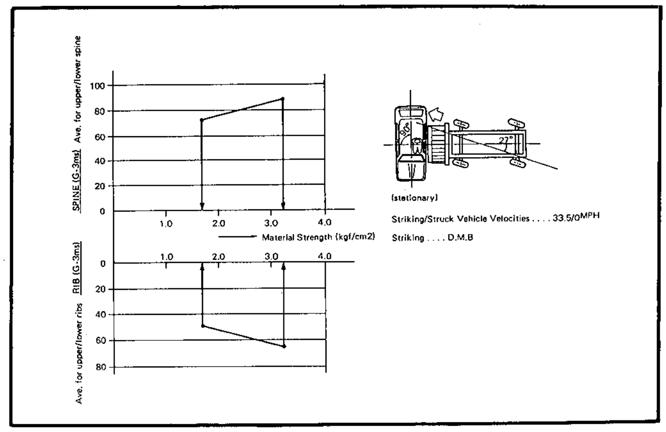

Figure 2. Relationship between front-end stiffness of striking vehicle and chest injury in crabbed moving barrier impacts

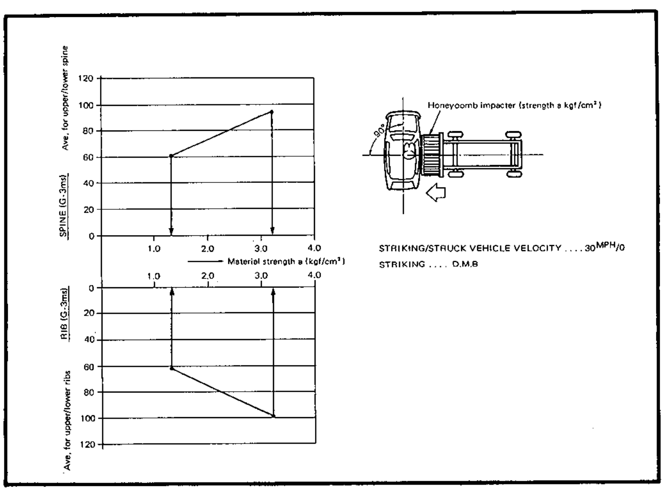

Figure 1. Relationship between front-end stiffness of striking vehicle and chest injury in 90 degree side impacts

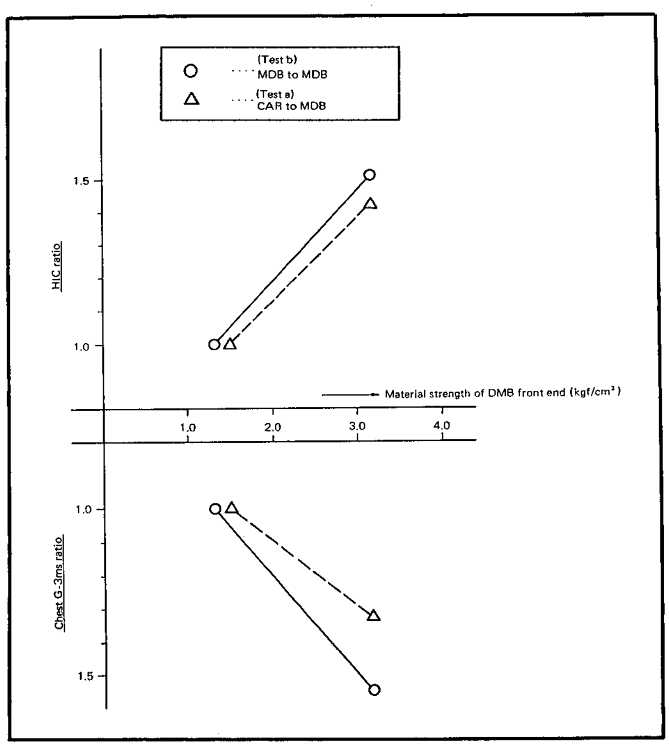

Figure 3. Relationship between front-end stiffness and injury level in car-to-car frontal collisions

Based on the injury levels sustained by the dummy in tests (a) and (b) when the DMB impacter had the same stiffness ratio, the rate of increase in head injury criteria (HIC) was found to be approximately 1.4 to 1.5 times. The rate of increase in chest G-3 ms was found to be roughly 1.3 to 1.6 times.

These test results confirmed that, in side impacts and frontal collisions, the injury levels incurred by the dummy in the struck vehicle are greatly affected by the front-end stiffness of the striking vehicle.

Relationship between Vehicle Crash Characteristics and Dummy Injury Levels

Basic concept of vehicle collision phenomena

The results presented in the foregoing section made it clear that the appropriate front-end stiffness should be a well-balanced compromise between two factors. It should reduce the aggressivity of the striking vehicle toward the impacted vehicle and also provide sufficient protection for the occupants of the former vehicle.

During the vehicle development process, front-end stiffness is generally determined on the basis of frontal collisions with a fixed barrier, as such crashes are thought to represent some of the most severe collision conditions. The following discussions therefore will focus on vehicle crash characteristics in frontal collisions.

A model like that illustrated in Figure 4 was devised in order to grasp the phenomena of frontal collisions macroscopically. With this model it is assumed that the dummy placed in the passenger seat and restrained by a seat belt does not suffer any secondary head impact. Under this condition, the G value generated on the dummy's chest, F(C), is determined by the combination of the vehicle's crash characteristics, F(B), and the restraint characteristics of the salt belt system, F(S). In addition, it is also assumed that the G value generated on the dummy's head, F(H), is determined by the combination of the G value generated on the dummy's chest and its neck characteristics, F(N).

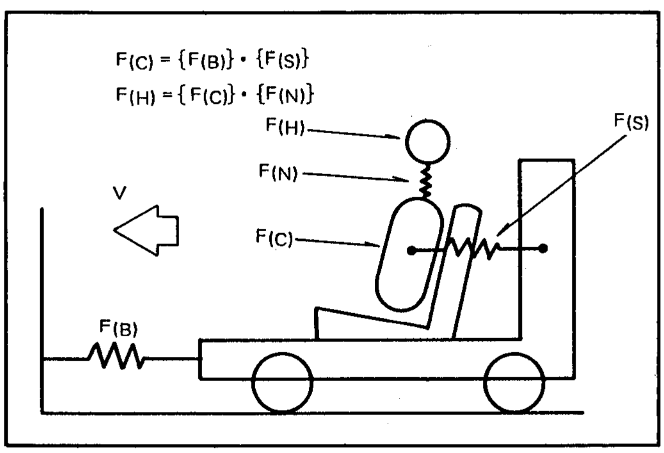

Figure 4. Concept of impact model

The relationship between the chest injury level (G-3 ms) and the head injury level (HIC) was investigated in sled tests, in which the dummy suffered no secondary impact. The results obtained are shown in Figure 5.

In this figure, the restraint characteristics of the seat belt system, F(S), and the dummy's neck characteristics, F(N), represent certain given values. Consequently, the head injury level can be estimated if the chest injury level is known. In other words, finding the G value generated on the dummy's chest will provide a basic value that can be regarded as a typical injury level for frontal collisions involving two vehicles.

Method of evaluating vehicle crash characteristics

One common index of crash characteristics is the so-called "ride down factor". This factor represents the proportion of the passengers' total pre-crash energy which is absorbed by the vehicle. The larger this value becomes, the more the body design is oriented toward occupant protection. However, this ride down factor has the disadvantage of not being very practical to apply, because it can only be quantified by carrying out simulation calculations.

In this work, an attempt was made to find an index of vehicle crash characteristics which could be used in place of the ride down factor. This attempt was made

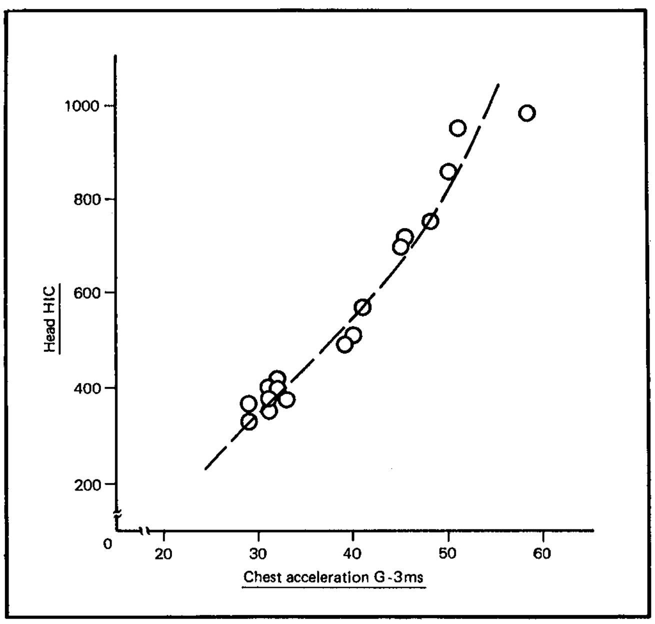

Figure 5. Sled test results showing relationship between dummy chest G-3ms and head HIC in crashes without secondary head impact

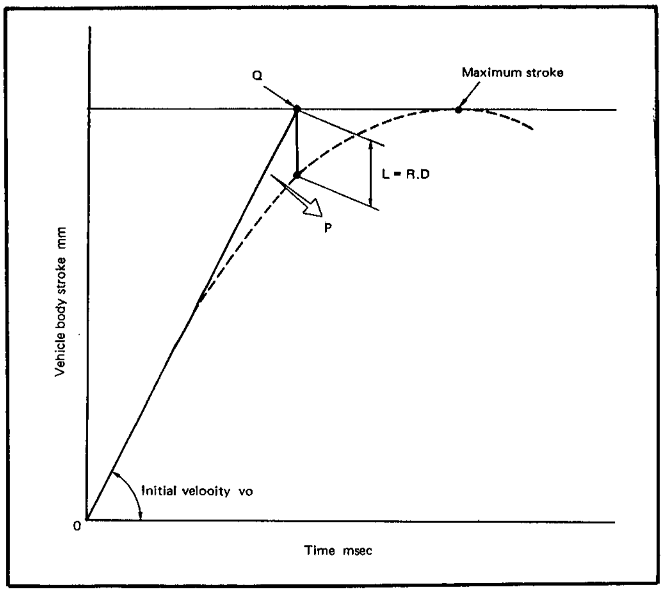

Figure 6. Definition of residual deformation

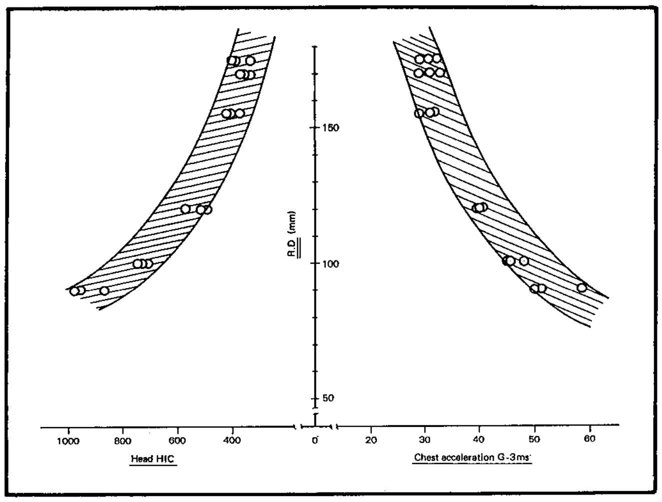

Figure 7. Sled test results showing relationship between dummy injury level and residual deformation

under the restrictive condition that analyses were limited to NCAP test report data.

In searching for an index to be used in evaluating crash characteristics, attention was focused on the vehicle displacement-time curve (S-T curve), which was derived from general collision test data.

A sample S-T curve for a frontal collision with a moving barrier is shown in Figure 6. The distance indicated by L in the figure is defined as the residual deformation. This concept of residual deformation was presented in a paper given by a Nissan research engineer at the Sixth ESV Conference. A brief explanation of this concept follows.

Residual deformation is expressed in millimeters and it indicates the extent to which the S-T curve moves away from point Q in direction P. Point Q represents the point of displacement under the initial velocity (Vo) for a vehicle having no stiffness. It is the baseline point for measuring maximum deformation.

Expressed differently, residual deformation refers to the amount of deformation that occurs during the second half of a collision. The larger the residual deformation, the longer the reaction force of the passenger restraint system decelerates, without showing any significant increase. This means that the crash characteristics of a vehicle become more favorable as the amount of residual deformation increases.

Following this line of thinking, sled tests were carried out to examine the relationship between the amount of residual deformation and the dummy injury level. The results obtained are shown in Figure 7.

The tests were conducted with the dummy placed in a fixed seat installed on the sled, and the collision velocity was set at about 34 mph. The amount of residual deformation was varied by changing the collision deceleration waveform of the sled. The restraint system specifications were made the same in all three tests conducted.

While the results in Figure 7 show some variation even with the same restraint system, good correlation is seen between residual deformation and the dummy injury level. As indicated above, the dummy injury level is reduced as residual deformation increases.

Analysis of NCAP test report data

An analysis was then made of NCAP test report data to determine if cars on the road today would show the same tendencies. The analysis included a total of 44 test reports on 1983 to 1986 models.

The relationship found between the injury level sustained by the passenger seat dummy (chest G-3 ms) and residual deformation is shown in Figure 8. In this

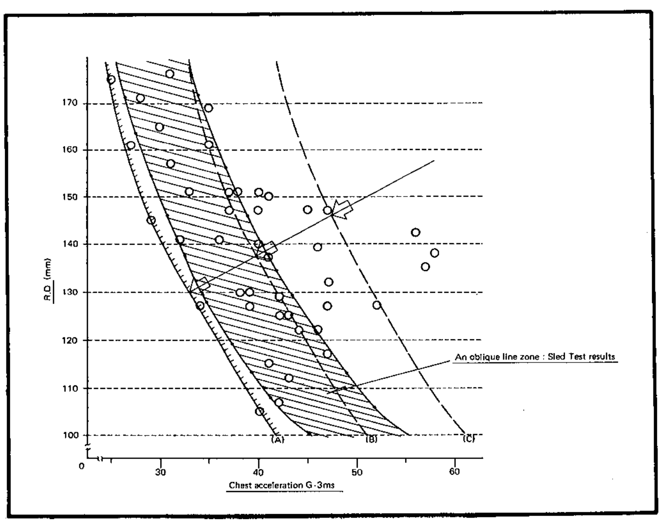

Figure 8. Relationship between dummy chest injury level and residual deformation (found from NCAP test data)

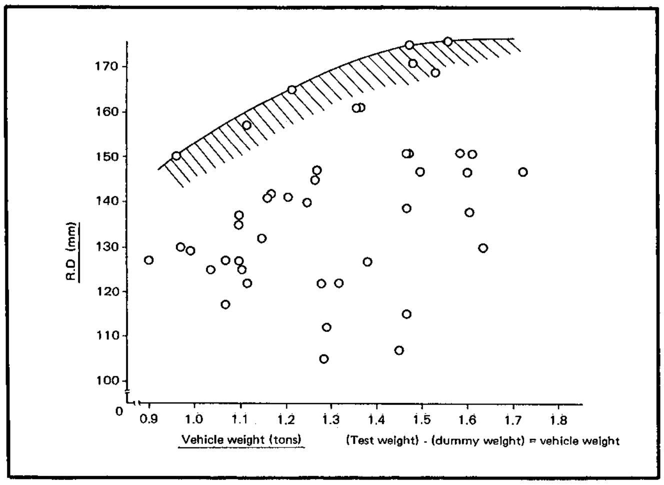

Figure 9. Relationship between vehicle weight and residual deformation

case, residual deformation was calculated using a deformation-time curve that was converted from the body sill deceleration waveform.

Lines A, B and C in the figure indicate variation in dummy injury levels between vehicles having the same residual deformation. It is assumed that this variation was caused by differences in their dummy restraint systems, i.e. the restraint characteristics of the seat belt system, F(S).

The vehicles making up zone (A) in the figure showed the same gradient in the results of the sled tests, which were carried out under the condition of uniform restraint characteristics. This indicates that their restraint characteristics were virtually identical to those of the vehicles in boundary zone (A). Among the restraint systems in use today, these fall into the category that shows good correlation with vehicle crash characteristics.

Experiments were then carried out to investigate which vehicle specifications and characteristics were related to residual deformation. To begin with, the relationship between vehicle weight and residual deformation was examined (Figure 9). Then, the relationship between the length of the front end and residual deformation was investigated (Figure 10).

From the results shown in Figures 9 and 10, it would appear that for existing vehicles residual deformation is limited by vehicle size, including vehicle weight and front-end layout.

At this point, we will examine this conclusion in more detail. It is assumed that collision phenomena can be represented as simple harmonic motions in order to grasp them macroscopically. Based on this assumption, the relationship between residual deformation and vehicle specifications can be expressed as illustrated in Figure 11. Since residual deformation equals $Vo\sqrt{\frac{M}{K}}$ (1 - sin 1), it is assumed to proportional to $\sqrt{\frac{M}{K}}$.

Load-displacement diagrams (F-S diagrams) were made using the load generated by the barrier at the time of impact, (F), and the displacement-time curve (S-T curve) for the vehicle. The vehicle's spring rate (K) was then found according to the concept illustrated in Figure 12.

In this figure, K is regarded as being a linear spring rate having the same area as the F-S curve up to a displacement 500 mm. (This displacement is thought to include the length of time for the generation of the maximum barrier load.)

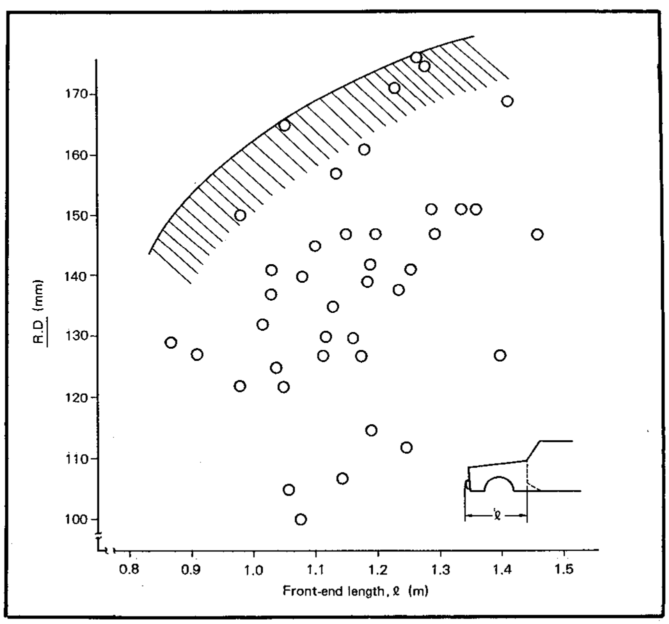

Figure 10. Relationship between front-end length and residual deformation (found from NCAP test data)

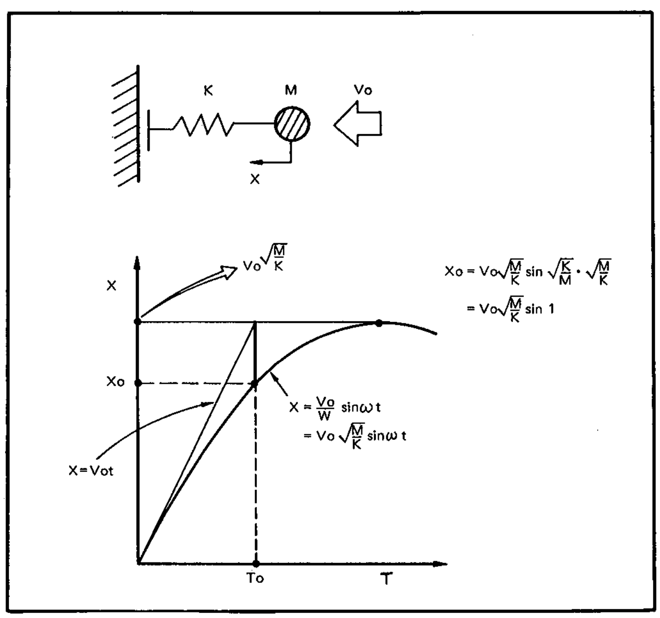

Figure 11. Relationship between residual deformation and vehicle specifications

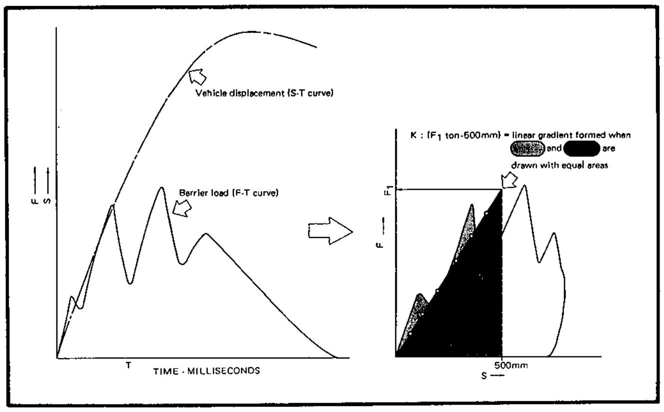

Figure 12. Definition of vehicle spring rate

In the above equation, the vehicle weight, M, was found by subtracting the weight of the dummy from the total weight at the time the test was conducted. Using this value calculated for each vehicle, the relationship between residual deformation and $\sqrt{\frac{M}{K}}$ was investigated and the results have been shown in the graph in Figure 13.

From the results in this figure, the following reason can be conceived for the variation in residual deformation when $\sqrt{\frac{M}{K}}$ represents certain given characteristics. The vehicle crash characteristics were represented as a simple linear spring rate (K) in order to treat them macroscopically. In actuality, however, the load-displacement curve (F-S characteristic) for existing vehicles is not a simple linear spring rate, as can be seen in Figure 12.

Several typical examples of F-S characteristics were then selected and their influence on residual deformation was investigated. Four types of variation characteristics were assumed relative to the baseline linear spring characteristic and their effect on residual deformation was examined. The results obtained are shown in Figure 14.

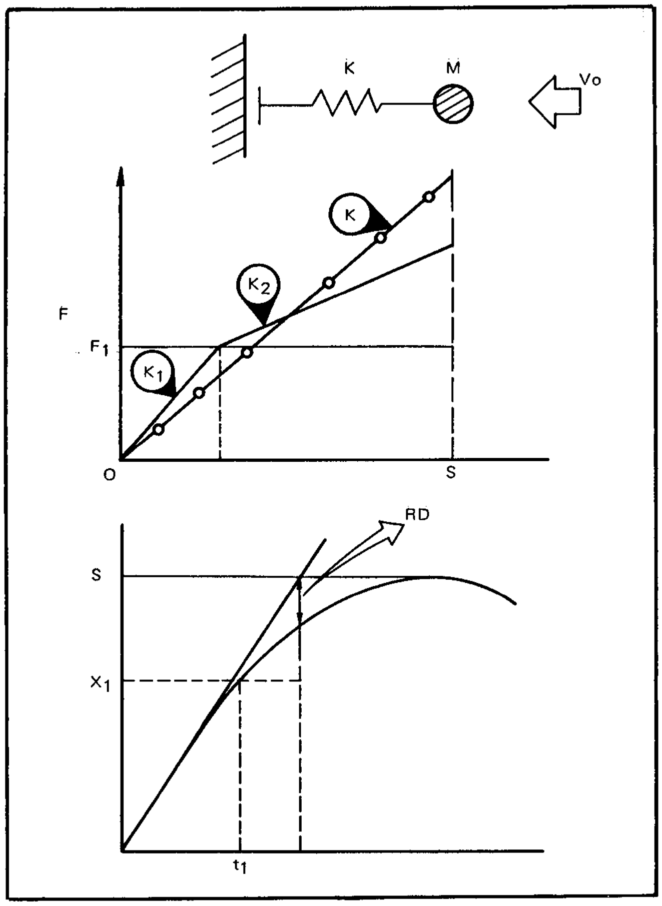

Figure 14. Fundamental relationship between vehicle spring rate and residual deformation

The results calculated for four specific cases relative to a baseline K up to a displacement of 500 mm have been plotted in Figure 15. The cases were: (1) $K_1 = \frac{2}{3}$ K; (2) $K_1 = K_2$; (3) $K_1 = 1.5$ K; and (4) $K_1 =$. It is seen from the results in the figure that residual deformation is greatly affected by changes in the initial characteristic (K_1), even when K is constant.

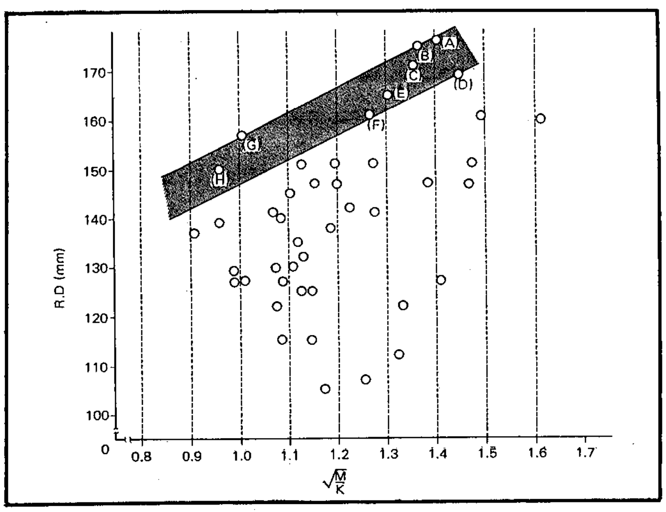

Figure 13. Relationship between vehicle characteristic $\sqrt{\frac{M}{K}}$ and residual deformation

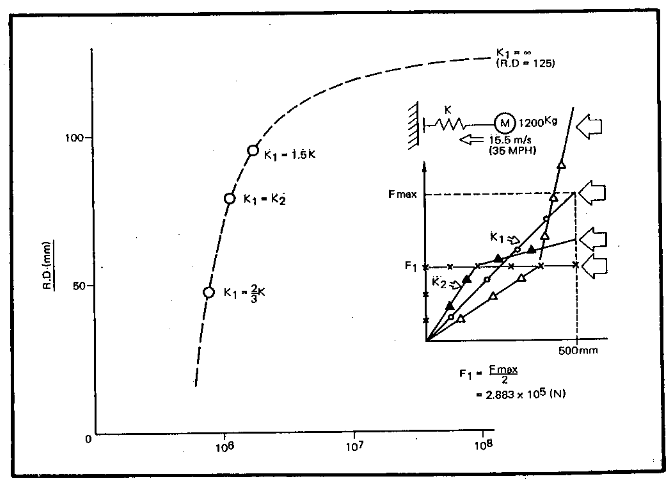

Figure 15. Change in residual deformation due to vehicle spring rate (K_1)

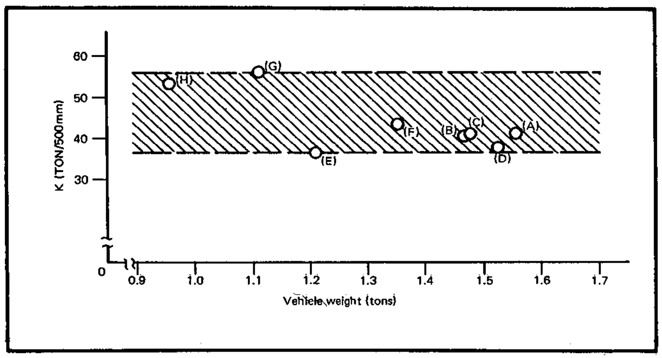

Figure 16. Relationship between vehicle weight and front-end spring rate when k = ton/500 mm

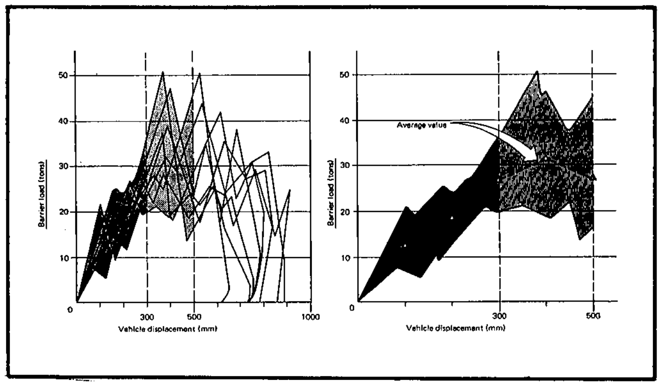

Figure 18. Relationship between barrier load and vehicle displacement (F-S diagrams)

In actuality, since K_1 of existing vehicles is even more complex, it can be assumed that variation will occur in residual deformation even when $\sqrt{\frac{M}{K}}$ is regarded as representing certain set characteristics, as shown in Figure 13.

From the foregoing results, it can be said that the vehicles forming the upper limit of the boundary zone in Figure 13 have good vehicle crash characteristics among the vehicles in use today. This indicates that it should be possible to utilize today's technology to control the front-end crash characteristics of existing vehicles so that they will fall within the boundary zone of this upper limit.

Next, an investigation was made of the relationship between vehicle weight and front-end stiffness for K up to a displacement of 500 mm. The vehicles examined were those forming a boundary zone in Figure 13, namely, those thought to have good crash characteristics. The results are shown in Figure 16.

From this figure it can be seen that vehicle front-end stiffness (K) falls within a certain range irrespective of vehicle weight.

The same conclusion was found when the relationship between K and vehicle weight was investigated for a displacement of 300 mm (Figure 17).

The foregoing results suggest that vehicle crash characteristics should be controlled, irrespective of vehicle weight, within the range of stiffness shown in Figures 16 and 17 in order to reduce injury to occupants of the striking vehicle in frontal collisions.

F-S diagrams like those shown in Figure 18 were also made for the vehicles forming the boundary zone in Figure 13. These diagrams were prepared from barrier load diagrams (F-T curve) for actual frontal collisions with a moving barrier and from vehicle displacement diagrams (S-T curve).

An average characteristic curve was then found from the F-S diagrams and a comparison was made with the characteristic values for existing DMBs. Those results are shown in Figure 19. The F-S characteristics calculated with the authors' method are relatively close to those proposed by the EEVC.

Example of a Stiffness Control Method for DMB Impacter

The previous section presented the results of investigations done on existing vehicles in an attempt to find a suitable balance for front-end stiffness. This section will describe the experiments carried out using the honeycomb impacter employed with the NHTSA DMB. In these experiments, an attempt was made to control stiffness by making holes and grooves in the original impacter so as to remove part of its volume.

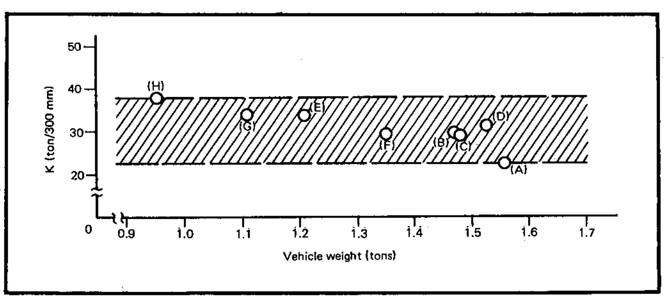

Figure 17. Relationship between vehicle weight and front-end spring rate when K = ton/300 mm

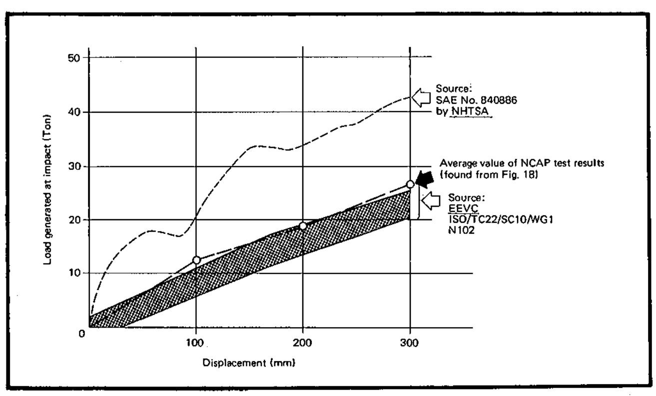

Figure 19. Comparison of load vs. displacement as found with existing DMB and from NCAP test results

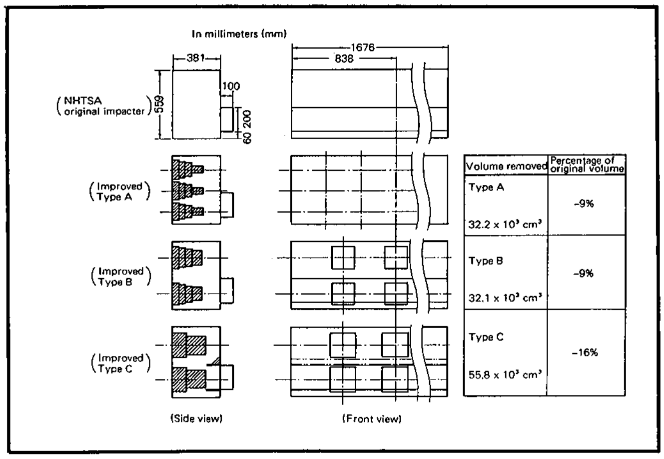

Figure 20. Example of improved DMB honeycomb impacter

The specifications of three types of improved impacters are shown in Figure 20. The relationship between the percentage of volume removed from the original impacter and the resulting stiffness (spring rate) for each type is shown in Figure 21. The load-displacement characteristics (F-S diagram) of these three types of impacters are shown in Figure 22 in comparison with those of existing DMBs. These characteristics were obtained in collisions involving the improved impacters and a moving barrier to which a load cell had been attached.

As the results in Figures 21 and 22 indicate, a certain degree of stiffness control has been achieved relative to the existing honeycomb impacter. It has been found that stiffness can be controlled within the targeted zone by using a combination of polyethylene foam and hard urethane foam. Further details concerning this material combination can be found in the

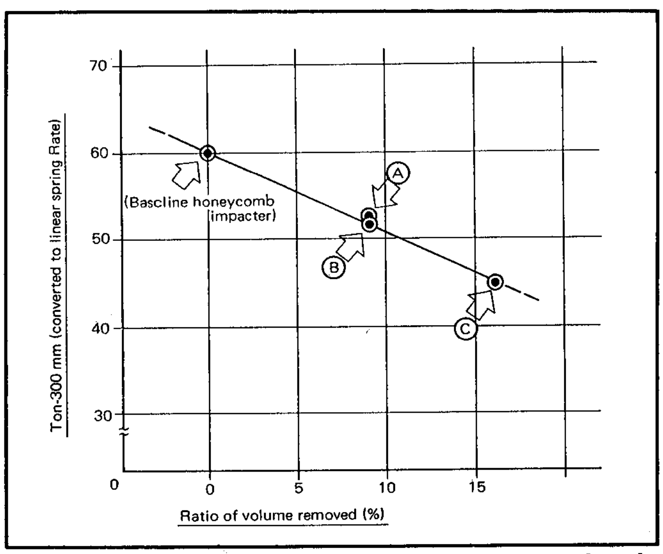

Figure 21. Relationship between percentage of volume removed from original honeycomb impacter and crash stiffness

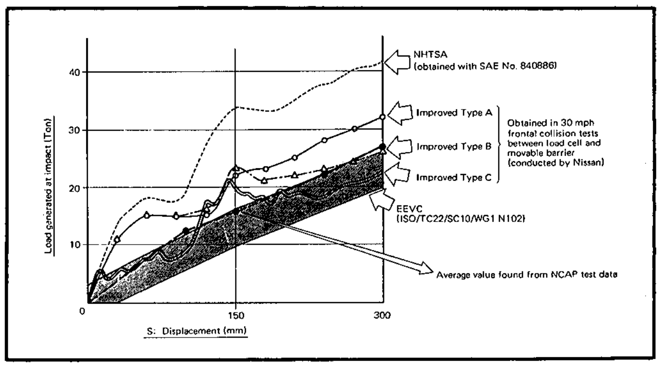

Figure 22. Comparison between F-S characteristics for improved honeycomb impacter and existing DMB

Japan Automobile Manufacturers' Association (JAMA) paper presented at this ESV conference.

Conclusion

This paper has presented the results of various analyses carried out to examine the relationship between vehicle front-end stiffness and dummy injury levels. These analyses were based on NCAP test report data for existing vehicles.

Assuring the safety of occupants in the striking vehicle in frontal collisions is, of course, an essential requirement. It is the authors' opinion that efforts must also be made to reduce the aggressivity of the striking vehicle toward other vehicles in rear-end and side impacts.

This investigation was carried out from the perspective of trying to achieve a suitable stiffness balance that would satisfy both requirements. The results obtained in the present work are summarized below.

(1) The injury level sustained by the dummy in the struck vehicle in side impacts and frontal collisions is greatly affected by the material strength of the DMB impacter.

(2) Residual deformation is an effective index for evaluating vehicle crash characteristics. Residual deformation can be found from a diagram showing the amount of vehicle displacement that occurs relative to time during a frontal collision with a barrier.

(3) There is a very strong correlation between residual deformation and dummy injury level. The latter tends to decrease as residual deformation increases.

(4) Residual deformation of existing vehicles is limited by vehicle size, including such factors as vehicle weight and front-end layout.

(5) Residual deformation can be controlled to some extent by varying the vehicle spring rate (vehicle crash characteristic K).

(6) Among vehicles in use today, those thought to possess optimum crash characteristics have a front-

end stiffness that falls within a certain set range irrespective of vehicle weight.

Postscript

If one considers front-end stiffness only in terms of aggressivity toward other vehicles in side and rear-end impacts, then the lower the vehicle spring rate is set, the better. However, a certain level of stiffness is naturally required in order to protect the occupants of the striking vehicle in frontal collisions, as well as to satisfy practical body strength requirements and meet existing safety standards. This means that front-end stiffness should be designed to provide a good balance between crashworthiness and aggressivity. The results of this investigation of existing vehicles suggest that the vehicle spring rate should be set within a certain range regardless of the vehicle weight. The range for 300 mm of displacement per ton is K = 22-38, while that for 500 mm of displacement per ton is K = 36-56.

Studies are now under way in many countries regarding different kinds of collision test methods in which a DMB is used to represent the impacting vehicle. The stiffness of the DMB impacter is a particularly important element in such tests. Thorough consideration should be given to the trade-off stiffness value which has been found using the approach described in this paper.

The present work has focused only on passenger cars. Similar research needs to be carried out for pickup trucks as well.

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Evaluation of the Safety Performance of Passenger Vehicles

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Abstract

The evaluation of the safety performance of passenger vehicles is in most test procedures restricted to the inherent safety. But in real-world accidents, the vehicle influences both its own passengers and the passengers of the colliding vehicle or the cyclists or the pedestrians. So test procedures that can only measure the performance of the vehicle and neglect the influence of the colliding vehicle will not be able to describe the safety performance completely.

Another problem is the separate evaluation of front-impact and side-impact. When side-impact protection is increased on the one hand side and the front of the vehicle becomes more stiff, due to increased front-impact protection, a high increase of safety will be measured, although no progress to side-impact protection may occur in real-world accidents.

By computer simulation, these effects can be described. A frontal impact with a movable deformable barrier only makes sense when the acceleration of the barrier is measured as well, and by mathematical simulation transformed to a dummy load of the "passenger of the struck vehicle". A side impact of the vehicle against a vehicle of the same type may show the "self-compatibility" and detect too stiff front-structures.

Objectives and Tools

The aim of this study is to assess the influence of various evaluation procedures on the development of passenger vehicles. The influence of the front-structure to the side impact is studied in this context. The frontal impact with a movable deformable barrier in particular is examined and compared to fixed barrier impact. The influence of impact velocity is examined.

Two aspects must be considered:

1. The protection of the passenger of a vehicle: This is the protection which a vehicle offers to its own passengers in different collision modes. In the context of the paper, this will be called "passenger-protection".
2. The protection of the occupants of the colliding vehicle and the other road-users (the partners on the road): This is the protection a vehicle offers to all the other road-users inside and outside of vehicles aside from its own passengers and in the different collision modes. In the context of the paper, this will be called "partner-protection".

But the concept of "partners" must be restricted to the other users of passenger-vehicles, the theme otherwise goes beyond the scope of a single paper.

Volkswagen dealt with this notion as far as accident statistics are concerned in the papers(3) and(8).

This study was undertaken by means of simulation of vehicle crashes and by structural optimization. The mathematical model to simulate front- and side-impact and the optimization strategy is described in details in(1),(7), and (10). The mathematical side-impact model is shown in fig. 1. The simulation of single-vehicle and of vehicle-to-vehicle crashes is possible with frontal and side-impacts. Thus a wide range of accidents occurring to passenger vehicles can be reproduced. Fig. 1 shows the configuration of a side-impact simulation. In the case of a frontal impact, the left frontal lumped-mass model of a vehicle is duplicated by its mirror-image. The simulation model determines the HIC and the head-, thorax-, and pelvis-acceleration of the dummies, and e.g. the different intrusions of the upper door and the sill.

The validation model was validated within the project, described in(1). It was validated in such a way that the "small" (740kg, 1630 p), the "medium" (950 kg, 2090 p), and the "large" (1230 kg, 2710 p) vehicle describe the average vehicle of each mass class. This and the following figures thus deal with "average" vehicles, and describe an "average" performance; they cannot be related to individual vehicles. The validation process is described in(1) and (5).

The Influence of the Front-Structure to Side-Impact Performance

It is obvious that in a 90-degree side impact the dummy load increases when the velocity of the striking vehicle increases. Fig. 2 shows this relationship for a side-impact of two small vehicles of 740 kg (1630 p). At an impact velocity of 9 m/s (20 mph) the dummy load is 31 g for the thorax and 40 g for the pelvis. At 15.5 m/s (35 mph) the dummy load increases to about 70 g for the thorax and 100 g for the pelvis. Fig. 3 shows the intrusion and the deformation of the vehicle. The intrusion of the side as well as the deformation of the front increases, in the range of 10 m/s (22 mph) up to 17 m/s (38 mph) nearly by the same amount. The total deformation of the vehicles is determined by both the side- and the front-structure of the colliding vehicles. This will be underscored by two following studies. The degree of influence of both structures will become apparent.

At first the degree of influence of the front-structure is studied. The side-impact now is simulated at a fixed impact-speed of 50 km/h (31 mph). The

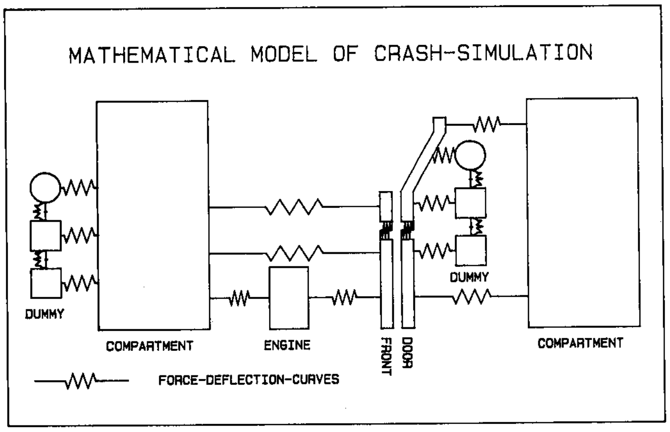

Figure 1

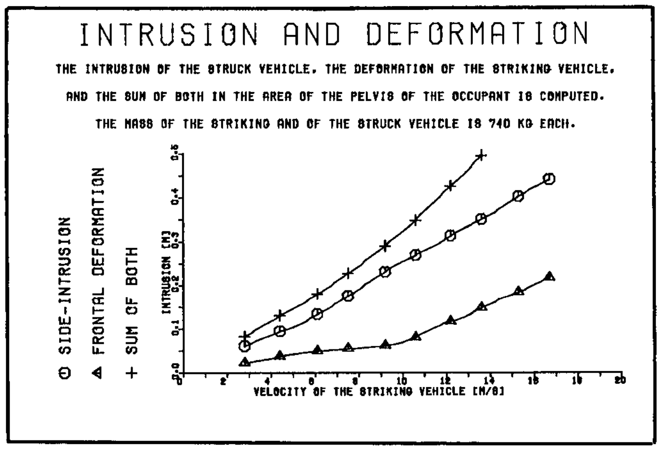

Figure 3

front-structure of the striking vehicle is changed. This is done by multiplying the forces of the force-deflection-curves by a constant factor that runs between 0.1 and 2.0. 0.1 means that the force-level of the front-structure is only 10% of the level that is observed today. 2.0 means that the present force-level of the front-structure is doubled. 1.0 refers to the present vehicles. The side-structure remains as it is today. Fig. 4 shows the dummy loads. At present force-level (1.0) there is a dummy load of 57 g for the thorax and 75 g for the pelvis. When force-levels are doubled, the dummy load increases to 75 g for the thorax and 100 g for the pelvis. When the force-levels are halved, the dummy load decreases to 25 g for the thorax and 35 g for the pelvis. That is even greater than the change of the dummy load that was previously determined, when the impact speed was increased from 9 m/s (20 mph) to 15.5 m/s (35 mph). Fig. 5 shows one of the reasons for this observation. At a force-level of 0.5, the intrusion of the struck vehicle is 31 cm (12 in). At a level of 1.0 this increases to 37.5 cm (14.8 in), and at a level of 2.0 it decreases to 45 cm (17.7 in). On the other hand, the deformation of the front-structure decreases much more: From 40 cm (15.7 in) at force-level 0.5 to 20 cm (7.9 in) at force-level 1.0, and 5 cm (2.0 in) at force level 2.0. Thus the front-structure offers much less deformation, when the force-level increases, and, additionally, the force-level at the moment of impact between dummy and structure is higher. This shows that the design of the front-structure is an important factor in side-impacts. It will be shown below that this is a conflict of aims with passenger-protection in frontal impacts. Measures implemented into the front-structure to achieve a good protection in side-impacts are a good example of partner-protection. The passengers of the vehicle with a front-structure, designed to offer side-protection, never will derive advantages from this measure, and personally they will never have an interest in such measures. On the other hand, when at least two vehicles of that model are built, they may indirectly take advantage from the other model of the same type, when that vehicle strikes their vehicle

The influence of the front-structure in the performance of vehicles in the case of side-impacts is not only an academic problem. In an effort to achieve better performance in frontal impacts, in the U.S. the velocity of the barrier impact has been increased to 35 mph in the "New Car Assessment Program (NCAP)". Manufacturers are indirectly forced to fulfill a higher barrier impact velocity (35 mph instead

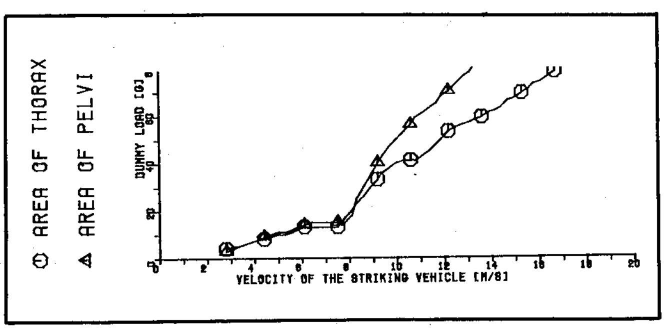

Figure 2

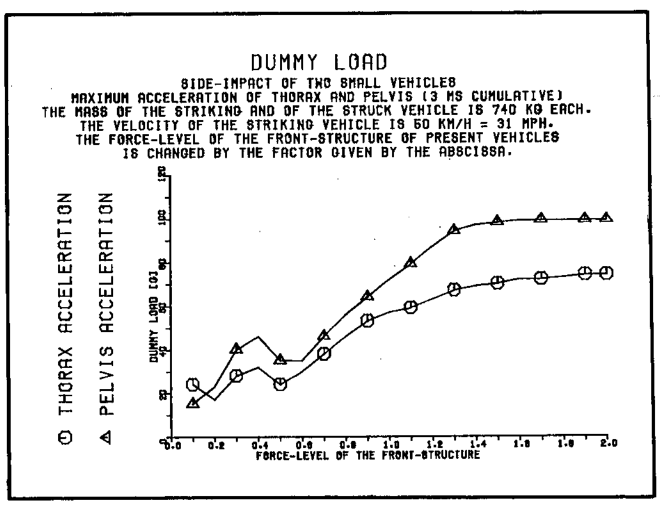

Figure 4

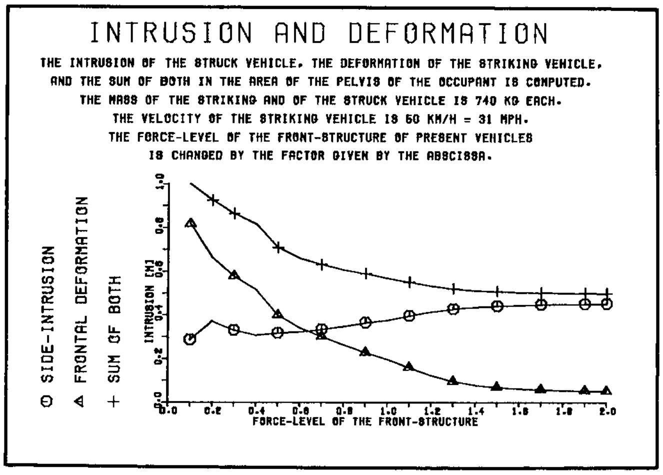

Figure 5

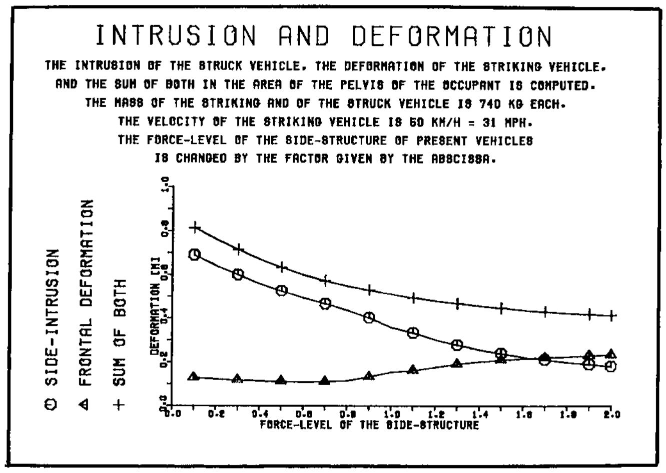

Figure 7

of 30 mph). When, in such a case, deformation length is not changed, this means a change of the force-level of the front-structure that is proportional to (35*35)/(30*30) = 1.36, the ratio of the kinetic energies. The results of such a measure in terms of increased dummy load in side-impact can also be seen in fig. 4; thorax acceleration increases from 57 g to 70 g and pelvis acceleration from 75 g to 97 g.

Side-protection will normally be achieved by measures in the side-structure of a vehicle. This was also simulated. Present force-levels of the side-structure now were changed by a predefined factor and the front-structure was kept at its present level. The same vehicles and the same impact-velocity was examined. Fig. 6 shows the result for the dummy load and fig. 7 shows the influence of the force-level of the side-structure on intrusion and deformation.

From these figures one may derive that the change of the front-structure may be compensated by a change of the side-structure. This is true up to a certain point. But one should realize what happens. Investments in the side-structure of a vehicle are made, not to achieve better performance in side-protection, but only in order to achieve the same performance levels as before when the front-structures of vehicles were less stiff. It is not easy to explain to a customer why he has to pay more for the same thing.

Fig. 8 and fig. 9 show this relationship between front- and side-structural stiffness for thorax and pelvis. These figures answer the question, how to achieve a certain dummy load at an impact-velocity of 50 km/h (31 mph) when the front-structure of the striking vehicle is modified. It can be seen that it is nearly the same amount of stiffness change that is required in the side-structure to compensate the change in the front-structure.

The curves of fig. 9 are not completely linear. The reason is oscillation in the system compartment, door/front, compartment (cf. fig. 1).

Principles of the Evaluation of Test-Regulations Regarding Compliance and Rating of Passenger Vehicles

If a legislative body or the community prescribes a test regulation, for example for the compliance and

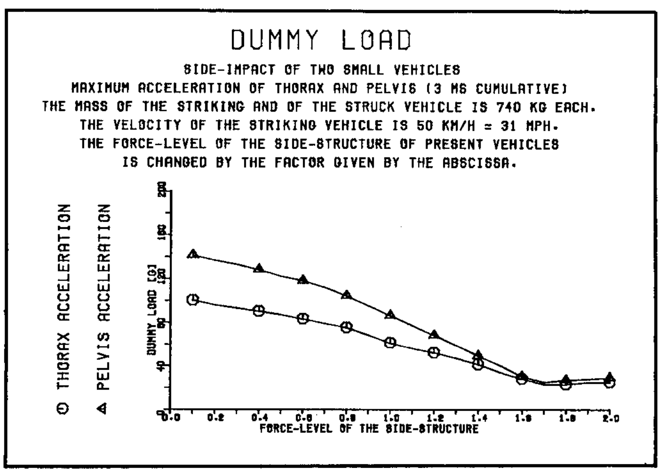

Figure 6

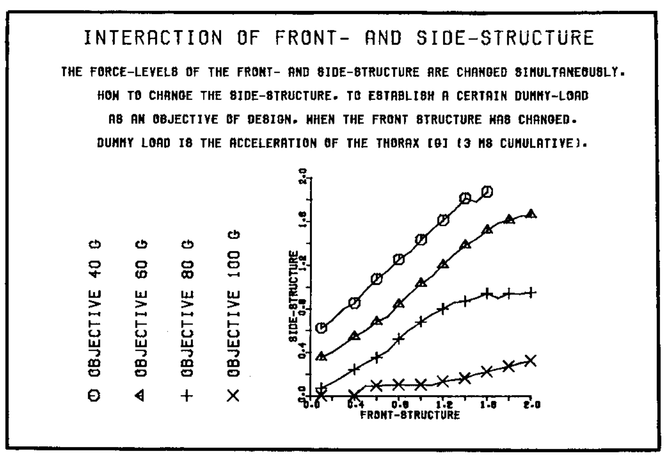

Figure 8

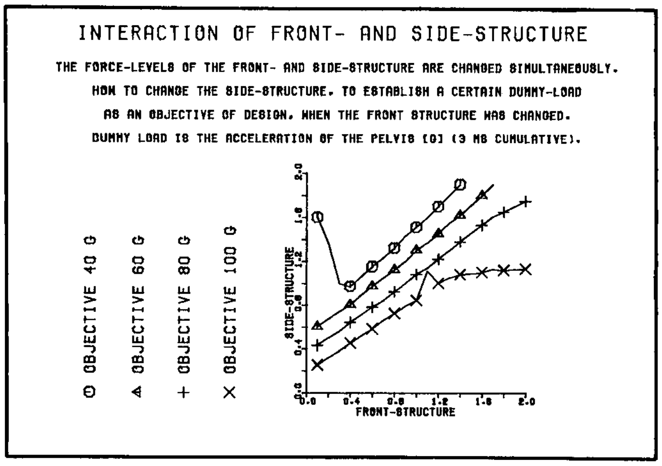

Figure 9

rating of passenger vehicles, this of course then has a profound effect on further vehicle development. It must be established which vehicles are considered best under the conditions of the tests prescribed and how the vehicles behave in accidents as a whole. This is possible with the aid of an optimizer coupled with programs simulating the tests. Such an optimizer investigates possible structural designs under the aspect of a target size, just as a structural design of a vehicle is subjected to a test in the real development process and is then either pursued further if considered particulary good, coming off well in terms of the rating, or constituting an improvement over the conventional solutions. Solutions which represent a deterioration with respect to the state of the art are not pursued further, just as they would be discarded by an optimization procedure.

The optimization process specifically applied and described below also emulates this development process in that it always retains a "store of experience" of the 30 best solutions "in its memory" and combines components of one solution with components of another solution of this store so as to find improvements. In addition, components are also chosen completely afresh according to random factors so as also to take the creative process into account. All in all, this imitates procedures in the real world of experience and creative work running in parallel, contributing equally to an improvement of existing solutions. This procedure was therefore used to examine the effect of various design goals. Approximately 70,000 individual simulations were carried out and more than 20 CP hours computing time on a Cyber 990.

This optimization program was originally developed to compute optimized vehicle structures for the three mass classes of "small", "medium", and "large" vehicles that show an optimal performance in an accident mock-up of 57 crashes. The mass classes and the collisions were designed so that the present accident scene is described as well as possible. (For details cf (1) or (2), (7), and (10)).

Fig. 10 depicts the deformation structure of the present vehicle fleet and fig. 11 the structure of the optimized vehicle fleet. Table 1 shows the performance of the present vehicle fleet in some collision types. Table 2 shows the performance of the optimized vehicle fleet at its maximum design-speed for frontal and side-impact, compared to the present vehicle fleet. The progress is obvious and impressive. Thus even in the collision of a "small" vehicle (740 kg, 1630 p) with a "large" vehicle (1230 kg, 2710 p) at a closing velocity of 99.4 km/h (62 mph), the HIC in the optimized "small" vehicle is less than 800.

The best performance in the optimized fleet can be seen in side-impact. This observation underscores the fact that side-protection together with an adequate design of the front-structure allows a high level of passenger-protection in the event of side-impacts.

It should be noted that each vehicle is an average vehicle of the respective weight class and that the investigations were carried out in 1980. Since these

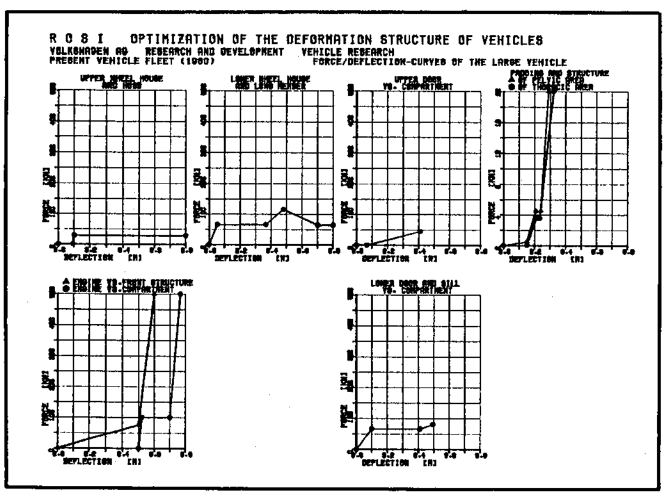

Figure 10

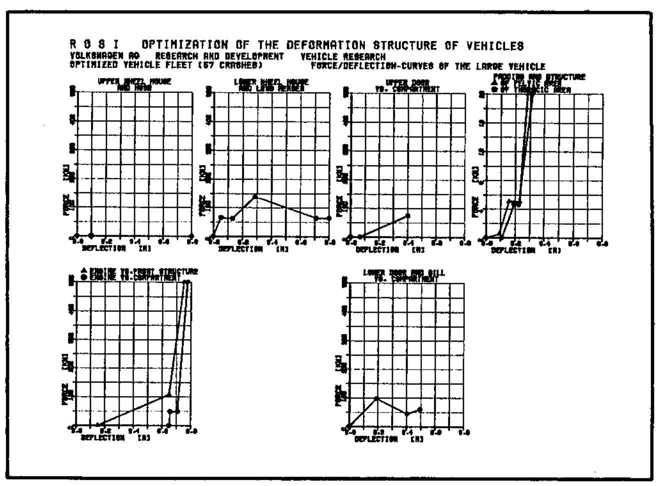

Figure 11

vehicles only fulfilled the function of creating initial situations for subsequent optimization, this fact is of no significance for further deliberations.

Design of Passenger Vehicles for Optimum Behaviour in a Frontal Fixed Barrier Impact

The current situation in the design of frontal structures is determined by the fixed-barrier-impact tests at 50 km/h or 31 mph and 56 km/h or 35 mph. A vehicle is assessed according to its behaviour in these tests without consideration of how the design regulation affects other types of collisions. Consequently, in an initial optimization run, a structure was established for the large vehicle which is an optimum with regard to the 50 km/h (31 mph) fixed barrier impact. This structure is shown in Fig. 12. In a second optimization run, the structure was optimized with regard to the 56 km/h (35 mph) fixed barrier impact. This structure is depicted in Fig. 13. The results of 6 frontal and 3 side collisions with participation of the modified large vehicle are compiled in Table 3.

In frontal impacts with a closing-velocity of 100 km/h (62.1 mph) the performance of both structures does not differ significantly and is better than the performance of present vehicles. At a closing-velocity of 112 km/h (70 mph) the same is true. In the first case, the vehicle designed for 50 km/h (31 mph) offers the better passenger-protection, in the second case the vehicle designed for 56 km/h (35 mph). In terms of partner-protection, both vehicles exhibit nearly the same levels. Table 3 shows three side-impact collisions, when the optimized "large" vehicle is the striking vehicle and the side-structure of the struck vehicle is not modified. The performance of these optimized front-structures is far away from that, which is possible when the vehicles are optimized with a realistic accident-environment in mind (e.g. the 57

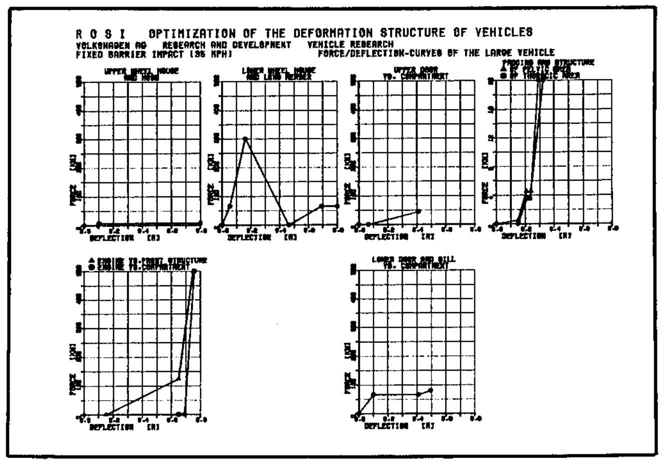

Figure 13

crashes, table 2). It is even worse than the performance of the present vehicle fleet.

Optimization of Vehicles with the Movable Deformable Barrier (Passenger-Protection

Efforts are being made to supplement the fixed-barrier-impact test by a frontal impact test with a movable barrier. This is quite obviously of advantage for a large vehicle of 1230 kg, for example, which in an impact with a 1000 kg barrier at 60 mph closing velocity, for example, has only to withstand a calculated velocity change of 26.9 mph. In the same test with a small vehicle of 740 kg, for example, a calculated velocity change of 34.5 mph results. This is well known and does not need to be constantly confirmed by full scale testing. The question remains, therefore, whether the test with a movable barrier places other demands on the structure of a vehicle, hence providing additional information compared to the fixed-barrier-impact test. The structure obtained if the large vehicle is optimized for behavior in the event of impact with a movable barrier is shown in Figs. 14

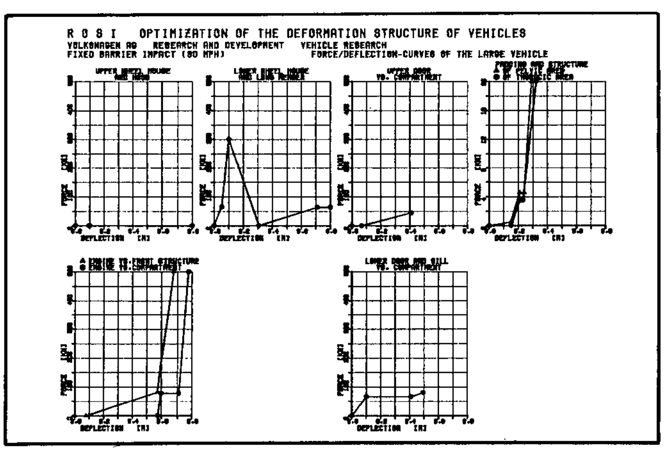

Figure 12

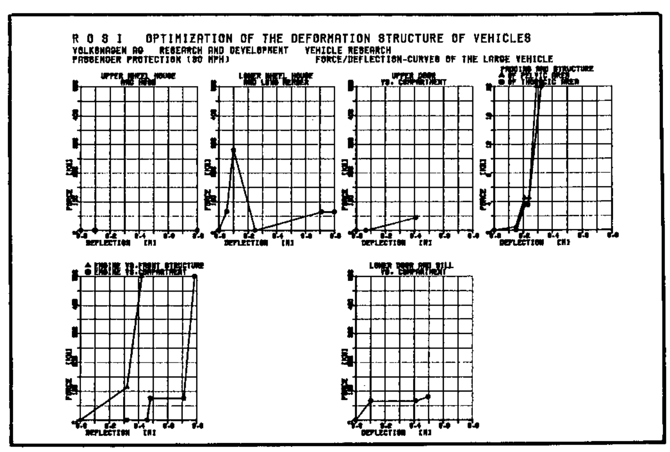

Figure 14

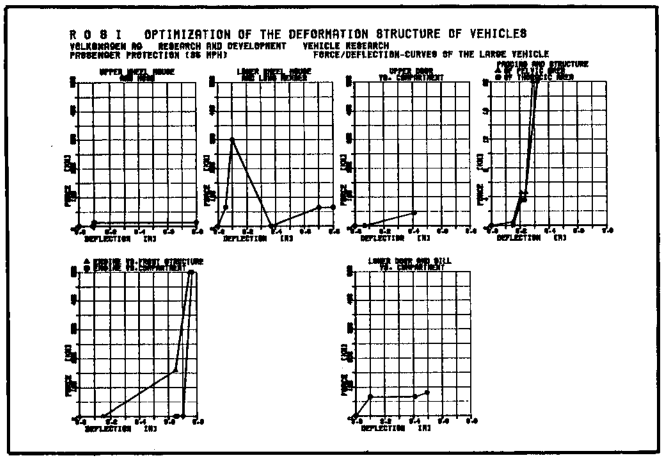

Figure 15

and 15. The movable barrier was substituted in this case for a vehicle of 950 kg (2090 p), the medium vehicle on the roads of the Federal Republic of Germany in 1980. The closing velocity in the first case (Fig. 14) is 60 mph and in the second case (Fig. 15) 70 mph. The member characteristic has a similar form to that of the fixed-barrier-impact test: an early high peak and then a collapse. The peak is shorter in the case of a 30 mph design (60 mph closing velocity) than for a 35 mph design (70 mph closing velocity). There are considerable differences in the distribution of the deformation energy to the regions in front of and behind the engine. In the case of the fixed barrier impact, a configuration was shown to be the optimum for both test velocities in which the deformation energy is introduced exclusively in the front/engine region, resulting in considerable deceleration of the engine before impact with the barrier (or the other party to the accident). Central positioning of the engine and equal distribution of the deformation energy in the region in front of and behind the engine is recommended for the 30 mph design. In the case of the 35 mph design (70 mph closing velocity), the distribution obtained is the same as for the fixed barrier impact, but at a lower force-level. These considerable differences show that the impact with a movable barrier makes use of the very special shape of the deformation response of the other fixed-barrier-impact test, and makes it imperative that there be special management of the deformation in the front/engine/cell chain. This is highly dependent, however, on the velocity and on the structure of the other party to the accident. All in all, it can be established in these investigations that the movable barrier rewards highly specialized measures for occu-

Table 1.

	STRUCTURAL OPTIMIZATION: PRESENT VEHICLES							
COLLISION MODE	HEAD				THORAX		PELVIS	
	HIC		ACC		ACC		ACC	
FRONTAL IMPACT VC= 100 KM/H = 62,5 MPH CLOSING-VELOCITY								
LARGE/LARGE	718	718	74	74	42	42	39	39
MEDIUM/LARGE	851	631	76	72	44	41	41	37
SMALL/LARGE	1314	551	90	71	57	39	50	37
FRONTAL IMPACT VC= 112 KM/H = 70 MPH CLOSING VELOCITY								
LARGE/LARGE	1431	1431	94	94	54	54	47	47
MEDIUM/LARGE	1668	1119	100	85	60	46	51	40
SMALL/LARGE	2488	914	117	82	71	44	60	39
SIDE IMPACT VA= 50 KM/H = 31,2 MPH IMPACT-VELOCITY								
LARGE/LARGE	262	165	54	29	59	22	89	24
MEDIUM/LARGE	321	132	58	26	64	21	94	23
SMALL/LARGE	400	102	63	23	70	20	99	22

Table 2. The dummy loads in some impacts of the large vehicle. A comparison between the current structure and an optimized structure.

COLLISION	DUMMY-LOAD					
FRONTAL IMPACT VC = 99,4 KM/H (VC = 62,1 MPH)	HIC		THORAX-ACC. [G]		PELVIS-ACC. [G]	
	PRESENT	OPTIMIZED	PRESENT	OPTIMIZED	PRESENT	OPTIMIZED
LARGE/LARGE	706/706	461/461	42/42	35/35	39/39	30/30
MEDIUM/LARGE	842/618	535/462	44/41	36/31	41/37	32/30
SMALL/LARGE	1275/521	772/485	56/39	43/35	50/37	40/31
SIDE-IMPACT VC = 46,1 KM/H = 28.8 MPH STRUCK/STRIKING						
LARGE/LARGE	262/165	37/130	59/22	25/22	89/24	38/24
MEDIUM/LARGE	321/132	69/103	64/21	38/20	94/33	58/22
SMALL/LARGE	400/102	35/109	70/20	50/20	99/22	65/22

Table 3.

STRUCTURAL OPTIMIZATION: BARRIER-IMPACT

COLLISION MODE	******* 30 MPH TEST-VELOCITY **********								********* 35 MPH TEST-VELOCITY ********							
	HEAD				THORAX		PELVIS		HEAD				THORAX		PELVIS	
	HIC		ACC		ACC		ACC		HIC		ACC		ACC		ACC	
FRONTAL IMPACT VC= 100 KM/H = 62,5 MPH CLOSING-VELOCITY																
LARGE/LARGE	420	420	52	52	34	34	26	26	422	422	70	70	35	35	39	39
MEDIUM/LARGE	766	360	75	52	43	34	41	26	689	495	75	73	42	37	41	44
SMALL/LARGE	957	362	79	35	44	24	43	27	927	429	79	70	44	35	43	40
FRONTAL IMPACT VC= 112 KM/H = 70 MPH CLOSING VELOCITY																
LARGE/LARGE	1437	1437	84	84	48	48	45	45	497	497	61	61	33	33	32	32
MEDIUM/LARGE	1322	795	88	75	49	44	43	36	1098	631	86	67	45	40	41	35
SMALL/LARGE	2045	599	107	70	61	41	54	33	1701	563	104	66	60	37	52	34
SIDE IMPACT VA= 50 KM/H = 31,2 MPH IMPACT-VELOCITY																
LARGE/LARGE	1004	163	70	32	68	24	106	27	364	163	57	32	58	25	94	28
MEDIUM/LARGE	757	132	72	30	70	24	105	26	417	135	59	30	62	24	96	27
SMALL/LARGE	677	110	74	28	71	23	103	25	479	95	62	27	67	22	98	25

pant protection against the specific structure of the barrier and is less inductive of generally useful measures. The dummy loads are shown in table 4.

Optimization of Vehicles with the Movable Deformable Barrier (Partner-Protection)

When vehicles are tested by allowing them to collide frontally with a movable, deformable barrier, it is a comparatively trivial demand to measure and evaluate not only the loading in the vehicle but also the loading in the barrier represents the—possibly innocent—other party to the accident, whose freedom from injury is something which is just as worthy of protection as the safety of the occupants of the vehicle being evaluated. In order to examine the determining action mechanisms for partner-protection, an optimization run was calculated in which the structure of the large vehicle was designed exclusively with the optimum protection of the occupants of an impacted medium vehicle in mind. In one case, a closing velocity of 60 mph was used as the bases, and in the other, 70 mph. The proposed structure of the large vehicle is shown in Figs. 16 and 17.

It is immediately obvious that the member characteristic is consistently of the same shape as for occupant protection: an early high peak (3200 kN at 150 mm or 100 mm) and then relief. The engine is now systematically designed in this manner hardly result in any improvement for the occupant of the medium vehicle at 30 mph (Table 5), but disadvantages arise for the occupant of the large vehicle. At 35 mph, the advantages for the occupant of the large vehicle in particular are more pronounced. With respect to lateral protection, the structures designed for partner-protection prove to be considerably less aggressive; the HIC in the laterally impacted vehicle, which was over 1000 for the vehicle concentrating entirely on occupant protection, falls to below 600 and thorax and pelvis retardation drop by 5 g – 10 g. A considerable advance is achieved compared with current vehicles as far as the specific aim of partner-protection in frontal collisions is concerned, but this is

Table 4.

STRUCTURAL OPTIMIZATION: PASSENGER PROTECTION

COLLISION MODE	******* 30 MPH TEST VELOCITY **********								******* 35 MPH TEST-VELOCITY **********							
	HEAD				THORAX		PELVIS		HEAD				THORAX		PELVIS	
	HIC		ACC		ACC		ACC		HIC		ACC		ACC		ACC	
FRONTAL IMPACT VC= 100 KM/H = 62,5 MPH CLOSING-VELOCITY																
LARGE/LARGE	587	587	68	68	40	40	33	33	390	391	52	52	29	29	29	29
MEDIUM/LARGE	710	330	76	54	42	32	41	24	699	389	75	47	42	28	41	32
SMALL/LARGE	1209	304	89	47	54	25	49	47	991	229	80	35	48	28	43	34
FRONTAL IMPACT VC= 112 KM/H = 70 MPH CLOSING VELOCITY																
LARGE/LARGE	1647	1639	96	96	51	51	47	47	737	736	65	65	40	40	31	31
MEDIUM/LARGE	1658	1225	103	81	59	45	51	41	1119	478	86	60	45	37	41	29
SMALL/LARGE	2193	807	117	74	71	42	60	34	1651	473	104	59	59	35	52	30
SIDE IMPACT VA= 50 KM/H = 31,2 MPH IMPACT-VELOCITY																
LARGE/LARGE	1259	163	75	29	73	22	109	24	858	157	71	32	70	25	106	28
MEDIUM/LARGE	1070	138	77	27	74	21	107	23	738	132	72	30	72	24	105	27
SMALL/LARGE	1026	105	78	24	76	20	107	22	637	114	74	29	73	23	103	26

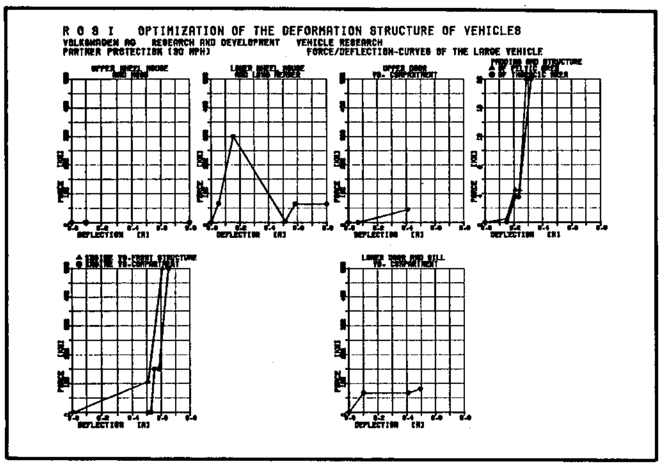

Figure 16

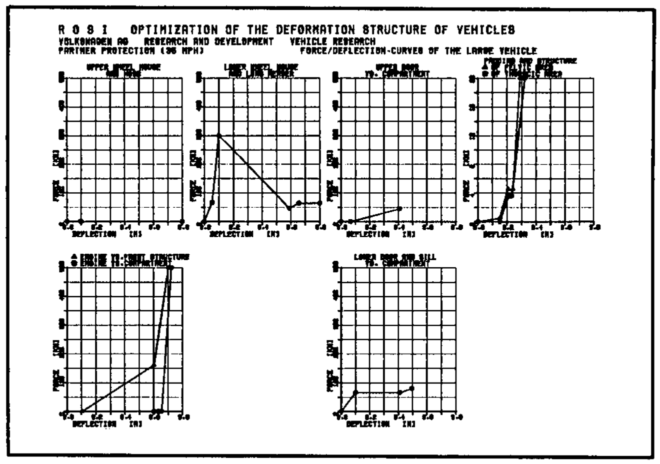

Figure 17

at most 5 g to 10 g higher than that which is achieved as a positive side-effect for other road users if occupant protection is implemented to the full.

This result also shows, however, that occupant and partner-protection need not be opposites as far as frontal impacts are concerned. If certain fundamental principles are observed, particularly with regard to engine position and the possibility of deformation between the engine and the front, it is quite possible to harmonize the two since the shapes of the long member characteristics differ only slightly.

Integration of Passenger-, Partner-, and Side-Protection

From the deliberations which have gone before, the idea suggests itself to integrate passenger-, partner-, and side-protection in one approach. This was attempted in a further optimization run: the front and side-structures of the large vehicle were released for optimization. The target was a weighted combination of the dummy loads which occur in the event of three collisions: The fixed barrier impact was weighted with the frequency of single-vehicle accidents, the collision between the large and medium vehicle with the frequency of vehicle-to-vehicle accidents and the side collision between two large vehicles with the frequency of side-collisions. The following frequencies were used(1):

0.345 fixed barrier impact
0.444 frontal impact medium/large vehicle
0.211 side-impact large/large vehicle
1.000

The factor of 0.444 for frontal impact between the medium and the large vehicle was applied half to the dummy load in the medium vehicle and half to the large vehicle. The dummy load used is a combination of HIC and acceleration over 3 ms for thorax and pelvis of the driver. The loads are combined in accordance with the procedure described by Gillis(4) according to the formula:

DBW = HIC + A THORAX * 12.525 + A PELVIS * 8.35

Table 5.

COLLISION MODE	STRUCTURAL OPTIMIZATION: PARTNER-PROTECTION ********** 30 MPH TEST-VELOCITY ******* HEAD HIC		HEAD ACC		THORAX ACC		PELVIS ACC		******* 35 MPH TEST-VELOCITY ******* HEAD HIC		HEAD ACC		THORAX ACC		PELVIS ACC	
FRONTAL IMPACT VC= 100 KM/H = 62,5 MPH CLOSING-VELOCITY																
LARGE/LARGE	681	681	75	75	40	40	41	41	915	915	81	81	46	46	44	44
MEDIUM/LARGE	680	807	75	79	42	43	41	43	643	1068	75	83	41	50	41	46
SMALL/LARGE	1033	604	82	76	50	40	46	42	899	826	80	81	46	44	43	44
FRONTAL IMPACT VC= 112 KM/H = 70 MPH CLOSING VELOCITY																
LARGE/LARGE	786	786	75	75	43	43	37	37	905	905	84	84	43	43	42	42
MEDIUM/LARGE	1170	880	86	78	47	44	42	39	913	996	86	86	44	44	41	43
SMALL/LARGE	1848	812	107	76	62	44	54	38	1551	958	102	87	57	44	50	43
SIDE IMPACT VA= 50 KM/H = 31,2 MPH IMPACT-VELOCITY																
LARGE/LARGE	469	173	63	30	63	23	99	25	798	139	70	30	68	24	105	26
MEDIUM/LARGE	521	103	64	27	66	22	101	24	785	113	71	28	70	23	105	26
SMALL/LARGE	581	106	66	24	70	20	101	22	669	97	73	27	71	22	103	25

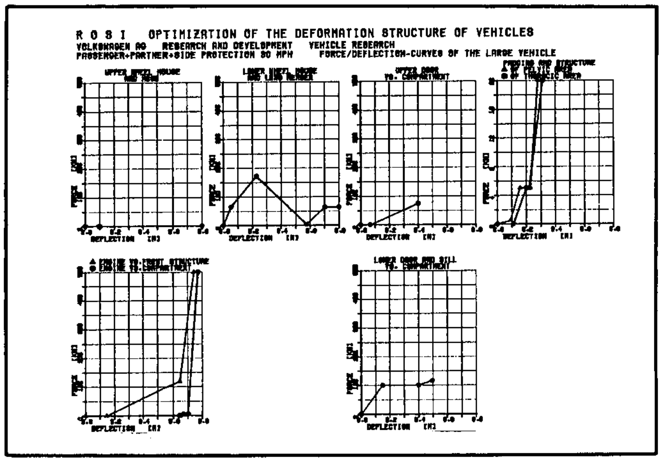

Figure 18

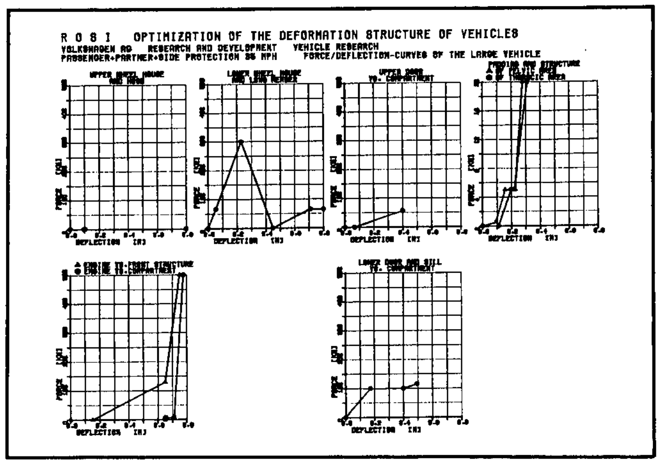

Figure 19

The design velocity chosen was 50 km/h for side-impact, and for frontal impact in the first optimization run, 30 mph for the fixed barrier and 60 mph for the vehicle-to-vehicle collision, and in the second optimization run, 35 mph for the fixed barrier and 70 mph for the vehicle-to-vehicle collision. Figs. 18 and 19 show the structural design proposed by the optimizer in these cases. In principal, the side-structure in both instances is the same. The engine is shifted uniformly to the rear and deformation displaced to the engine/front region. The shape of the member characteristic is different, dependent on the design. At a design velocity of 30 mph it is flatter (maximum force-level at 220 mm and 170 kN). The subsequent dropping away is also not so pronounced, and does not occur until 580 mm. In the case of the 35 mph design, the front is stiffer. The peak is also reached at 220 mm, but the force-level is 300 kN and the member characteristic already drops away at 450 mm.

This is reflected in the results, which are shown in Table 6.

Possibilities of an Abbreviated Integrated Test Procedure

The test procedure developed in Section 7 requires three tests with at least three identical vehicles. The extent to which the vehicle-to-vehicle frontal-impact test is necessary to achieve the desired result will be examined below. To do this, the same optimization process was repeated with the following collisions:

Frontal fixed barrier impact
Lateral vehicle-to-vehicle collision

The frontal vehicle-to-vehicle impact was deleted. Since the frontal fixed barrier impact now represents all frontal crashes of the large vehicle, it was weighted at 62%, and the side-impact at 38%.

Figs. 20 and 21 show the characteristic curves for frontal design velocities of 30 mph and 35 mph. In the side-region, the characteristic curves obtained are, in principle, the same as those in the optimization runs described in Section 7. The engine position is also identical. In the design for a 30 mph fixed barrier

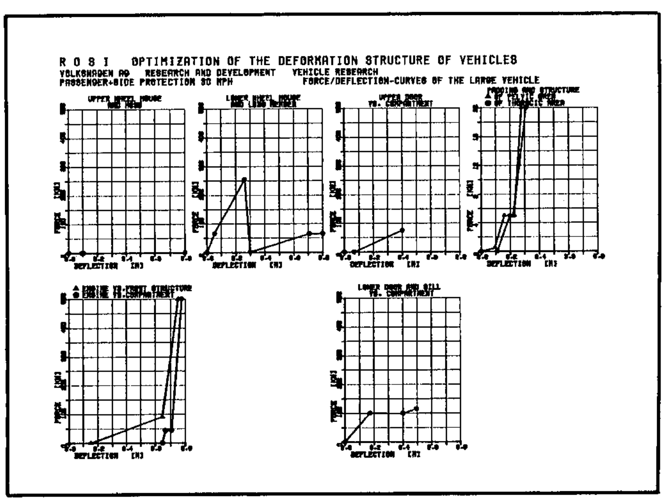

Figure 20

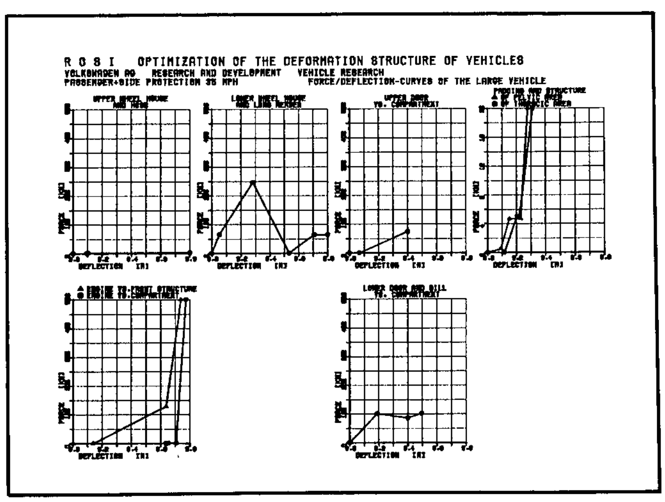

Figure 21

Table 6.

STRUCTURAL OPTIMIZATION: BARRIER-TEST, FRONTAL IMPACT AND SIDE-IMPACT

COLLISION MODE	********** 30 MPH TEST-VELOCITY ******* HEAD HIC		HEAD ACC		THORAX ACC		PELVIS ACC		******* 35 MPH TEST-VELOCITY ******* HEAD HIC		HEAD ACC		THORAX ACC		PELVIS ACC	
FRONTAL IMPACT VC: 100 KM/H = 62,5 MPH CLOSING-VELOCITY																
LARGE/LARGE	493	493	54	54	33	33	26	26	347	347	64	64	30	30	38	38
MEDIUM/LARGE	739	430	74	58	42	35	40	29	667	372	74	66	41	31	41	39
SMALL/LARGE	949	330	79	53	44	29	43	25	916	250	79	41	44	29	43	36
FRONTAL IMPACT VC: 112 KM/H = 70 MPH CLOSING VELOCITY																
LARGE/LARGE	1712	1711	114	114	58	58	47	47	473	473	59	59	33	33	31	31
MEDIUM/LARGE	1698	965	91	80	48	45	46	39	1080	621	86	67	45	41	41	33
SMALL/LARGE	1667	826	103	79	59	44	51	38	1656	554	103	66	59	40	51	32
SIDE IMPACT VA: 50 KM/H = 31,2 MPH IMPACT-VELOCITY																
LARGE/LARGE	33	180	19	32	32	26	48	30	89	180	31	32	44	26	61	30
MEDIUM/LARGE	190	136	46	27	57	22	90	24	285	115	51	28	58	23	92	25
SMALL/LARGE	255	105	50	24	62	21	95	23	345	80	54	25	64	21	96	24

impact, the deformation energy in the engine-front region is now increased by approximately 20% and the member characteristic given a higher peak with immediate drop: a peak of 250 kN at 250 mm, already dropping away at 300 mm. The more regular curve observed in the case of the 30 mph optimization in Section 7 is therefore abandoned in favour of a more pointed curve. In the 35 mph design, the engine characteristics also agree with those obtained in Section 7 and the member characteristic is somewhat less pointed than before. It drops away somewhat later.

The dummy loads resulting from these characteristic curves are shown in Table 7. They differ only insignificantly from the figures given in Table 6 for the more complex optimization. The structure designed for 30 mph in the collision with 70 mph closing velocity constitutes an exception. In this case, the member characteristic is very closely matched to the energy expectations of a 30 mph fixed barrier impact, as a result of which there is no further deformation energy available at higher impact velocities and the very high forces take effect via the high level of cell stiffness. For this reason, in a revised calculation, the member characteristic was altered such that it drops away more moderately not until 590 mm (Fig. 22). In this way, more deformation energy is available. The changed structure proved to be less aggressive only at a closing velocity of 100 km/h compared with the medium and small vehicles, but resulted in higher loads in the large vehicle in the fixed barrier impact (i.e. the impact of the large vehicle with the large vehicle) (Table 8). Passenger-protection is reduced at the closing velocity of 100 km/h. Pelvis load in the large vehicle in collision with the small vehicle is then 39 g in the revised calculation compared with 27 g.

At a collision velocity of 70 mph, considerably better results are obtained across the board, because the modified structure now possesses the necessary deformation energy. In the side-impact the dummy loads fully correspond because the front structure was changed in a region which has no further influence on side-impact at the velocity under consideration.

Table 7.

STRUCTURAL OPTIMIZATION: BARRIER-TEST AND SIDE-IMPACT

COLLISION MODE	********** 30 MPH TEST-VELOCITY ******* HEAD HIC		HEAD ACC		THORAX ACC		PELVIS ACC		******* 35 MPH TEST-VELOCITY ******* HEAD HIC		HEAD ACC		THORAX ACC		PELVIS ACC	
FRONTAL IMPACT VC: 100 KM/H = 62,5 MPH CLOSING-VELOCITY																
LARGE/LARGE	501	501	54	54	37	37	31	31	404	403	69	69	34	34	38	38
MEDIUM/LARGE	786	454	75	59	43	38	41	30	671	371	75	66	42	31	41	38
SMALL/LARGE	956	349	79	55	44	32	43	27	915	291	79	39	44	28	43	34
FRONTAL IMPACT VC: 112 KM/H = 70 MPH CLOSING VELOCITY																
LARGE/LARGE	2358	2357	139	139	63	63	51	51	524	524	63	63	37	37	32	32
MEDIUM/LARGE	1777	999	93	81	52	45	47	39	1108	691	86	71	45	43	41	34
SMALL/LARGE	1735	850	102	79	58	44	52	38	1691	612	104	71	59	41	52	34
SIDE IMPACT VA: 50 KM/H = 31,2 MPH IMPACT-VELOCITY																
LARGE/LARGE	40	182	22	32	34	27	54	31	48	174	24	31	37	26	57	29
MEDIUM/LARGE	224	117	48	29	57	23	91	25	220	104	48	27	57	22	91	24
SMALL/LARGE	288	82	52	25	63	21	95	24	285	105	52	24	63	21	95	23

Table 8.

STRUCTURAL OPTIMIZATION: BARRIER-TEST AND SIDE-IMPACT (MODIFIED STRUCTURE)

********** 30 MPH TEST-VELOCITY *******

COLLISION MODE	HEAD HIC		HEAD ACC		THORAX ACC		PELVIS ACC	
FRONTAL IMPACT VC= 100 KM/H = 62,5 MPH CLOSING-VELOCITY								
LARGE/LARGE	698	698	74	74	42	42	42	42
MEDIUM/LARGE	659	624	70	72	42	40	40	41
SMALL/LARGE	854	367	74	66	43	33	42	39
FRONTAL IMPACT VC= 112 KM/H = 70 MPH CLOSING VELOCITY								
LARGE/LARGE	634	634	67	67	40	40	35	35
MEDIUM/LARGE	1108	767	86	72	45	44	41	36
SMALL/LARGE	1715	671	102	72	58	42	51	35
SIDE IMPACT VA= 50 KM/H = 31,2 MPH IMPACT-VELOCITY								
LARGE/LARGE	40	182	22	32	34	27	54	31
MEDIUM/LARGE	224	117	48	29	57	23	91	25
SMALL/LARGE	288	82	52	25	63	21	95	24

It can thus be established that a design for fixed barrier impact and side-impact already simultaneously provides an adequate description of the result that would be obtained if a vehicle-to-vehicle collision were also considered, if it is ensured that sufficient deformation energy is available. It cannot be reasonably expected from a cost-oriented optimization procedure that limited resources are applied where no improvement in the target function is achieved.

Conclusions

It has been shown that the design of the front-structure of a vehicle has a high influence in side-protection. Changing the force-level of the front-structure in a 50 km/h (31 mph) side-impact from 0.5-times to 2.0-times the force-level of present vehicles is—in terms of dummy load—the same, as if the impact velocity was changed from 20 mph to 35 mph. Thus the partner-protection, the performance of the structure of a striking vehicle from the point of view of a passenger of a struck vehicle, should be part of vehicle testing.

Different test procedures were studied at different test speeds:

1. Fixed barrier impact
2. Frontal impact with a movable deformable barrier.
3. Integration of a fixed barrier impact, an impact with a movable deformable barrier and a side-impact.
4. Integration of a fixed barrier impact and a side-impact.

The side-impact always was a test of self-compatibility, meaning that the vehicle was struck by a vehicle of the same type. The influence of the front-structure was thus measured by this test together with the performance of the side-structure.

The optimum front-structure in terms of the fixed barrier impact shows a good performance in all frontal collisions. The performance in side-impact is worse than for present vehicles.

The optimum front-structure in terms of a test with the movable deformable barrier takes advantage of the specific test velocity and so the overall performance is not as good. No additional information is gained by this test procedure, compared to fixed barrier impact. The structure of the long member is nearly the same as in the case of the fixed barrier impact, but the behavior of the engine now depends on closing velocity. Thus the integration of a fixed barrier impact, an impact with a movable deformable barrier and a side-impact does not provide more information than the integration of a fixed barrier impact and a side-impact. This test configuration (fixed barrier impact and side-impact) helps to ensure partner-protection as well as passenger-protection. This means that with this test configuration measures for the passengers of a tested vehicle are evaluated as well as measures implemented to protect passengers of other, struck vehicles.

The tests with a movable deformable barrier are redundant, compared to fixed barrier impact. But when such tests are conducted, the acceleration of the movable deformable barrier must be measured and evaluated as well, as it constitutes a substitute for the struck vehicle. Without measuring the acceleration of the movable deformable barrier, this test is completely misleading, since the performance of a "bad" vehicle could otherwise be improved by merely screwing a sheet of lead into the vehicle.

The acceleration of the movable deformable barrier could be evaluated by simulation in the following manner: The acceleration is the input of a simulation program describing one or more average vehicles. From the resulting dummy loads, the performance of the tested vehicle in terms of partner-protection is derived. As shown in section 5 and 6, the movable deformable barrier does not provide much additional information.

The test procedures were examined at 30 mph and 35 mph test velocity in the frontal impact. 35 mph increases the conflict of aims between the safety design of frontal impacts and side-impacts. The best procedure would thus be to optimize the front-structure for 30 mph with enough deformation energy for 35 mph. That could be tested in a 35 mph barrier impact without dummies, which only reflects whether there is a sufficiently low intrusion of the steering-wheel and the splash-wall etc.

The higher the design-velocity for frontal impact, the higher are the force-levels of the front-structure. This leads to a conflict of aims with the side-impact. The integration of the fixed barrier impact and a side-impact of a vehicle with a vehicle of the same type provides the opportunity to study and evaluate the vehicles, how they solve this conflict of aims.

The possibilities of computer-aided compliance(6,9,11) should be investigated in order to be able to attain the aims of this effort, without the proposed tests actually being carried out on completed vehicles. If only full-scale testing with finished vehicles is possible, progress can be achieved only when the experience with the completely developed vehicle is available, and thus progress can only influence the development of successor models of the vehicles now under development. Such progress will be available for customers not for approximately ten years. If subsystem- or component-testing is sufficient to attain the aims of this work, it would be possible to influence the structure of vehicles, now under development. Efforts are therefore necessary to improve the reliability of subsystem- and component-testing. This should be supported by the governmental organizations, since an increase in safety at the earliest possible time is in the interests of the public.

This work should not be misunderstood to advocate an increase of the number of tests, but to produce more information and more subtle knowledge by the tests through appropriate test procedures.

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The Safety Problem for Passengers in Frontal Impacts - Analysis of Accident, Laboratory, and Model Simulation Data

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Abstract

Injuries to the right, front seat passenger account for nearly one-quarter of the total harm to passenger car occupants. Approximately half of this harm is associated with frontal impacts.

The objective of the frontal crashworthiness research and development effort at the National Highway Traffic Safety Administration is to assess the safety problem associated with occupants of passenger cars involved in frontal impacts and to identify potential remedies for the problem. The focus of this paper is on passenger protection. Several other reports within the frontal crashworthiness program have focused on driver protection and vehicle structures.

Accident data, laboratory component data, sled and crash test data, and analytical simulation efforts form the basis for the definition of the passenger protection safety problem and the identificaton of alternative countermeasures. This paper presents the progress and results in each of the areas related to this program:

- The NHTSA's accident data files are being utilized to characterize the crash conditions and resulting injuries associated with right, front seat passengers. The head/face body region accounts for approximately one-third of the harm to passengers. The chest and abdominal body regions also account for approximately one-third of the harm. Lower extremities account for 16% of the harm.
- The MVMA 2-D Model is being implemented to simulate the crash environment described by the accident data to better understand the injury mechanisms and to test the effectiveness of alternative countermeasures. A case study is presented and the results indicate that the lower portion of the instrument panel is an important component in managing the forward kinetic energy of an impacting occupant.
- An extensive laboratory effort is underway to characterize the material properties and geometry of the primary interior components associated with injury causation - instrument panels, windshields, headers, and A-pillars. This data is providing input data for the MVMA 2-D model and in identifying countermeasures. Examples of typical force-deflection characteristics are illustrated.

Passenger Safety Assessment

The objective of the frontal crashworthiness research and development effort at the National Highway Traffic Safety Administration (NHTSA) is to assess the safety problems associated with occupants of vehicles involved in frontal impacts and to identify potential remedies for the problem.

This paper focuses on the safety problem of right, front seat passengers. Several other reports within the frontal crashworthiness program have focused on driver protection and vehicle structures(1,2).

A three phase approach is being implemented including:

Phase I. Problem Definition. Review accident data and test results. Identify critical risk parameters and characterize current fleet performance. Establish a basis for postulating and analytically evaluating possible mitigation concepts.

Phase II. Concept Analysis. Characterize baseline vehicles and component performance. Develop information on body region loading as a function of impact conditions and establish injury severity relationships. Postulate reasonable mitigation concepts and analytically evaluate their effect on harm.

Phase III. Concept Development and Evaluation. Expand the experimental characterization of baseline performance. Confirm the predicted performance of promising mitigation concepts through testing. Develop test procedures and test devices.

The material presented in this paper relates to the work that has been completed within the first two phases - Problem Definition and Concept Analysis.

The initial focus of this effort has been directed at defining the safety problems and identification and evaluation of interior compartment countermeasures for the unrestrained passenger. However, the provision of improved interior compartment protection is important to the concept of "built-in" or "softened" car surfaces which provide protection with or without the use of seat belts or air bags. This concept is discussed by Wilson(3) as related to work being performed at General Motors related to the steering system and side impacts. Improved frontal interior compartment protection will be effective for occupants that use seat belts or air bags since 1.) belt restrained occupants in severe frontal crashes can contact frontal interior surfaces, 2.) belt or air bag restrained occupants can contact frontal interior surfaces in side impacts or rollovers, and 3.) occupants in air bag equipped cars involved in crashes below the deployment threshold can contact frontal interior surfaces. The safety problems associated with the restrained passenger will also be analyzed as part of the continuing efforts of this project and careful consideration will be given to the effects of restraint usage on overall injury mechanisms. The development and evaluation of potential countermeasures for passenger protection will be based on the needs of both the restrained and unrestrained occupant and the associated safety problems as they are projected to exist in the future highway environment.

As to the crash configuration, this effort focuses on frontal crashes that do not involve rollover or ejection of the occupant. Figure 1 compares the magnitude of harm for crashes involving rollover or ejection. As can be seen, those crashes where there is no rollover or ejection account for approximately 90% of the harm in frontal impacts.

Phase I - Problem Determination

The following sections present analyses conducted using accident data and crash test data to better define the safety problem for right, front seat passengers. The purpose of these analyses is to establish the basis for postulating and analytically evaluating possible mitigation concepts.

Injuries to the right, front seat passenger account for approximately 25% of total harm to passenger car occupants. Approximately half of this harm is associated with frontal impacts.

The concept of harm is defined by Malliaris, et al (4,5) and is calculated from the schedule of economic costs of highway casualties as presented in "The Economic Cost to Society of Motor Vehicle Accidents"(6). The harm weight factor associated with each AIS level as used in this paper is as follows:

AIS	Harm Weight Factor
6	264.9
5	232.5
4	56.7
3	9.2
2	3.0
1	0.7

The accident data analyses presented in the following sections utilize the concept of harm applied to the

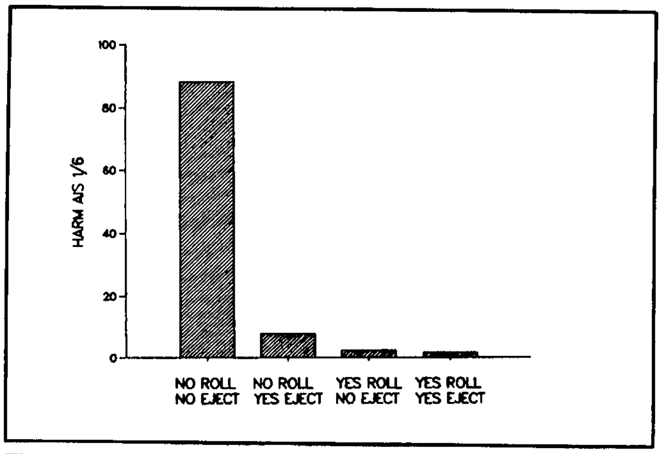

Figure 1. Distribution of ahrm by rollover and ejection

National Accident Sampling System (NASS) files for the years 1979 to 1984.

Distribution of Harm by Body Region and Source of Injury

Table 1 presents information on the distribution of harm to unrestrained passengers by body region injured. Table 1 data is obtained from the NASS files. As noted from the table, the head and face body regions account for approximately one-third of the harm. A large percentage of this harm (especially for the face) is obtained as a result of minor or moderate (AIS 1 and 2) injuries. The chest and abdomen body regions also account for approximately one-third of the total harm. In contrast to the head/face region, the harm associated with these body regions are primarily due to serious (AIS 3-6) injuries. Lower extremities account for 16% and upper extremities account for 8% of the total harm. Neck injuries account for 3% of the harm.

Figure 2 indicates the ranking by harm of body regions for AIS 1 and 2 injuries. The head and face body regions account for approximately 50% of the harm, and the lower extremities account for almost 30% of the harm.

Figure 3 indicates the distribution of harm by body regions for serious injuries. The torso (chest and abdomen) accounts for almost 50% of the harm associated with serious injuries. The head body region accounts for approximately 30%. Figure 3 also illustrates the distribution by body region of the number of serious injuries in addition to harm. The counts of injuries shown are for AIS 3 to 6 injuries and AIS 5 and 6 injuries. For AIS 3-6 injuries, the lower extremity body region is ranked about equal with the chest body region with each accounting for one-quarter of the serious injuries. The head accounts for 16%. For AIS 5 and 6 injuries, the chest and head regions are the most frequent and each accounts for

Table 1. Distribution of harm by body region.

Body Region	Harm % of Total (Column) AIS 1-6	Harm % of Row AIS 1,2	AIS 3-6
Chest	24.1 %	15 %	85 %
Head	24.4 %	24 %	76 %
Lower Extremities	16.4 %	59 %	41 %
Face	13.0 %	80 %	20 %
Abdomen	10.4 %	1 %	99 %
Upper Extremities	8.2 %	57 %	43 %
Neck	3.1 %	36 %	64 %
Other	0.4 %	100 %	0 %

Notes:

Files: 1981-85 NASS

40% of the injuries. The abdomen accounts for 15% and the neck accounts for 5% of the injuries.

Torso Injuries

Chest and abdominal injuries account for approximately one-third of total harm. This harm is almost completely derived from serious AIS 3-6 injuries. When only considering harm associated with serious injuries, the chest and abdominal regions account for approximately half the total harm. (Included in the chest body region is the chest, back, and shoulders.)

The instrument panel is the primary contact source for both serious and minor/moderate injuries to the chest and abdomen.

For the chest, arteries severed, artery lacerations, skeletal fractures, and all systems crushed account for most of the system involvements. Liver lacerations account for the highest percentage of the abdominal system harm.

Head/Face Injuries

The head and face body regions comprise approximately one-third of total harm. Most of the harm to the head is a result of serious injury (AIS 3-6) while the majority of the facial injury is at the AIS 1 and 2 severity level. When considering AIS 3/6 injuries alone, head and face injuries account for about 30% of harm.

For minor/moderate injuries, the windshield is the primary injury source and the instrument panel is ranked second for head and face injuries. Figure 4 indicates the source of harm for serious injuries to the head. For the head, the windshield is ranked first, the A-pillar second, and the instrument panel third as sources of injury.

Brain concussions and skeletal fractures account for the majority of head serious injury harm, and skeletal fractures account for almost all of the facial serious injury harm. At the AIS 1 and 2 injury levels, contusions, abrasions and lacerations to the skin make up the majority of facial harm, and brain concussions account for the majority of head harm.

Lower Extremity Injuries

The lower extremities include the pelvis, thigh, lower leg, knee, and ankle. Of the 16% of total harm accounted for by the lower extremities, about 40% of this is classified as serious injury.

Figure 5 shows the breakdown of the lower extremities region for the harm associated with serious

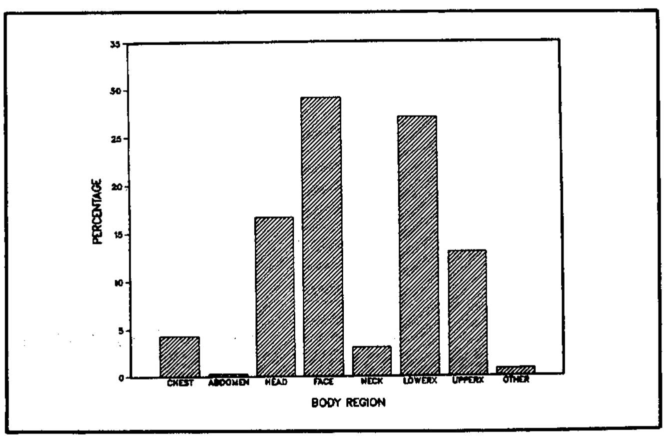

Figure 2. Distribution of harm for AIS 1 and 2 injuries

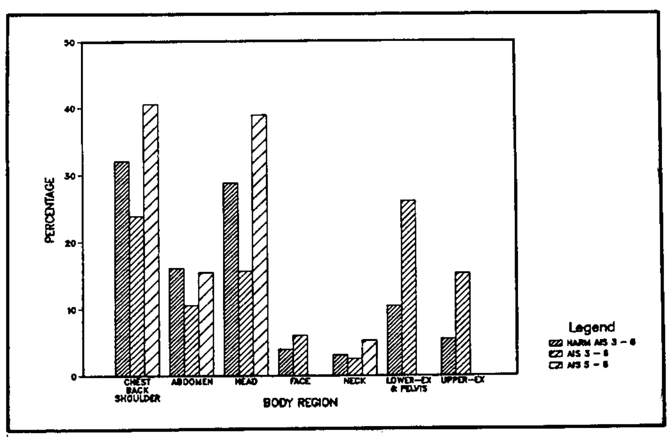

Figure 3. Distribution of harm for AIS 3-6 injuries

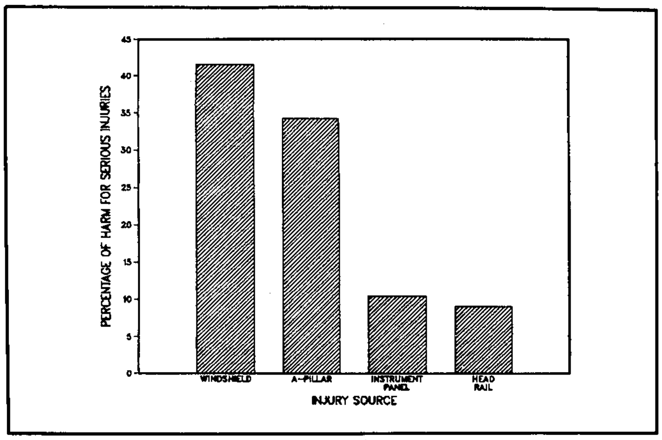

Figure 4. Distribution of harm by source of serious head injury

injuries. The thigh and ankle/foot regions account for most of the harm. The instrument panel is almost always the source of serious injury for the various lower extremities except for the ankle/foot. For the ankle/foot region, the side interior and floor are the major sources of injury.

Skeletal fractures and joint fractures and dislocations account for the majority of serious injury harm. Contusions, abrasions and lacerations of the skin, and fractures account for most of the AIS 1 and 2 injury harm.

Neck Injuries

The neck accounts for approximately 4% of total harm. Of this, about 65% is classified as serious. Severance of the spinal cord comprises 75% of the serious injury harm.

The windshield and non-contact sources are the primary cause of harm associated with AIS 1 and 2 injuries, while the windshield is the primary cause of harm for serious injuries.

Crash Severity

The distribution of overall harm by delta v is shown in Figure 6. Figure 7 shows the distribution of harm by delta v for individual body regions. Because of the small sample size when the data is categorized in this manner, the body regions were aggregated into three regions - head/face, chest/abdomen, and lower extremities. Fifty percent of the harm for the head/face region and the lower extremity region occurs below 20 mph. The 50th percentile delta v for chest/abdomen harm occurs at about 35 mph.

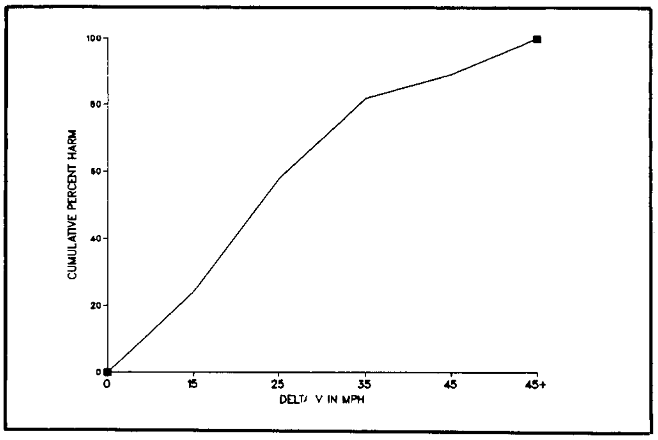

Figure 6. Distribution of harm by delta v

Crash Configuration

This section presents the crash configurations associated with total harm. Crash configuration includes items such as object struck, direction of force, and general area of damage.

Figure 8 presents the principle direction of force. The twelve o'clock position accounts for approximately 60% of the harm. Figure 9 presents the distribution of harm by body region and direction of force. As can be seen, there is little difference between body regions, and the twelve o'clock direction accounts for the majority of harm.

The specific area of impact along the front of the vehicle is indicated in Figure 10. Distributed damage accounts for 42% of the harm, damage to the right side of the vehicle accounts for 35% of the harm, and

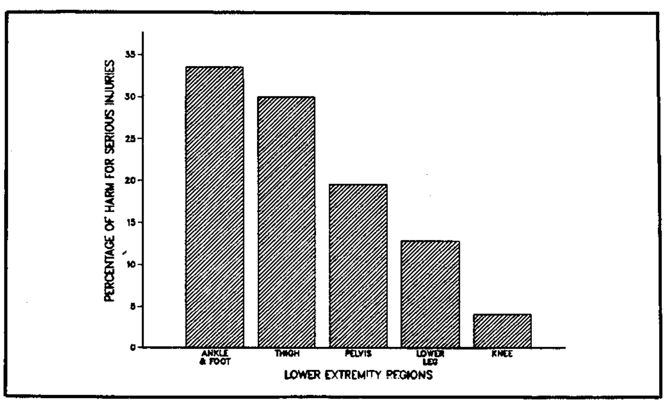

Figure 5. Distribution of harm by specific body regions for serious lower extremity injury

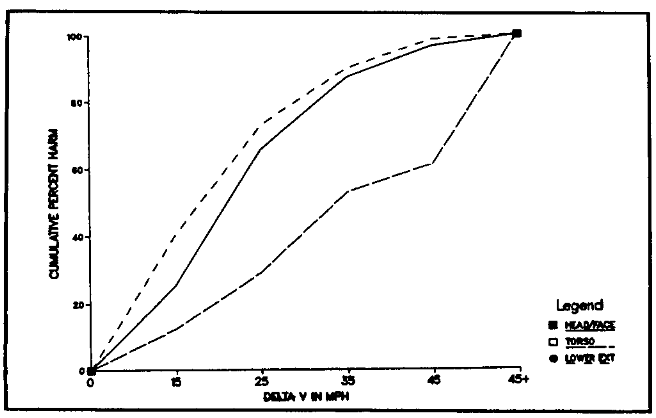

Figure 7. Distribution of harm by delta v and body region

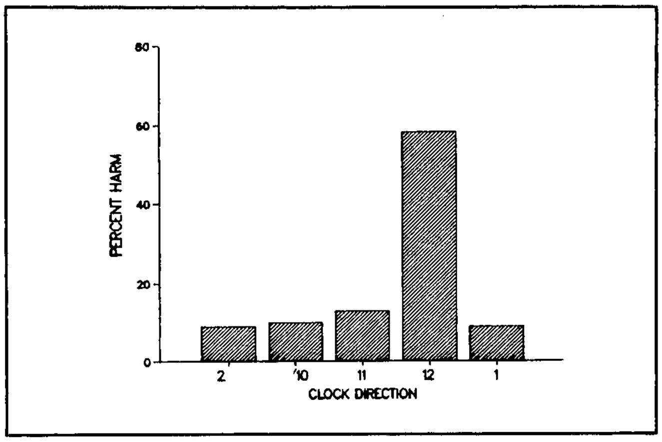

Figure 8. Distribution of harm by direction of force

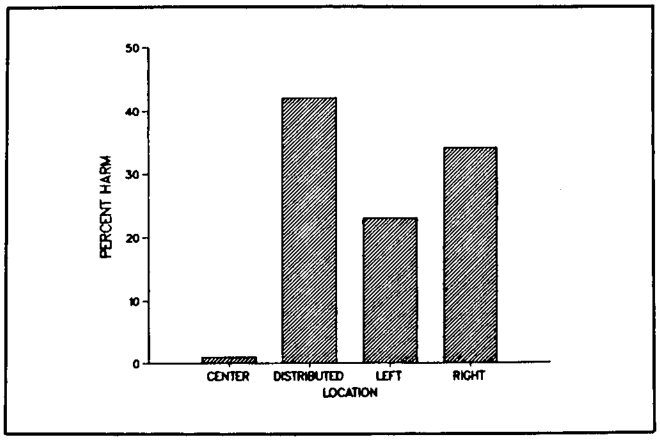

Figure 10. Distribution of harm by impact area

damage to the left side accounts for 23%. Figure 11 shows the specific area of impact of body regions. Head and face injuries occur mostly from offset impacts. In contrast, the chest receives most harm from distributed impacts. In the case of abdominal injury, harm is equally divided between offset and distributed.

Figure 12 indicates the object contacted. Vehicle to vehicle crashes make up 60 to 70% of the harm. The balance is equally distributed between trees and poles and other objects. As seen in Figure 13, this distribution is fairly similar across body regions injured.

Crash and Sled Test Results - Occupant Motion and Impact Response

As part of the initial work in the frontal crashworthiness project, a series of crash and sled tests were performed with unrestrained and restrained drivers and front seat passengers. The purpose of these tests were to provide baseline data for the vehicle structures, driver, and passenger protection programs. This baseline data has been utilized to better understand the injury mechanisms involved and to provide data for validating occupant and structures analytical modeling efforts.

The crash test series consisted of car-to-barrier tests, car-to-car tests in both full frontal and offset configurations, and car-to-pole tests. Both Part 572 and Hybrid III dummies were utilized in restrained and unrestrained conditions. The barrier equivalent velocities for these tests were approximately 30 mph.

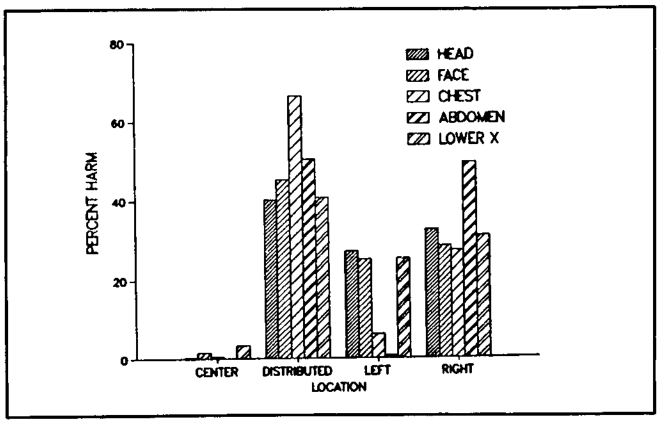

Figure 11. Distribution of harm by impact area and body region

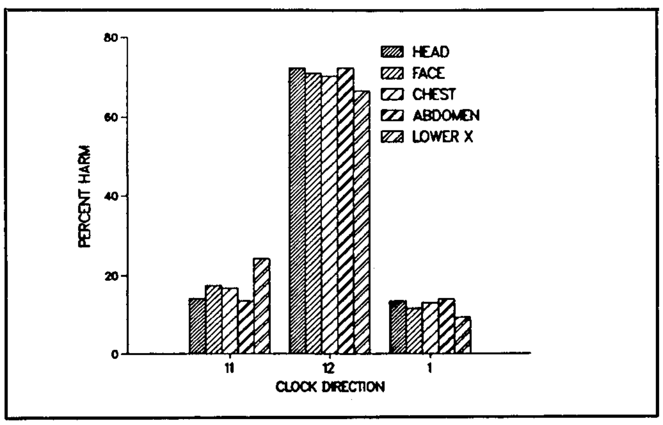

Figure 9. Distribution of harm by direction of force and body region

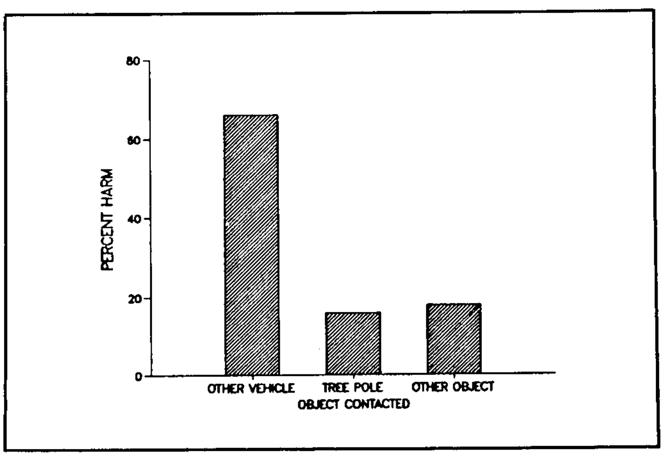

Figure 12. Distribution of harm by object contacted

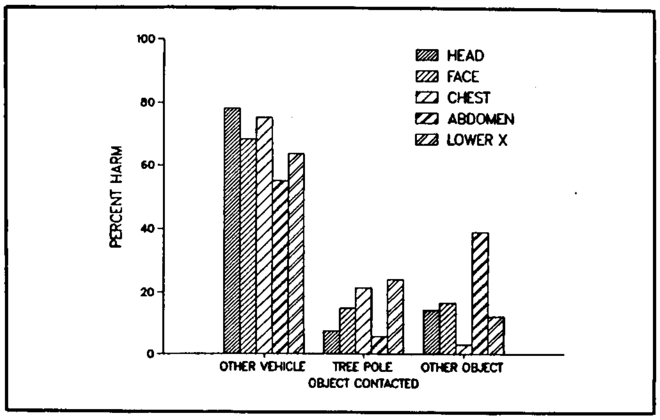

Figure 13. Distribution of harm by object contacted and body region

Table 2 indicates the passenger dummy response in terms of the FMVSS 208 injury criteria for head, chest, and femur. The only tests shown in the table are those which included an unrestrained passenger dummy. The high-speed films of these tests were reviewed to observe the passenger dummy kinematics. In general, the dummy moved forward in the seat until the knees made contact with the lower instrument panel. As the knees penetrated the lower instrument panel the forehead contacted the upper part of the windshield. The head rotated backward towards the spine. The head begins to penetrate through the inter plastic layer of the windshield and then plows down the windshield toward the upper intrument panel. As the knees continued to push into the lower instrument panel and head plowed down the windshield, the chest and shoulders of the dummy would contact the mid-instrument panel.

The manner in which the knees contacted and pushed into the instrument panel had an important effect on the dummy kinematics. In the Accord car-to-car tests, the knees slid under the mid-instrument panel directly into the lower instrument panel. The dummy remained fairly vertical until head contact. The head contacted the windshield, fractured the glass, tore a hole completely through the windshield (glass and plastic inter layer), and plowed down the windshield. The head did not appear to hit the upper instrument panel. In contrast, in the Accord car-to-pole test the knee contacted the instrument panel at the juncture of the middle and lower instrument panel. While the crash configuration in this case was different, it is felt that the initial knee contact was significant in affecting subsequent contacts. As the knees pushed into the panel, the entire panel was distorted and pushed upward. The occupant started rotating forward after contact. The head in this case fractured the windsield after contact but did not penetrate the plastic layer. As the head slid down the windshield it hit the top of the instrument panel. Another example of how the knee contact can affect the occupant kinematics was observed in the Fuego tests. In the Fuego, the knees contacted the panel at the intersection of the middle and lower portions of the panel. The knees then slid vertically up the plane of the mid-instrument panel while pushing into the

Table 2. Unrestrained passenger dummy response summary.

Seq Num	DOT Num	Subject Vehicle	Barrier Equivalent Velocity	Test Type	Partner Vehicle	Dummy Type	HIC	Chest G (3 msec)	Left Femur	Right Femur
70	804	Fuego	29.9	FF	Accord	Hyb III	1472	67	-2339	-2168
50	877	Omni	30.0	FF	Omni	Part 572	2256**	107**	-1055	-1207
44	865	Accord	29.9	LO	Fuego	Hyb III	553	50	-893	-410
40	812	Accord	29.9	FF	Celebrity	Hyb III	440	85	-2404	-1804
71	806	Omni	30.0	FF	Celebrity	Hyb III	1645	59	-1757	-862
34	785	Accord	28.4	FF	Accord	Part 572	733	90	-1877	-1717
46	846	Omni	29.8	CP	LC Pole	Hyb III	914	54	-1956	-1053
68	796	Fuego	30.0	FF	Fuego	Hyb III	1362	57	-2225	-2719
36	795	Omni	30.0	FF	Omni	Part 572	*	102**	-1497	-1846
80	865	Fuego	28.8	LO	Accord	Hyb III	662	50	-1559	-532
37	804	Accord	30.1	FF	Fuego	Hyb III	995**	90	-2025	-1507
35	796	Fuego	30.0	FF	Fuego	Part 572	1052**	93**	-2621	-2422
30	773	Celebrity	30.0	FB	FRB	Part 572	770 **	64	-1393	-798
81	877	Omni	30.0	FF	Omni	Hyb III	1432**	82	-1073	-1041
31	776	Celebrity	29.7	FB	LCB	Hyb III	655	72	-163	*
39	810	Omni	20.0	FF	Celebrity	Hyb III	549	38	-565	-644
67	785	Accord	31.6	FF	Accord	Hyb III	959**	77	-1634	-1681
45	819	Accord	30.0	CP	LC Pole	Hyb III	1176	65	-1011	-538
33	783	Fiero	29.8	FB	LCB	Hyb III	623	68	-1523	-2311
42	816	Omni	28.5	FF	Concord	Hyb III	*	69	-1370	-1577
41	815	Accord	30.0	FF	Concord	Hyb III	923	91	-2743	-2291
43	824	Fuego	30.0	FF	Celebrity	Hyb III	1828**	60	-2723	-1919
69	795	Omni	30.0	FF	Omni	Hyb III	1498**	60	-1109	-1983
47	847	Fuego	29.9	CP	LC Pole	Hyb III	1279	73	-1768	-1924
32	779	Fiero	29.7	FB	LCB	Part 572	546	70	-1049	-1067

* NO DATA
** QUESTIONABLE DATA

Table 3. Comparison of passenger sled test data.

Test	HIC	Head Acc.	Chest Acc.	Chest Defl.	Femur Left	Femur Right
1980 Ford Mustang						
1	410	139	66	0.6	-508	-1942
2	476	145	54	0.6	-701	- 951
3 •	766	115	106	1.6	-870	-1955
4 ••	688	83	103	1.7	-982	-1537
5 ••	560	77	91	1.9	-773	-1557
6 ••	654	109	93	1.7	-863	-1521
1983 Chevrolet Celebrity						
7	739	96	68	1.6	-1659	- 971
8	590	90	67	1.3	-1806	-1142
9	612	75	70	1.5	-1970	-1227
10	685	101	59	1.5	-1534	-1311

• Ethafoam 900 in glove box area
•• Ethafoam 900 in glove box area and Ethafoam 600 in instrument panel

panel. The chest impacted both the legs and the mid-instrument panel. The head rotated severely backward after contacting the windshield and slid down the windshield until it contacted the upper instrument panel. The occupant slid off the seat and pocketed under the instrument panel.

It should be pointed out that in most cases the dummy's head did not penetrate through the windshield as in the Accord car-to-car tests. In most cases, the windshield was fractured and the head plowed down the glazing. Also, as to head injury criteria, the highest HIC values appear to have resulted when contact consisted of a combination of both windshield and upper instrument panel impacts.

In addition to the crash tests, a series of Hyge sled tests were conducted to simulate a frontal impact of a 1980 Ford Mustang and a 1983 Chevrolet Celebrity (7,8). Hybrid III dummies were utilized in unrestrained conditions. The simulation was for a barrier equivalent velocity of 30 mph. Table 3 presents the results of these tests for the passenger dummy. HIC values ranged from approximately 400 to 800, and chest accelerations ranged from 60 to 100. For some of these tests, foam was added under the instrument panel to simulate various parts. The occupant kinematics in both vehicles are fairly similar. The knees made initial contact with the lower instrument panel. The head contacted the windshield and rotated backward. The head slid down the windshield and contacted the upper part of the instrument panel. During this time, the chest contacted the mid-instrument panel. In the Celebrity test, the knees first impacted at the juncture of the middle and lower instrument panels. This resulted in the entire instrument panel being displaced upward as the knees pushed into the panel.

MCR Technology, Inc. conducted several sled tests at 20 mph barrier equivalent impact velocity as compared to the 30 mph tests discussed in the previous paragraphs(9). The tests utilized a 1980 Chevrolet Citation sled buck. The tests were conducted using different size dummies in unrestrained conditions. Based on the results of this test series, they concluded that the major risks for the 50th percentile unrestrained occupant at 20 mph are:

1. High angular accelerations of the head, resulting from windshield impact.
2. Possible injury to the larynx area, resulting from impact with the upper corner of the instrument panel.
3. Possible neck injuries resulting from windshield impact.
4. Possible injuries to facial bones from windshield.
5. Head and facial lacerations from windshields.

Possibly because of the lower impact speeds of these tests, the chest interaction with the middle instrument panel did not appear to be as critical as in the previous tests that were conducted at higher speeds. This agrees with the findings of the accident data discussed in an earlier section in which the vehicle delta v's associated with chest injuries are higher than for head and lower extremity injuries.

The test series also included several tests with different size occupants. For the 5th percentile occupant, they indicated that all of the above were a problem with the addition of high head accelerations and/or possible injuries to facial bones resulting from impact with the upper corner of the instrument panel. For the 95th percentile occupant, they indicated additional problems caused from contact with the windshield header.

As part of this study, MCR Technology, Inc. analyzed compartment modifications in an attempt to mitigate the problems noted. The analysis was performed using both the MVMA 2-D analytical model and hardware modifications on a sled buck. Promising results were obtained in testing modifications including making the windshield softer (tested analytically only), extending the instrument panel, and adding padding to the header area. The modification of the instrument panel was designed to induce earlier chest deceleration so as to reduce the impact speed of the head/windshield contact. The modification was also aimed at reducing the amount of whole body bending to reduce the risk of spinal injuries.

An analytical study of occupant kinematics conducted previously by NHTSA investigated the motions of restrained and unrestrained occupants in frontal collisions(10). For the unrestrained right, front seat passenger analyses, different size occupants, different impact speeds, and several types of typical crash pulses were analyzed. Charts and graphs are presented that indicate body segment impact speed by segment displacement for different vehicle impact speeds. It was reported in this analysis that at the 15 mph crash speed no thorax contact with the instrument panel was produced. It is stated that the interaction begins to occur somewhat above 15 mph but certainly below 25 mph. This result was for the 50th percentile occupant.

Passenger Compartment Intrusion

As noted in the previous sections, the timing of contacts with the interior components can have an important effect on occupant response. Intrusion of the instrument panel or the toeboard during a crash can have a critical effect on this timing and it was therefore investigated. The NHTSA's National Crash Severity Study accident data collection effort collected data on specific component intrusion. The extent of intrusion for instrument panels and toeboards are shown in Figures 14 and 15. As can be noted, the probability of toeboard intrusion is much higher than for the instrument panel. It should be noted that these intrusion measurements are the static or residual intrusion. The dynamic intrusion during the crash event could be greater especially for instrument panels. Figure 16 indicates the probability of receiving a serious injury by delta v for different levels of intrusion. The cases with intrusion have a higher probability than those vehicles without intrusion. To increase the sample size, the cases in this analysis included both drivers and passengers and all frontal components including the steering assembly, instrument panel, toeboard, etc.

As part of the crash test series referenced above, frontal static intrusion was determined for several of the vehicles(11). Each of the vehicles selected for the static intrusion analysis had a series of pre and post-test measurements taken. Intrusion measurements were obtained for the underlying instrument panel

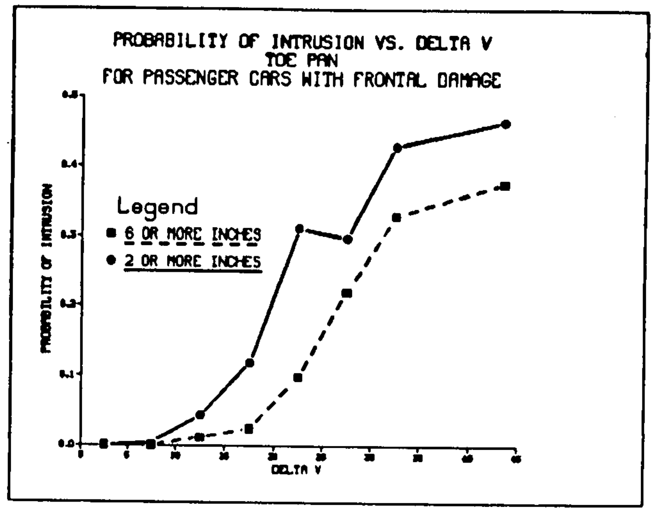

Figure 15. Extent of intrusion for toe pan in frontal impacts

structure rather than from the panel itself. Measurements were taken after removing the instrument panel at five elevations across the entire front of the occupant compartment. Table 4 summarizes the peak static intrusion measurements for the right, passenger side of the vehicle. Intrusion measurements are given for the five elevations. From the table, it can be seen that the maximum intrusion levels occurred below the cowl on the firewall and toeboard. The off-set crashes resulted in fairly high levels of intrusion considering the fact that these off-sets were run on the left side of the vehicle. The crash test matrix did not include any right side off-sets which presumably would have resulted in greater amounts of intrusion to the passenger side. (The primary purpose of this test series was to observe steering assembly intrusion effects and camera coverage focused on the driver side of the

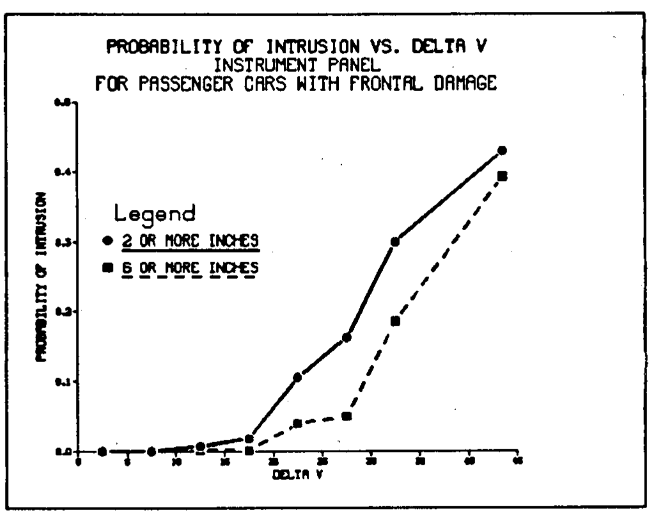

Figure 14. Extent of intrusion for instrument panel in frontal impacts

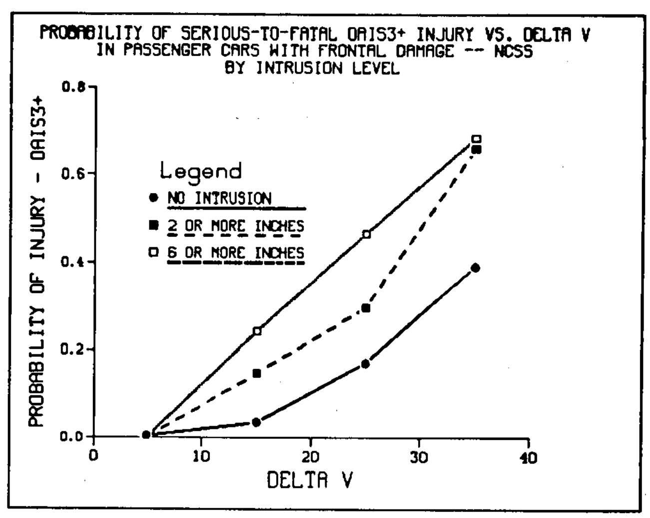

Figure 16. Probability of receiving a serious injury for different levels of intrusion (any frontal component intrusion)

Table 4. Static frontal intrusion measurements.

Vehicle	Test Configuration		Peak Static Measurement at Each Level* (inches) A	B	C	D	E
Celebrity	Celebrity/Omni	Lt Off-set	1.8	2.4	3.5	3.5	2.5
Omni	Omni/Omni	Lt Off-set	5.4	6.8	6.9	8.4	7.3
Honda	Honda/Fuego	Lt Off-set	7.9	9.8	10.6	8.6	8.5
Honda	Honda/Honda	Lt Off-set	4.3	6.1	6.9	6.1	6.4
Fuego	Fuego/Fuego	Lt Off-set	1.8	2.6	3.1	2.4	1.9
Honda	Honda/Honda	Full Front	.8	2.2	3.1	4.6	4.6
Fuego	Fuego/Barrier	Full Front	4.8	3.4	4.2	4.3	4.4
Omni	Omni/Pole	Lt Off-set	3.2	5.7	8.5	9.3	7.3
Honda	Honda/Pole	Center	2.0	2.8	7.6	8.4	10.0
Honda	Honda/Pole	Lt Off-set	4.1	6.3	7.8	8.3	8.5
Fuego	Fuego/Pole	Center	2.2	3.1	4.7	5.7	7.8
Fuego	Fuego/Pole	Lt Off-set	1.9	2.8	3.3	4.8	4.7

* Notes:

1. Intrusion measurements indicated above are for the right side of the occupant compartment.

2. Levels A&B – Cowl level
C&D – Firewall
E – Firewall/Toeboard Juncture

compartment. Also, to allow for more detailed investigation, dummies were only placed on the driver side. Thus, it is not possible to relate dummy readings with the magnitude and location of intrusion.)

Instrument panel intrusion has also been analyzed analytically using the MVMA 2-D computer model (12). The study examined the effect of interior surface intrusion on occupant response in frontal collision. Both restrained and unrestrained occupants were simulated. The study concluded that for the crash conditions simulated, intrusion increased the restrained occupant's head and chest accelerations while no clear trend emerged for the unrestrained occupant.

In addition to the displacement of the instrument panel because of exterior crushing, the instrument panel can be displaced because of the knees pushing into the panel. This type of displacement was observed in reviewing the films for both the crash and sled tests as discussed in the previous section. This internal displacement can also effect occupant responses and it should be considered in evaluating mitigation concepts.

Instrument Panel Fracture

In several of the crash and sled tests the instrument panel fascia fractured and broke apart. A number of hard copy reports of investigated accidents contained in the NASS files were reviewed to determine if this condition existed in the field. A large percentage of the cases reviewed showed that the panel had fractured and broken apart. Figures 17 through 20 illustrate a sample of the fractured instrument panels. This type of failure could increase the risk of injury because of the laceration potential, exposure of rigid components under the instrument panel, and the loss of integrity. The loss of integrity could decrease the ability of the panel to absorb impact energy. While the cases that were examined were not selected in a strictly random fashion and are not necessarily representative of all accidents, it does indicate another characteristic that should be considered in postulating countermeasures.

Instrument Panel Materials Testing

Work on testing instrument panel material characteristics for energy absorption, loading, failure, and sensitivity to temperature and impact speed has been reported along with other desirable characteristics (13,14,15.) Reference 13 indicates that impact resistance is a key property because of the role of the instrument panel in occupant safety during collisions and, also, because the instrument panel is easily damaged during the assembly process. It is indicated in this reference that the ductile failure mode as opposed to a brittle mode failure is desirable for energy management and avoiding lacerations to occupants during a crash. Reference 14 presents two concepts for desirable performance characteristics related to the loading and failure point of instrument panel retainer material. One is to maximize the failure energy of the material, and the second is to maximize the rectangularity of the force versus deflection curve. Another goal indicated in this reference is to identify materials that are insensitive to impact conditions including impact speed and temperature.

The following sections of this report present laboratory results which were conducted as part of this project related to characterizing the performance of instrument panels.

Phase II - Analytical Characterization of the Accident Environment and Mitigation Concept Development and Analysis

As indicated previously, Phase II of the project involves: 1.) the characterization of baseline vehicles and component performance and 2.) the postulation and analytical evaluation of mitigation concepts. This phase of the project has included both the laboratory collection of mechanical properties for interior components and the analytical characterization of the accident environment. This characterization is being used to better understand the injury mechanisms as discussed in the previous sections, to identify alternative mitigation concepts, and to evaluate the effectiveness of these concepts in reducing harm. It is stressed that the evaluation of mitigation concepts in this phase of the work in through analytical methods to allow for the investigation of a broad array of alternatives, it is not until Phase III that the more extensive hardware testing and evaluation will be performed on the most promising concepts.

CASE NO. 83-80-076S　　　80 AMC EAGLE DL

Occupant Injuries

Body Region	Injury Source	AIS
Thigh	Instrument Panel	3
Thigh	Instrument Panel	3
Head	Unknown	2
Knee	Instrument Panel	2
Knee	Instrument Panel	2
Lower-leg	Instrument Panel	2

Delta-v 20

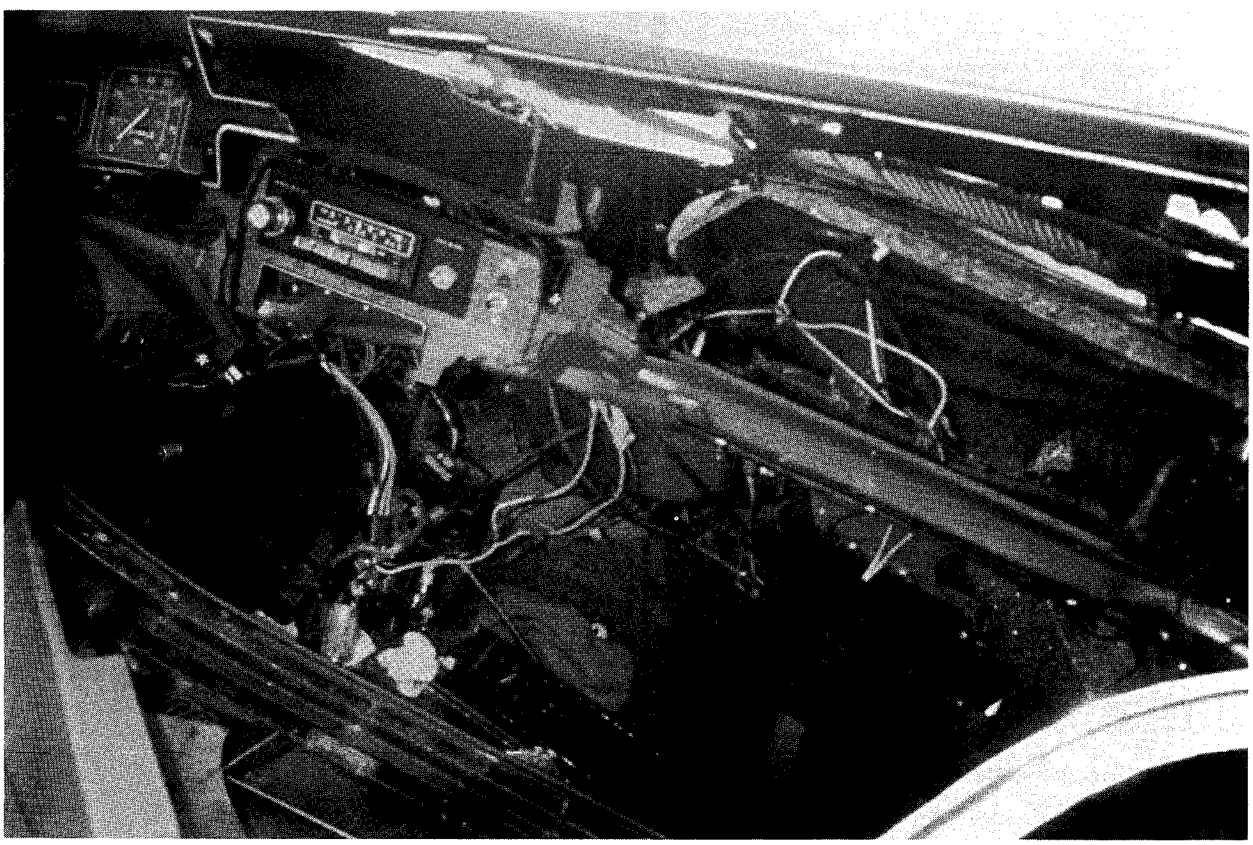

Figure 17. Example of fractured instrument panel

Interior Component Performance Data Collection

Interior component data was necessary to obtain a better understanding of component impact performance. Based on the problem definition work, interior component performance data was collected for the following body region loadings:

1. Knee to Lower Instrument Panel.
2. Chest/Abdomen to Middle Instrument Panel.
3. Head to Upper Instrument Panel.
4. Head to Windshield.

MGA Research Corporation, under contract with the Transportation Systems Center, has developed test equipment and test procedures for acquiring this type of information. Component impact tests are conducted by securing a body part form - head, chest, or knee—to a ram and then propelling the ram into the instrument panel or windshield at a specified velocity and angle. Dynamic and static force-deflection data for instrument panels and windshields have been collected for a wide range of vehicles. Friction and damping characteristics associated with instrument panels and windshields are also being developed.

Figure 21 illustrates the areas of the instrument panel being loaded to obtain force-deflection data. The sequence of the tests is based on the passenger simulations discussed in the problem definition section. The sequence of the tests start with loading the

CASE NO. 81-51-090K 81 CHEVROLET CITATION

Occupant Injuries

Body Region	Injury Source	AIS
Face	Windshield	1
Face	Windshield	1
Face	Windshield	1
Face	Windshield	1
Lower leg	Instrument Panel	1
Lower leg	Instrument Panel	1

Delta-v 47

Figure 18. Example of fractured instrument panel

lower instrument panel with a knee form, the mid-instrument panel is then loaded with a chest form while the knee form is kept in place, and then the upper instrument panel is loaded with a head form. In most cases it was not possible to obtain data for the head striking the upper instrument panel since the panel was badly damaged from the previous tests. The windshield tests were conducted using both Part 572 and Hybrid III head forms. Figures 22 to 24 illustrate several of the test devices utilized.

The impact velocity of the tests were also based on the initial simulation efforts, and the velocities utilized were 15 mph for the knee to instrument panel loading and 20 mph for the other loadings - chest and head to instrument panel and head to windshield. Parametric

CASE NO. 84-55-005T
76 CHEVROLET MONTE CARLO
Occupant injuries

Body Region	Injury Source	AIS
Thigh	Instrument Panel	3
Pelvis	Instrument Panel	2
Head	Windshield	1
Pelvis	Instrument Panel	1

Delta-v 27

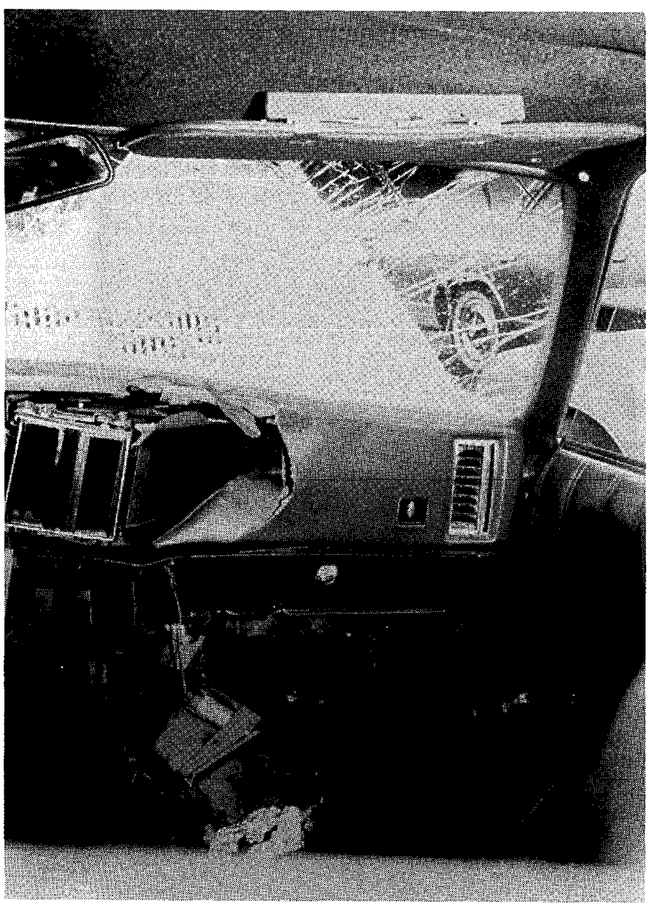

Figure 19. Example of fractured instrument panel

CASE NO. 80-52-171L
Occupant Injuries 76 TOYOTA COROLLA DELUXE

Body Region	Injury Source	AIS
Abdomen	Instrument Panel	4
Face	Windshield	1
Chest	Instrument Panel	1
Knee	Instrument Panel	1
Neck	Unknown	1
Head	Windshield	.

Delta-v Unknown

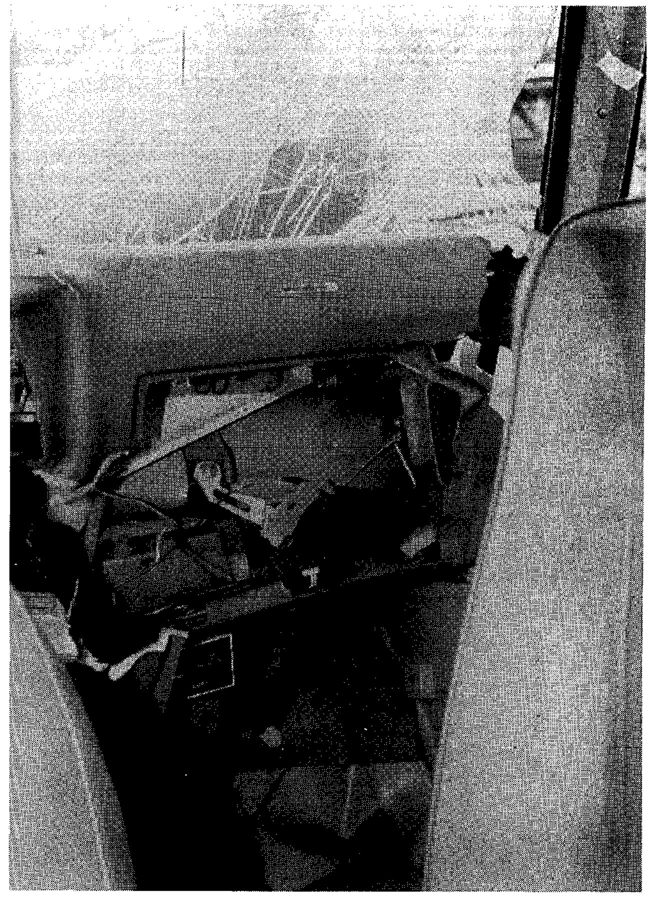

Figure 20. Example of fractured instrument panel

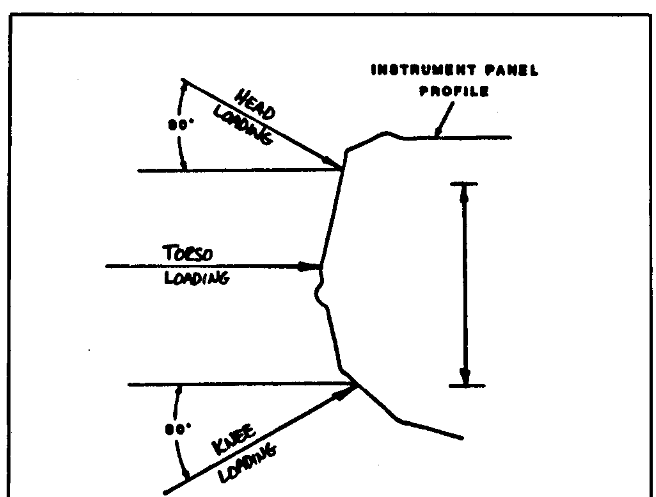

Figure 21. Instrument panel loading points

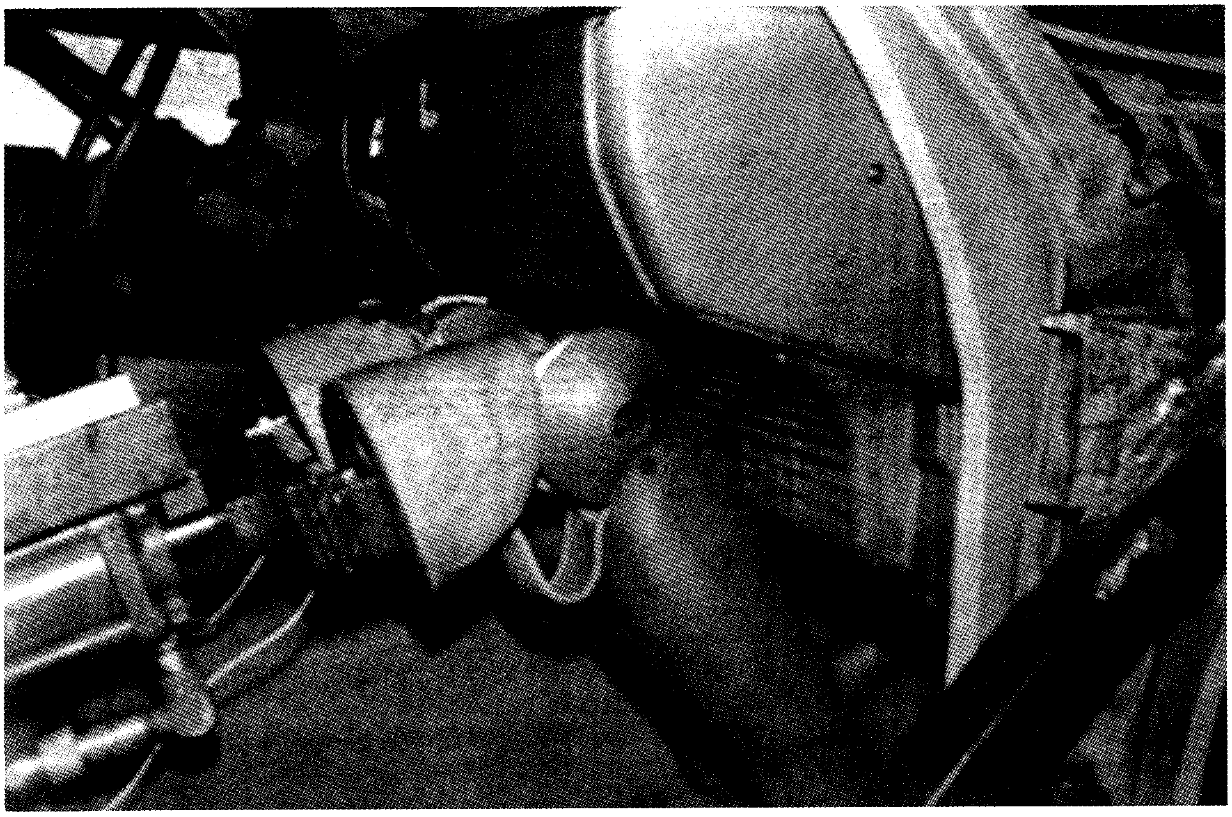

Figure 22. Instrument panel/femur static test set-up

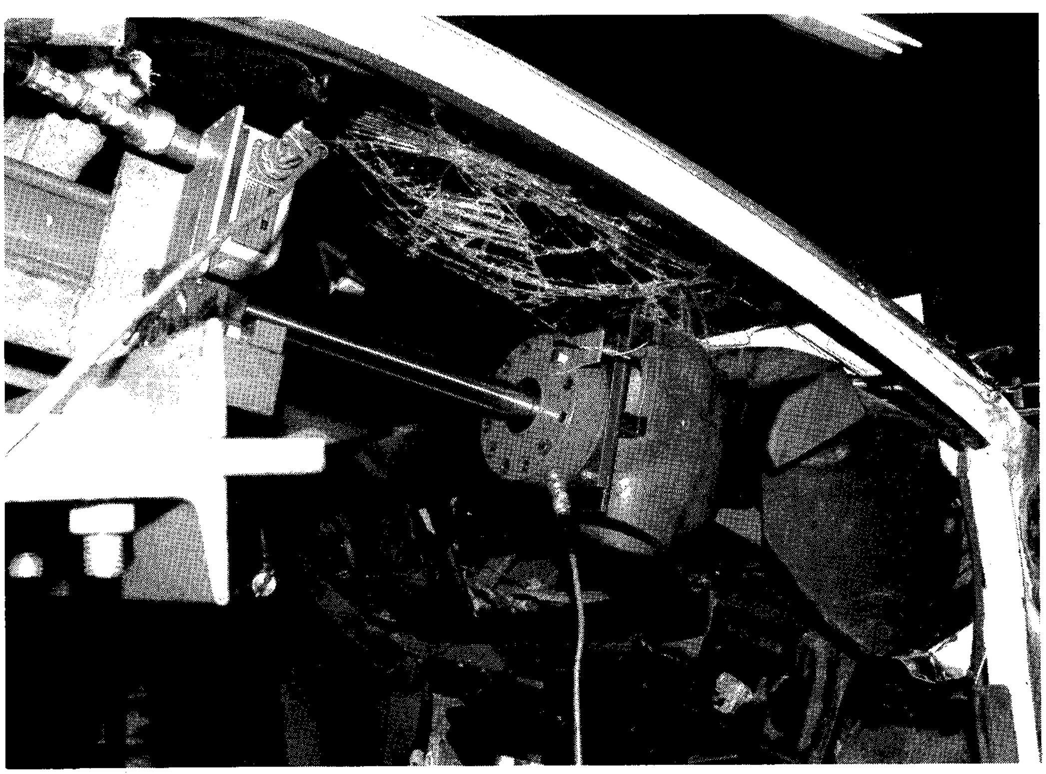

Figure 23. Instrument panel/head dynamic test set-up

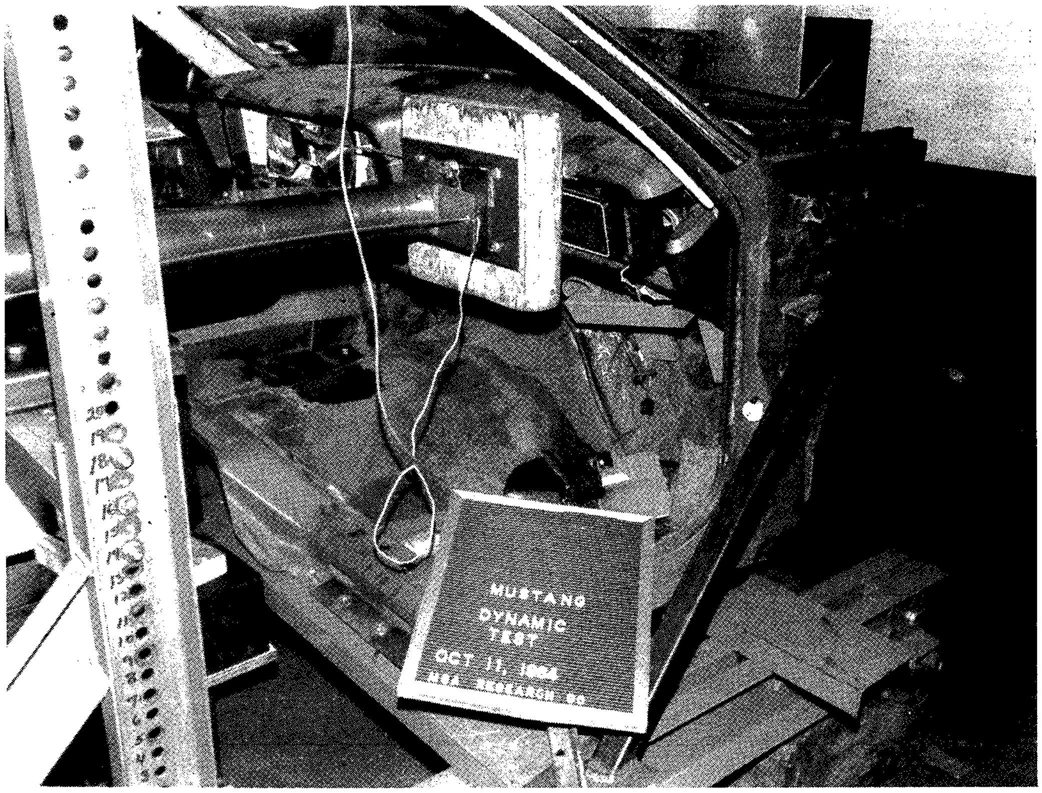

Figure 24. Instrument panel/chest dynamic test set-up

tests for a limited number of vehicles were conducted to determine velocity sensitivity characteristics.

The development of the test equipment and procedures, discussion of the tests and results, and force-deflection data are presented in several references(16,17).

Instrument Panel and Windshield Force-Deflection Characteristics

Typical dynamic force-deflection properties derived from the instrument and windshield component tests are shown in Figures 25 to 27.

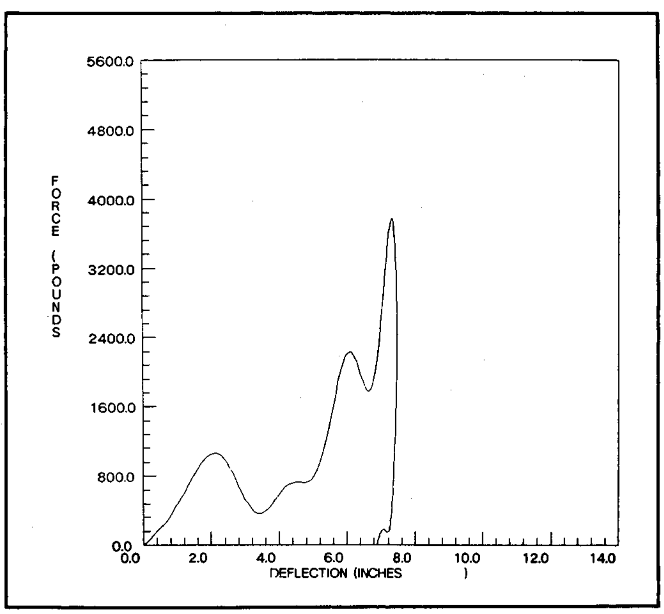

Figure 25. Example of dynamic chest to instrument panel force-deflection

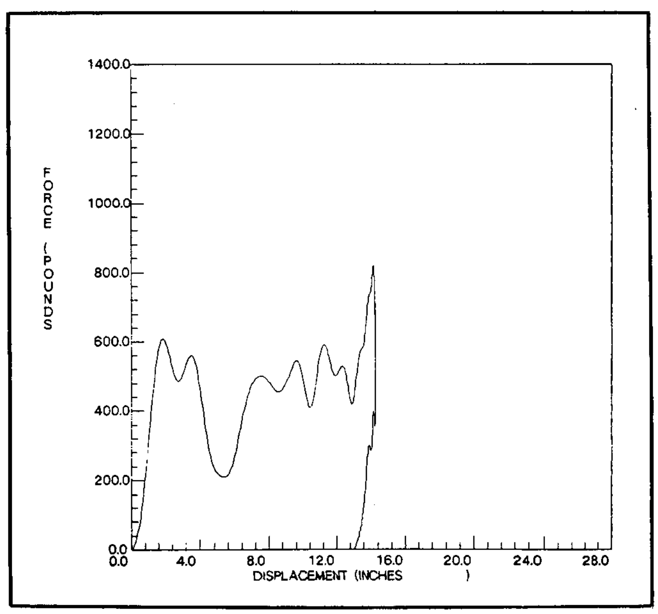

Figure 26. Example of dynamic femur to instrument panel force-deflection

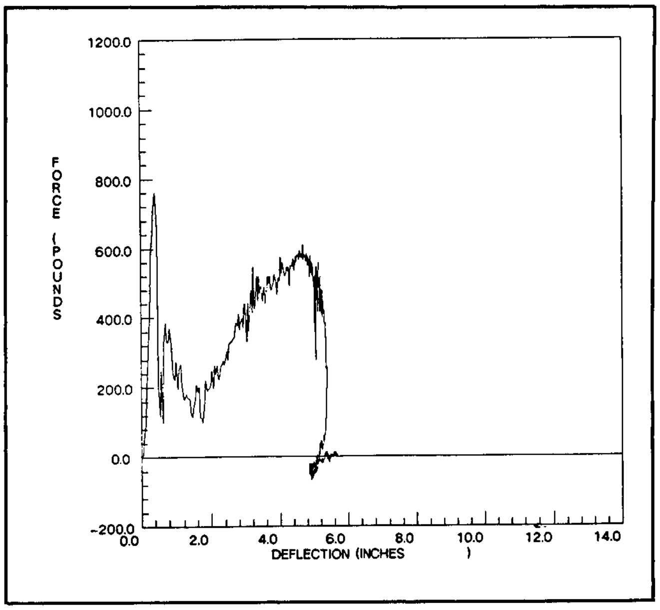

Figure 27. Example of dynamic head to windshield force-deflection

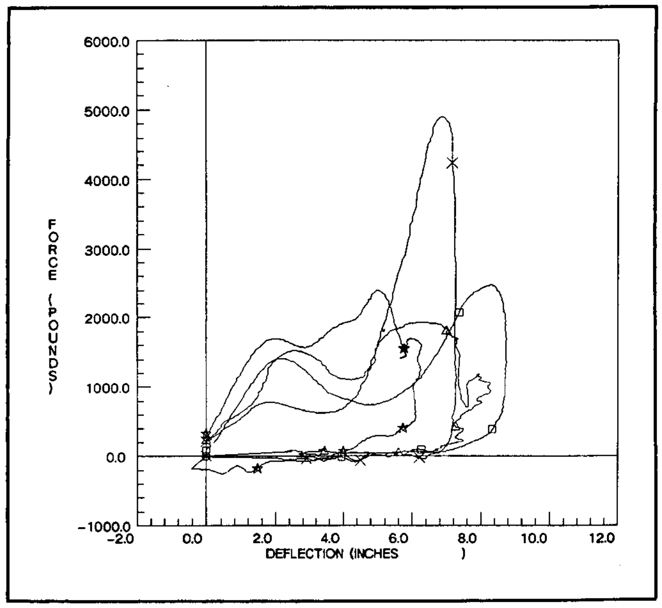

Figure 28. Example of four chest to instrument panel force-deflection characteristics

Data on instrument panel force-deflection characteristics were collected on a wide variety of make/model vehicles covering the period 1965 to 1984. Wide differences on how these instrument panels managed the impact energy were observed. It is these differences that may provide a better understanding of the injury mechanisms discussed previously and to help identify possible countermeasures.

Figure 28 indicates the differences in how four instrument panels managed the energy delivered to them in the 20 mph impact test. These tests are for the chest to mid-instrument panel loading. Similar differences are seen for the knee to lower-instrument panel loading. Differences in characteristics include the initial stiffness of the material, the peak force, the deflection at the peak force, and the total deflection. Table 5 indicates the range of these and other characteristics observed in the passenger cars tested.

Table 5. Impact performance for middle and lower instrument panels. performance characteristics for 31 tested vehicles.

	Mean	Std. Dev.
Middle Instrument Panel (20 MPH Impact)		
Stiffness-First 2" (lb/in)	561	278
Stiffness-First 4" (lb/in)	282	102
Peak Force (lb)	2294	761
Deflection at Pk. Force (in)	6.5	3.3
Peak Deflection (in)	7.9	2.3
Lower Instrument Panel (15 MPH Impact)		
Stiffness-First 2" (lb/in)	314	117
Stiffness-First 4" (lb/in)	173	72
Peak Force (lb)	1351	759
Deflection at Pk. Force (in)	7.3	4.4
Peak Deflection (in)	8.9	3.4

Again, these tests were conducted at impact speeds of 20 mph for the chest and 15 mph for the knee contacts. Figures 29 to 34 indicate how several of these characteristics have varied over time. In general, instrument panels have become softer over time within the first four inches of deflection. (See Figures 29 and 30). Despite this the peak forces have generally risen for more recent model year vehicles (See Figures 31 and 32). This difference is explained by examining the deflection at the point of peak force. Figures 33 and 34 indicate the deflection of the instrument panel at the point of peak force. The older vehicles with high initial stiffness, reached the maximum peak force within a couple of inches of deflection. For the newer vehicles with the softer initial stiffness, the peak force was reached close to the point of maximum deflection from bottoming out on the firewall or other hard spots.

It should be stressed again that these characteristics were obtained from specific impact velocities. The velocity sensitivity of instrument panels was examined in this project, and Figure 35 presents the results of one instrument panel tested at four different impact speeds. As can be seen, the stiffness and the penetrating forces through the panel vary by impact speed. Because of this velocity sensitivity, countermeasures must be "optimized" at the speed ranges accounting for the majority of harm.

The velocity sensitivity of windshields was also examined. Figure 36 presents the force-deflection characteristics at two different impact speeds. The initial spike appears very sensitive to impact velocity and the penetrating forces are generally higher with increasing velocity.

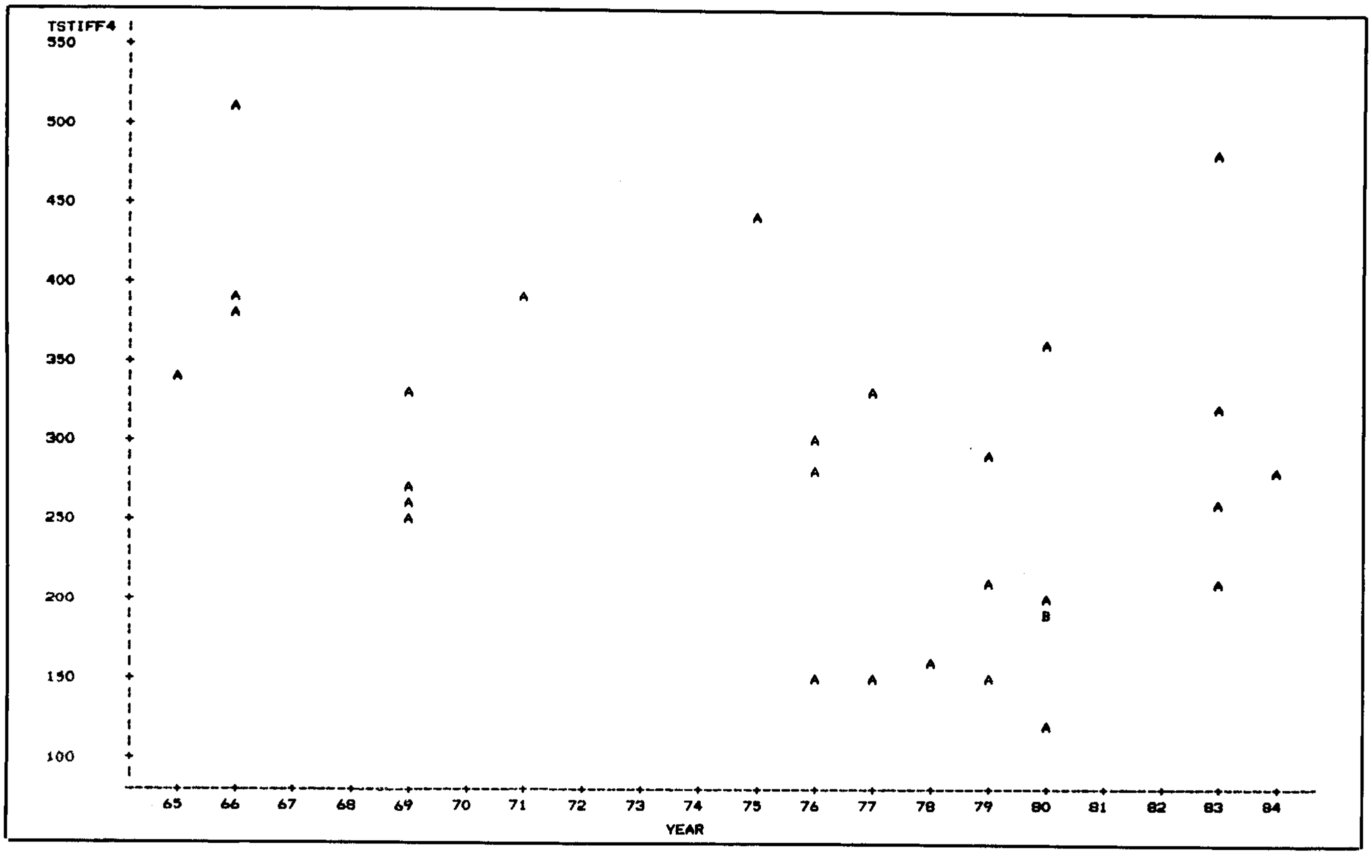

Figure 29. Chest/instrument panel stiffness during first four inches

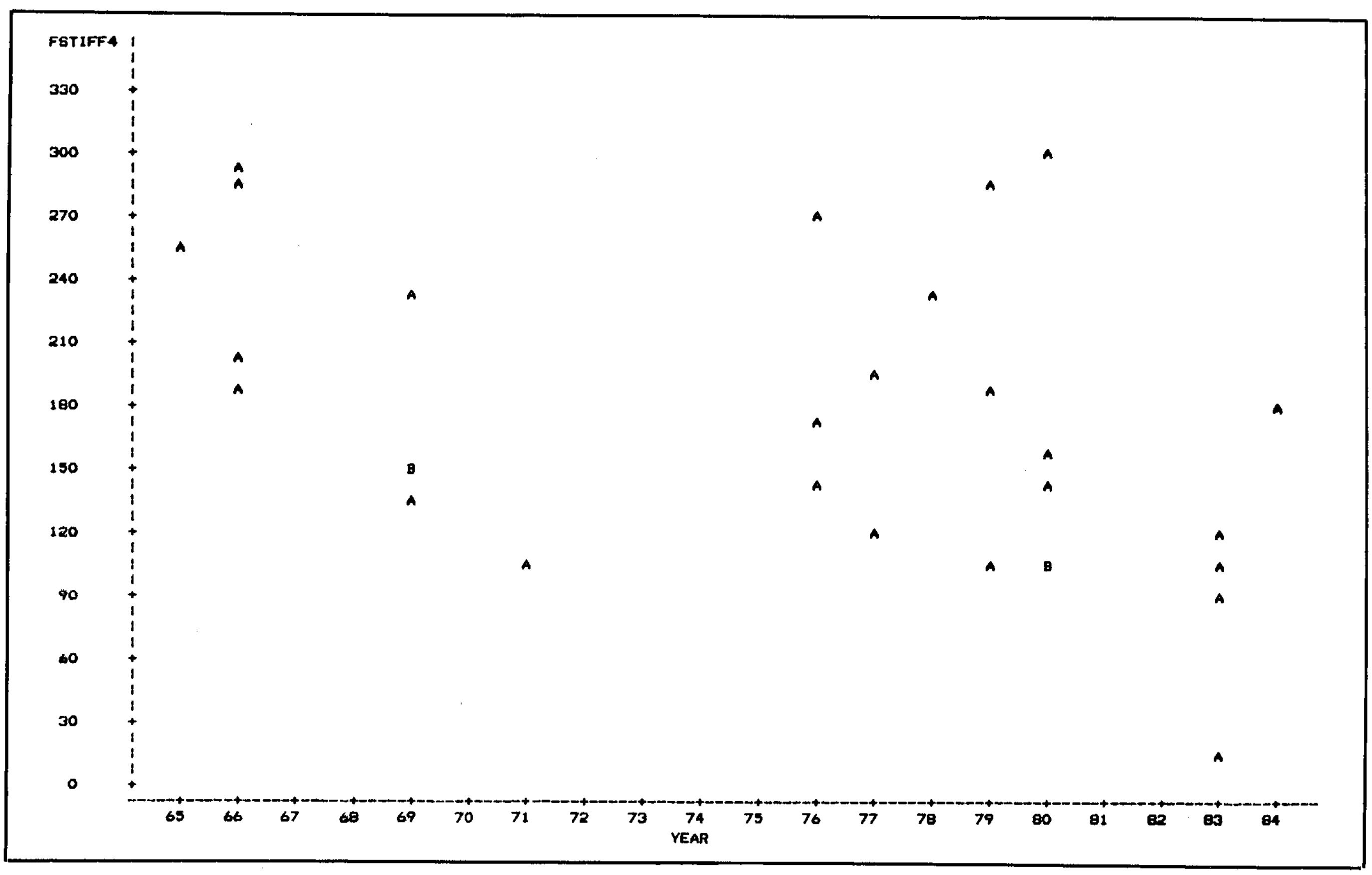

Figure 30. Femur/instrument panel stiffness during first four inches

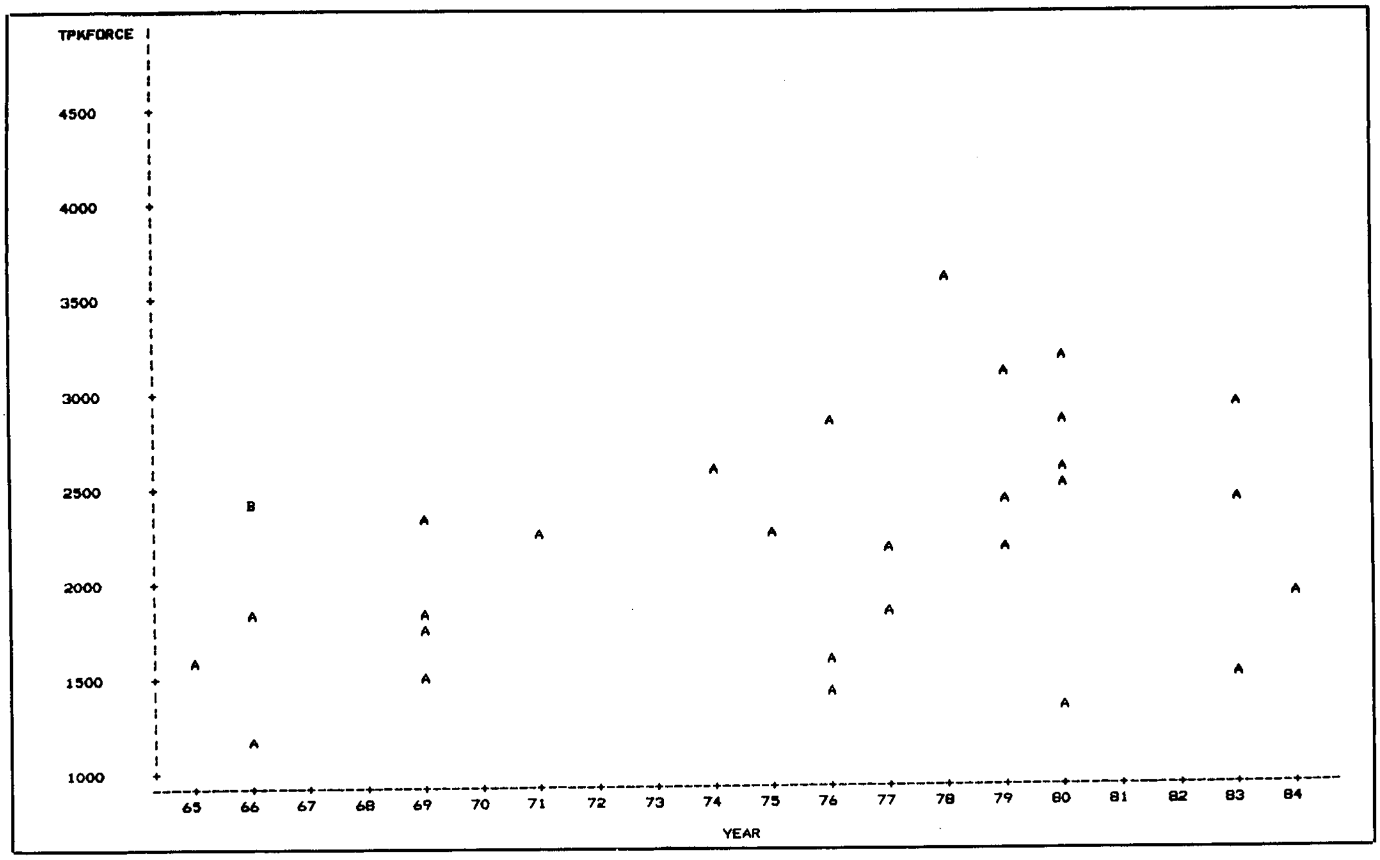

Figure 31. Chest/instrument panel peak force

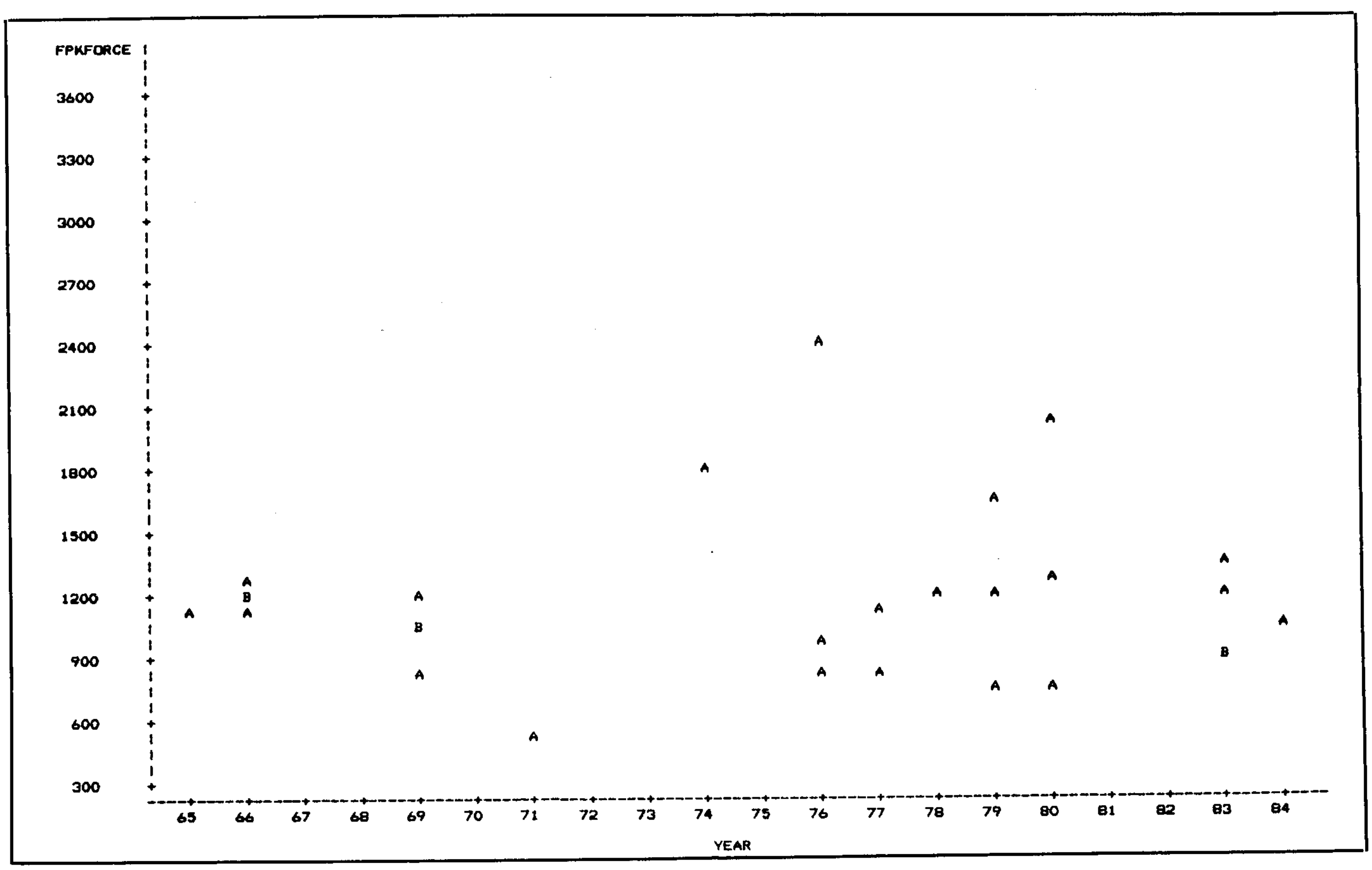

Figure 32. Femur/instrument panel peak force

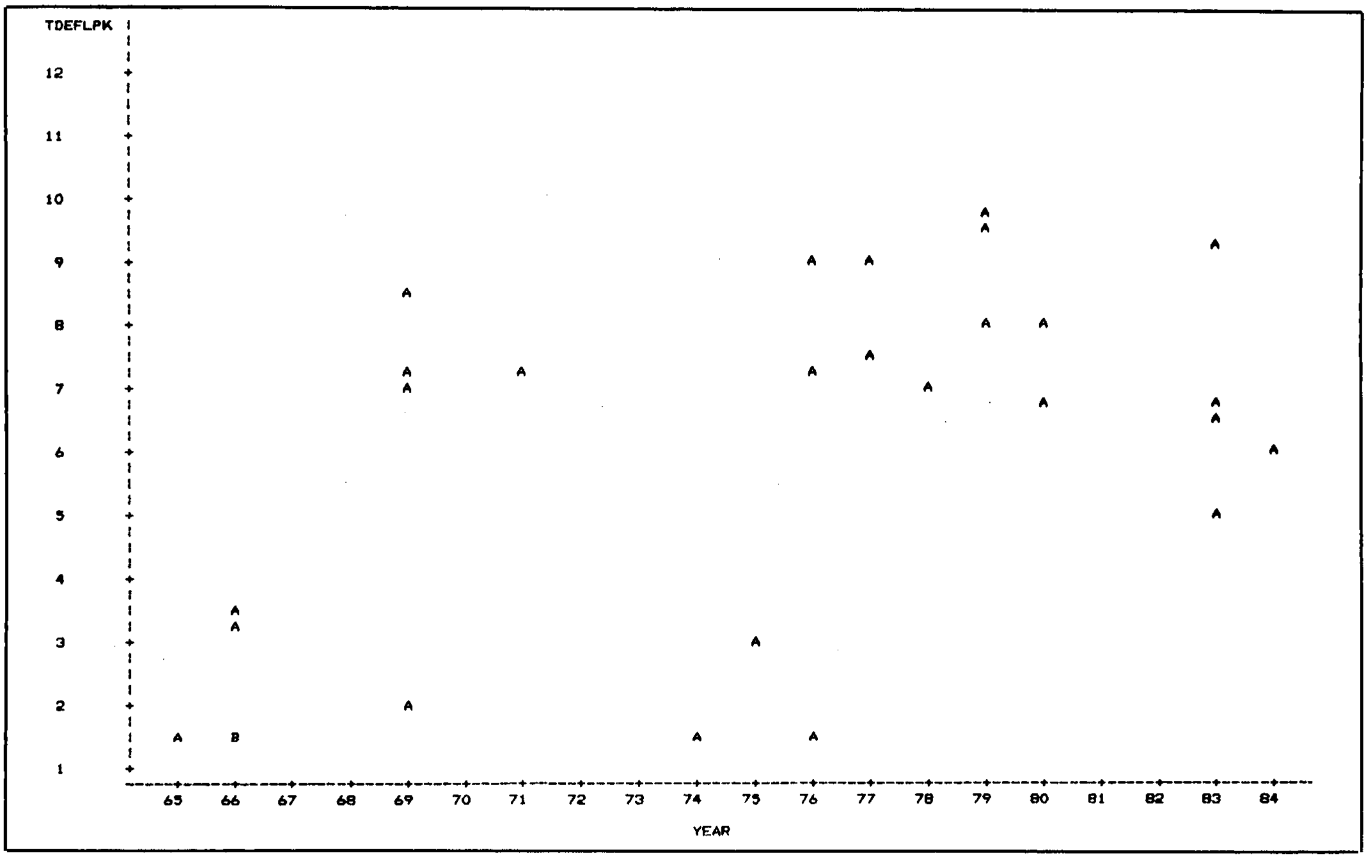

Figure 33. Chest/instrument panel deflecton at peak force

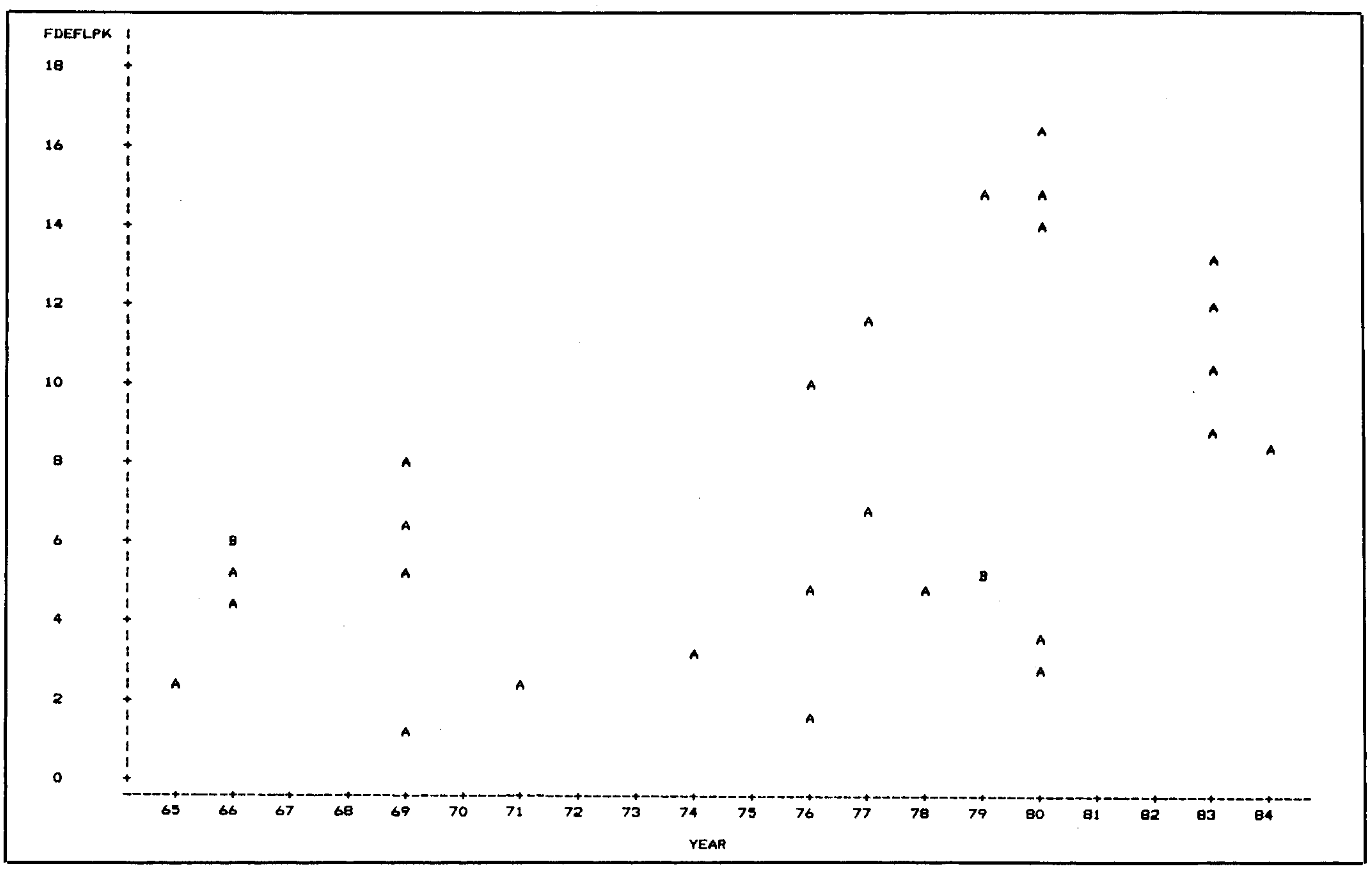

Figure 34. Femur/instrument panel deflection at peak force

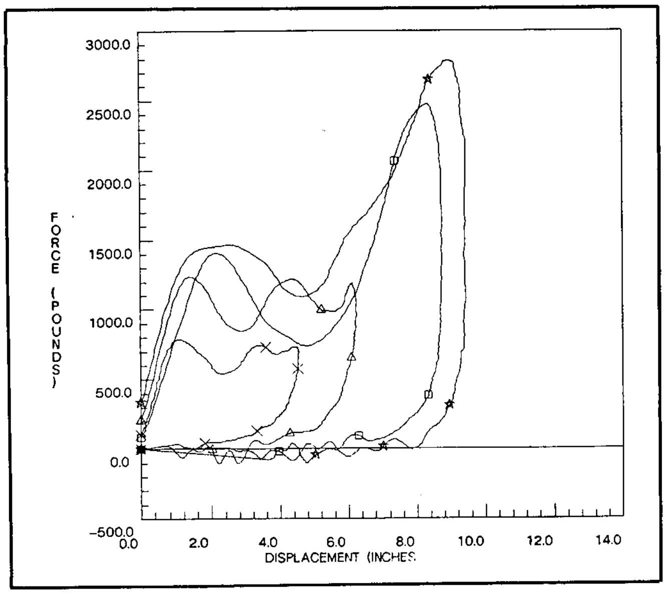

Figure 35. Chest/instrument panel velocity sensitivity (10,15,20,25 mph)

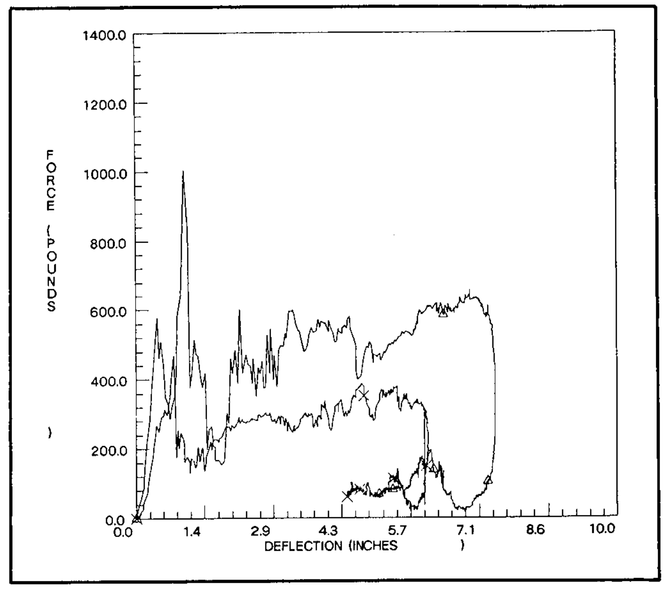

Figure 36. Head/windshield velocity sensitivity (15 and 21 mph)

Analytical Characterization of Baseline Vehicles—Analytical Model

The MVMA 2D Crash Victim Simulator is being utilized in this effort to help characterize the baseline condition of an unrestrained front seat passenger impacting the vehicle interior surfaces. The MVMA 2D model is a two-dimensional crash victim simulator used to predict occupant kinematics and response in a crash environment. The computer model is a large and complex program. It features a nine-mass, nine-segment representation of an occupant with 14-degrees of freedom; an extensible multi-joint neck and flexible shoulder joint; modeling of contacts between the victim and the vehicle by means of contact sensing ellipses attached to the body and a real-line representation of the vehicle interior; specification of materials properties of the vehicle and occupant in terms of general force-deflection relationships; and tabular three-degree of freedom vehicle accelerations. The model's many options and features provide the automotive safety engineer considerable flexibility in defining a crash event. This flexibility, however, imposes considerable demands on the specification of the input data.

Input data

The initial task of the modeling effort was to assemble the required input data. Input data was varied and was acquired from a variety of sources:

Occupant Data

Two basic sets of occupant input data was acquired for this effort. An MVMA representation of the Hybrid III anthropomorphic dummy obtained from MCR Technology Inc. and a humanoid representation obtained from the University of Michigan Transportation Research Institute. Both sets of data were differentiated into three basic anthropomorphic sizes: the 95th and 50th percentile male and the 5th percentile female.

The Hybrid III representation was used to verify the model performance against the crash tests in which an unrestrained Hybrid III passenger dummy was used and the humanoid representation was used in the simulations to predict injuries for comparison with NASS accident data and for the computation of potential harm.

Vehicle Interior Data

Vehicle interior data was divided into two sets. One set consisted of a real-line representation of the passenger compartment (geometric data) and the other set consisted of the force-deflection characteristics of the vehicle surfaces.

Geometric data was obtained from passenger car specification forms published by MVMA, from SAE J1100 measurements obtained from the manufacturers, and from an in-house vehicle interior measurement effort at DOT/TSC. These sources were combined to provide a real-line representation of the interiors of the subject vehicles. The passenger compartment was represented in the model by 11 line segments: windshield and windshield-header, top, middle, and lower instrument panels, seat back and seat cushion, roof, firewall, toeboard, and floor.

The force-deflection data was obtained from the laboratory vehicle component characterization effort carried out by the MGA Research Corporation. As

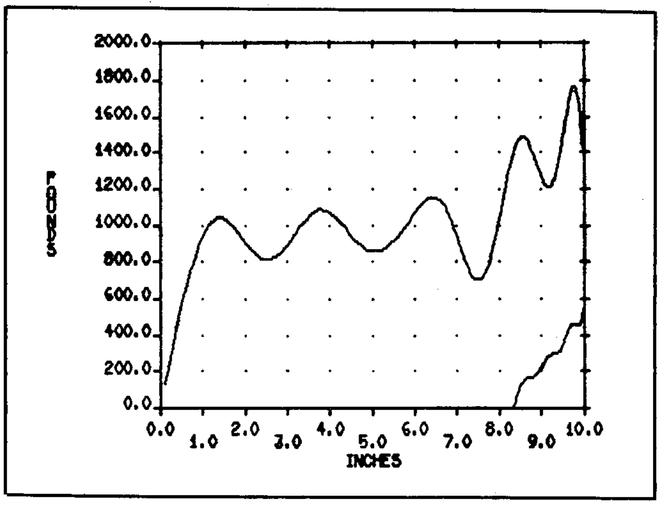

Figure 37. MGA experimental celebrity middle instrument panel force-deflection data

discussed above, this effort consisted of obtaining force-deflection characteristics of the windshield and instrument panels for a wide variety of passenger cars. Both static and dynamic force-deflection data were obtained. A typical vehicle force-deflection data set consists of windshield dynamic data, passenger-side lower and middle instrument panel static and dynamic data, and driver's side static lower instrument panel data. These data are installed on the NHTSA VAX 11/780 computer in Washington D.C. (VERN).

Force-deflection characteristics of the remaining vehicle surfaces (seat, roof, floor, etc) were obtained from data published by MCR Technology. The force-deflection characteristics of these surfaces were considered generic, and applied across all vehicle models.

Vehicle Crash Pulse Data

The vehicle deceleration time-history during the crash was obtained from data provided by NHTSA sponsored crash tests. These data, which are installed on the NHTSA VAX 11/780 in Washington D.C., were screened for data quality and for its applicability to characterizing passenger compartment decelation. The chosen accelerometer data was then converted to MVMA format. The formatted crash pulse was matched with the corresponding vehicle interior file when constructing the MVMA 2D input deck.

Modeling Approach

Selection of the initial baseline cases was based on the availability of NHTSA crash tests of unrestrained passengers. The approach of the modeling effort was to reconstruct tests in which unrestrained passenger data was available. This approach has the dual purpose of establishing the veracity of the model input data as well as creating verifiable baseline cases for the study of passenger compartment injury mitigation scenarios. Test data from the NHTSA crash test data base includes head accelerations, chest accelerations, chest deflection, femur forces, and dummy neck response. The MVMA 2D crash victim model can generate the same output data, so close comparison of the model output with the actual crash event can be made. In addition, high speed films of the crash events were viewed and analyzed. The occupant kinematics observed in the test films is compared to the stick figure output of the MVMA model to further insure correspondence between the actual crash event and the model's simulation.

There is currently available on the NHTSA VAX11/780, unrestrained passenger data on six vehicles:

1983 Chevrolet Celebrity
1983 Ford Mustang
1983 Renault Fuego
1984 Honda Accord
1983 Dodge Omni
1984 Pontiac Fiero

Of these, two, the Celebrity and the Mustang, are Hyge sled test crash simulations and the rest are vehicle tests. As discussed previously, the vehicle test matrix contains full frontal tests (car to car), half offset tests, and narrow object crash tests.

Work is continuing on constructing baseline MVMA 2D cases for these vehicles. Appendix A contains a typical baseline case (1983 Celebrity) which is verified against a Hyge sled test simulation of a 30 mph full frontal crash.

Model Case Study

The objective of this case study was to establish procedures for applying injury mitigation techniques using the MVMA 2D model and to test the effectiveness of the model in showing differences in injury response. This study was to investigate the effects of changes in instrument panel force-deflection characteristics on the body kinematics and injury response of an unrestrained automobile passenger in a frontal collision.

In this pilot study, the effects of changes in instrument panel stiffness was considered. The approach of this exercise was to examine the effect of the rate of energy absorption of the instrument panel surfaces on the resultant kinematics and injury response of the passenger/occupant. It was noted that the experimental force-deflection curves of the middle and lower surfaces of the Celebrity instrument panel (passenger side) differed in the rate at which energy (force x deflection) was absorbed by the panel during the MGA impact tests (Figures 37 and 38). In the lower portion of the instrument panel, resistance to penetration was relatively low early in the event, and gradually built-up to a peak as penetration continued. The middle part of the instrument panel, on the other hand, showed a more rapid build-up of resistance to penetration and a leveling off of penetration resistance until material build-up at deeper penetrations caused an increasing resistance.

A panel which absorbs impact energy similar to the lower panel can be characterized as "soft", i.e. a gradual build-up of resistance as the body form penetrates. The middle portion of the Celebrity instrument panel can be characterized as a "medium" stiff panel, i.e., a panel with a relatively constant rate of energy absorption. Similarly, a third type of panel can be conceived that can be characterized as "stiff"; i.e., a panel which exhibits a high rate of energy absorption early in the event with a tailing off of resistance as penetration continues. Thus three types of instrument panels can be envisioned: a "soft" panel which gradually builds up penetration resistance, a "medium" panel which has essentially a constant rate of energy absorbtion, and a "stiff" panel which can be characterized by a high rate of energy absorbtion early in the impact event.

This approach to the study of unrestrained passenger impacts is essentially an exercise in energy management. That is, a study of the rates at which the kinetic energy of the impacting occupant is absorbed by the surfaces of the vehicle interior. In this study, windshield properties were held constant and only the stiffness properties of the middle and lower portions of the passenger side instrument panel were varied. Impact interactions were considered for the knee into the lower instrument panel and the chest into the middle portion of the instrument panel.

"Soft", "medium", and "stiff" instrument panel surfaces were modeled from the experimental data provided by MGA for both the lower and middle portions of the Celebrity instrument panel. Force-deflection properties of the modeled surfaces were assigned so that the total energy and maximum deflection of the panel surfaces remained within the constraints of the experimental data. Figures 39 and

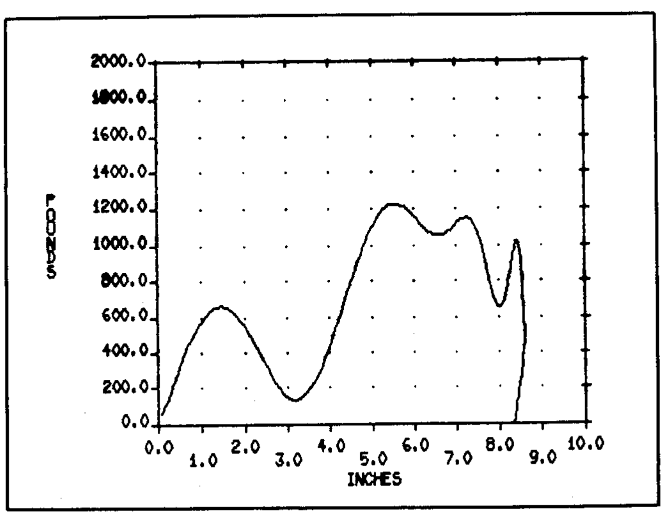

Figure 38. MGA experimental celebrity lower instrument panel force-deflection data

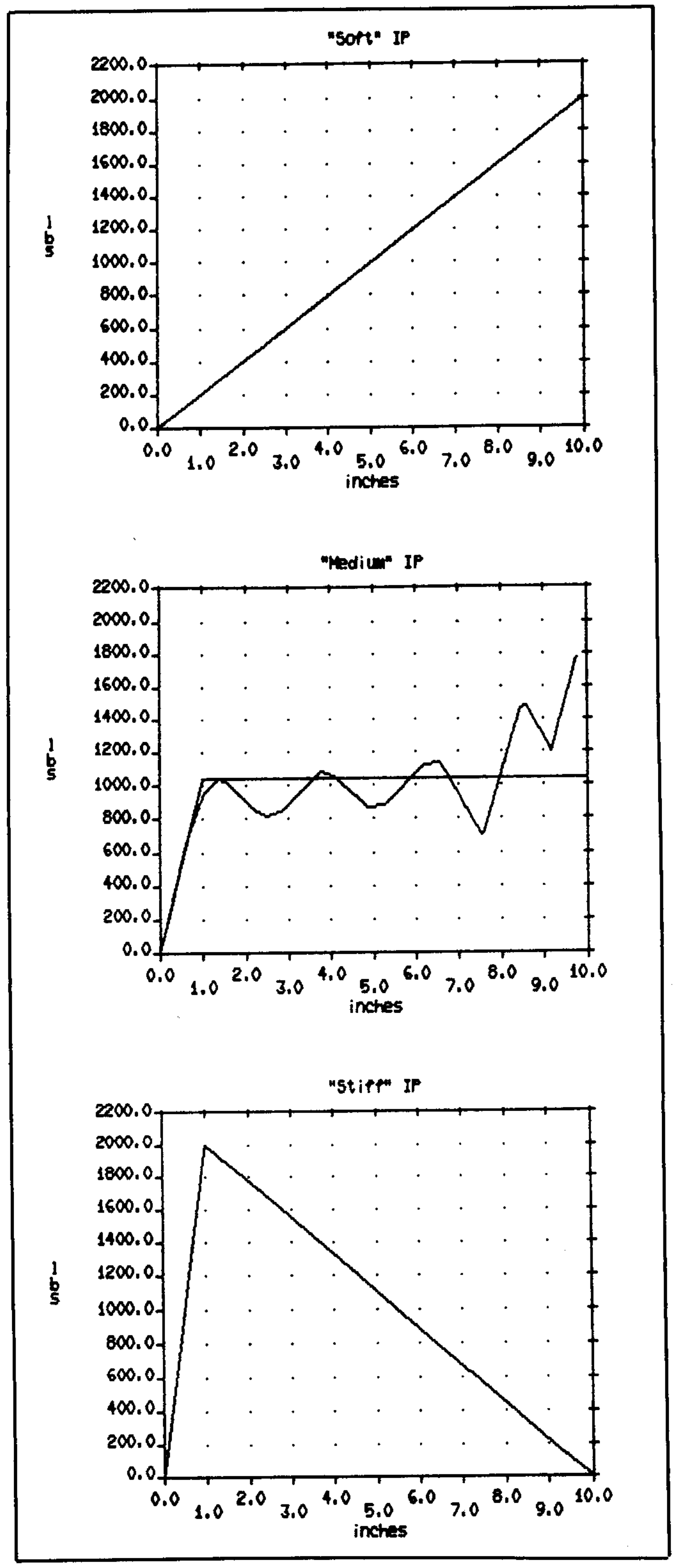

Figure 39. Modeled middle instrument panel material specifications

40 illustrate the force-deflection curves of each panel. The MGA experimental data is superimposed on the graphs. It can be seen from the figures that the original Celebrity vehicle had essentially a "soft"

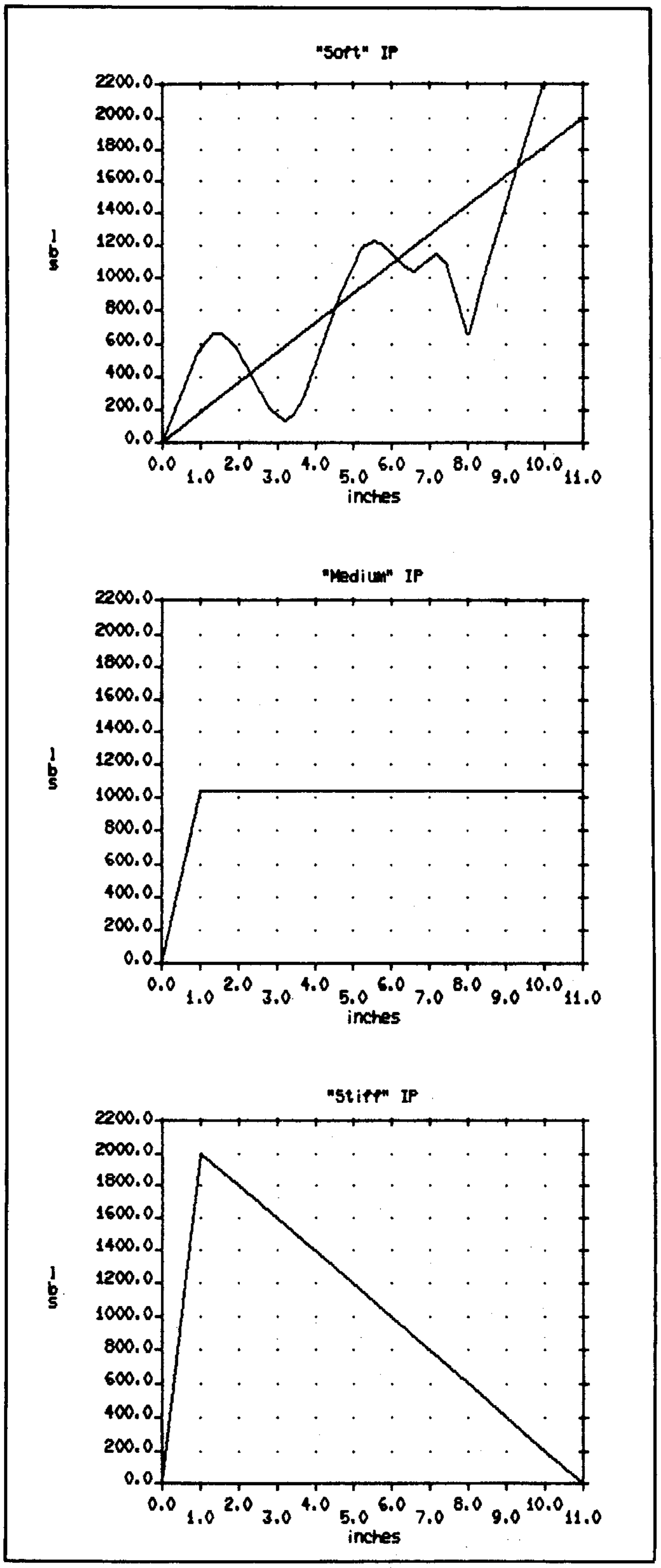

Figure 40. Modeled lower instrument panel material specifications

Table 6. - Test Matrix.

MIDDLE IP	soft	medium	stiff
LOWER IP	soft	soft	soft
MIDDLE IP	soft	medium	stiff
LOWER IP	medium	medium	medium
MIDDLE IP	soft	medium	stiff
LOWER IP	stiff	stiff	stiff

lower panel and a "medium" middle panel. Table 6 illustrates the simulation test matrix.

Simulation results are summarized in Tables 7 thru 9. In general, test results were more significant across the range of lower panel stiffnesses than across that of the middle panel stiffnesses. The knee strike into the lower panel occurred first, consequently the knee-lower panel interaction played an important role in the subsequent head and chest impacts.

In general, the cases involving a "stiff" lower instrument panel gave poorer results than any of the other cases. The stiff lower panel slowed penetration of the knees and the absence of significant penetration into the lower panel (approx. 5-inches) prevented the upper body from translating forward and impacting the middle of the instrument panel. Consequently, the forward momentum of the head and upper body caused the occupant to "jackknife". The head hit the windshield with a consistently higher impact velocity, plowed down the windshield, and hit the top of the instrument panel with considerable impact (17-18 mph). Chest interaction with the middle of the instrument panel was only incidental, so that the kinetic energy of the impacting occupant was almost totally absorbed by the head and knee impacts.

Head injury criteria (HIC) and peak femur forces with the "stiff" lower panel were the worst of all the cases run regardless of middle instrument panel stiffness (note: Because of this, the case of the "medium" middle panel was not run; the "soft" and "stiff" middle panel cases were virtually identical since the middle panel played almost no role in stopping the forward momentum of the impacting occupant).

The use of a "soft" lower panel more equally distributed the dummy's forward kinetic energy be-

Table 7. Injury criteria results, MVMA 2D simulation.

		INJURY CRITERIA			
		Middle Instrument Panel			
		soft	medium	stiff	
Lower Instrument Panel	soft	1024	881	828	HIC
		69	69	71	Chest clip
		1589	1592	1589	Femur Force
	medium	699	716	772	HIC
		46	51	54	Chest Clip
		961	908	962	Femur Force
	stiff	1057	-	998	HIC
		42	-	45	Chest Clip
		1625	-	1625	Femur Force

Table 8. Occupant penetration, MVMA 2D simulation.

Lower Instrument Panel		MAXIMUM DEFLECTION (inches) Middle Instrument Panel: soft	medium	stiff	
	soft	5.51	5.50	5.50	Windshield
		2.98	1.29	0.62	Middle IP
		10.13	10.09	10.07	Lower IP
	medium	6.20	6.20	6.21	Windshield
		1.89	0.72	0.42	Middle IP
		9.69	9.61	9.53	Lower IP
	stiff	4.54	–	4.53	Windshield
		0.02	–	0.00	Middle IP
		4.80	–	4.80	Lower IP

tween the head, chest, and knees. HIC was reduced approximately 10% and peak femur forces are moderately improved (2%). The "soft" lower panel however, allowed the knees to penetrate the lower portion of the instrument panel over 10 inches. This caused the chest to strike the middle of the instrument panel at a considerably higher contact velocity with a resulting increase in chest g's. Chest g's are significantly higher (approximately 60%) and exceeded the chest injury criteria level regardless of the stiffness of the middle panel.

The "medium" stiff lower panel combinations showed the best apparent improvement in injury criteria. HIC is decreased on the average about 30% from the "stiff" lower panel, and about 20% from the "soft" lower panel test runs. Chest g's are about 20g's lower than the "soft" lower panel cases and only slightly higher than the "stiff" lower panel cases (where chest to middle instrument interaction was minimal). Peak femur forces are also improved, approximately 40% less than the other lower panel configurations. Peak femur force however is applied over a longer duration with the "medium" panel than for the "soft" and "stiff" lower panel configurations (i.e., the "medium" panel femur force traces were more of a "square-wave" type than the "soft" and "stiff" panels where the femur force peaked then fell off). Thus the "medium" stiffness lower panel absorbed more of the occupant's kinetic energy by maintaining a consistent, yet moderate resistive force over a longer period of time.

This preliminary study of unrestrained occupant response during a frontal crash shows the lower portion of the instrument panel to be an important component in managing the forward kinetic energy of an impacting occupant. This is as expected, since the knee is the closest body part to the impacting surface and thus hits first consequently affecting the response of subsequent body impacts.

The initial comparisons made here show that a lower panel which absorbs knee impact at constant, yet moderate rate allows management of the occupant's forward kinetic energy such that subsequent head and chest impacts are mitigated.

Table 9. Occupant contact velocity, MVMA 2D simulation.

Lower Instrument Panel		OCCUPANT CONTACT VELOCITY (mph) Middle Instrument Panel: soft	medium	stiff	
	soft	28.3	28.3	28.3	Head
		14.9	14.8	14.7	Chest
		13.2	13.2	13.2	Knee
	medium	28.5	28.5	28.5	Head
		11.2	11.1	11.1	Chest
		13.2	13.2	13.2	Knee
	stiff	29.0	–	29.0	Head
		1.2	–	0.7	Chest
		13.2	–	13.2	Knee

Discussion and Summary

The objective of this program is to assess the safety problem associated with right, front seat passengers in frontal impacts. Injuries to the front seat passenger account for nearly one-quarter of the total harm to passenger car occupants, and approximately half of this harm is associated with frontal impacts. The primary focus of this initial effort is directed at defining the safety problems and the identification and evaluation of countermeasures for the unrestrained passenger. The safety problems associated with the restrained passenger will also be analyzed as part of the continuing efforts of this project. The development and evaluation of potential countermeasures will be based on the needs of both the restrained and unrestrained occupant and the associated safety problems as they are projected to exist in the future highway environment.

The present phase of this work is directed at defining the countermeasures to be analytically evaluated in reducing the harm associated with individual body regions.

1. The harm associated with chest and abdominal injuries to passengers are primarily composed of serious (AIS 3-6) injuries. The instrument panel is the primary source of injury. Based on the accident, sled test, and analytical data, it appears that these injuries occur at higher impact speeds compared to head, lower extremity, and injuries to other body regions. Mitigation concepts being explored in the present analytical phase of work include performance (force-deflection characteristics) modifications to the instrument panel to better manage chest contact loads at these higher speeds, and also concepts that would spread the loads to other body regions that may be less vulnerable. The failure mode of instrument panels with the resulting loss of integrity will also be further explored. Intrusion characteristics at the mid-instrument panel level are being investigated as to its affect on the force-deflection characteristics of the panel.

2. The harm to the head and face body regions consist of a large number of both non-minor (AIS 1

and 2) and serious injuries, and mitigation concepts need to be explored for both levels of injury. Anti-lacerative glass concepts are being explored in several NHTSA programs to address facial laceration harm (18). As to the serious injuries, the accident data indicates that the windshield is the primary source for the head. Based on the component testing and the sled/crash test work, mitigation concepts must be explored that address the ideal impact loading characteristics of the windshield itself. Also, the instrument panel performance and geometry characteristics that influence the manner in which the head first contacts the windshield needs to be evaluated. The precise manner of this initial contact appears to have a large effect on the extent of head rotation and neck bending, the level of acceleration, the degree to which the head penetrates into the windshield, and the degree to which the head plows down the windshield and makes contact with the instrument panel.

An area to be explored further includes the fact that the accident data indicate that the windshield is the major cause of serious head injuries, while the crash and sled test data indicate that the HIC levels rise to the levels that are associated with the most serious injuries(19) when the head impacts not only the windshield but also the header or upper instrument panel. This contradiction may be because when there are multiple head contacts in an accident, the windshield is the more obvious contact and, thus, may be implicated more often than the other components. Alternatively, other indicators of injury in addition to HIC such as angular acceleration may need to be considered in understanding the injuries caused by windshield contacts. Several reports based on laboratory and analytical efforts have reported significant levels of head rotational acceleration for head to windshield contacts(20,21). In exploring mitigation concepts, the performance characteristics of the top of the instrument panel, the cowl area, and the juncture of the windshield and instrument panel will be considered for possible head contacts. Also, additional indicators of injury such as head rotational acceleration and neck bending will be analyzed.

Also, as indicated by the sled test work and the analytical modeling, the mid-instrument panel characteristics must also be considered for possible head contacts by smaller occupants, and the header characteristics must be addressed for head contacts by larger occupants. Head to A-pillar contacts which is a major cause of serious injury is being addressed by another program at NHTSA and a paper being presented at this conference is presenting the status of that program.

3. Lower extremity injuries account for a large percentage of both non-minor and serious injuries. The failure characteristics of the lower instrument panel must be examined as to its possible role in increasing the laceration potential to lower extremities and in exposing hard spots under the panel. The lower instrument panel needs to be examined for both its ability to better manage the direct impact loads of the lower extremities, and, also, its ability to influence the kinematics of the occupant to improve the loading characteristics of the head/windshield and the chest/middle instrument panel contacts. Based on the crash and sled tests and the analytical modeling, the geometry and the force-deflection characteristics are critical in influencing the kinematics. Intrusion related to the lower instrument panel and the toeboard also needs to be further investigated as to how it affects the force-deflection characteristics of the panel and occupant kinematics. Entrapment of the lower leg in cases of toeboard intrusion and its association with increased risk of lower extremity injuries needs to be explored. The potential displacement upward of the instrument panel when the knees push into the panel needs to be considered in the development of mitigation concepts.

4. As to whole body motions, the preliminary analytical work performed has indicated that the relative geometry and force-deflection characteristics need to be considered in the development of mitigation concepts. The deceleration of each body segment must be phased so as to optimize the proper loading and relative rotation of each body segment. Modifications suggested in the literature related to this concept have stressed maintaining the torso in an upright posture during the crash event(9,22).

The Phase II—Concept Analysis work is continuing to explore these safety problems and alternative mitigation concepts using the MVMA 2-D model. Future studies will expand on the approach of the case model study. Cases will evaluate the consequences of the positioning of the vehicle interior impact surfaces and occupant, an optimization of the force-deflection properties of the impacted surfaces for each of the baseline vehicles, and a consideration of the effect of structural intrusion on injury mitigation schemes. Future work will focus on both the windshield and the instrument panel surfaces. The baseline case models will be expanded to include different sized occupants as well as a spectrum of crash speeds and crash modes. In addition, the safety problems and potential countermeasures associated with restrained passengers will be analyzed to insure that any mitigation concepts are compatible for both the restrained and unrestrained occupant.

Based on the results of the Phase II analysis, Phase III will be initiated to explore a limited number of mitigation concepts in a laboratory setting by incorporating the modifications into baseline vehicles and conducting either sled or crash test of these vehicles.

The discussion and conclusions in this paper represent the opinions of the authors and not necessarily those of the NHTSA. The United States Government does not endorse products or manufacturers. Trade or manufacturer's names appear herein solely because they are essential to the object paper. This document is disseminated under the sponsorship of the Department of Transportation in the interest of information exchange. The United States Government assumes no liability for the contents of use thereof.

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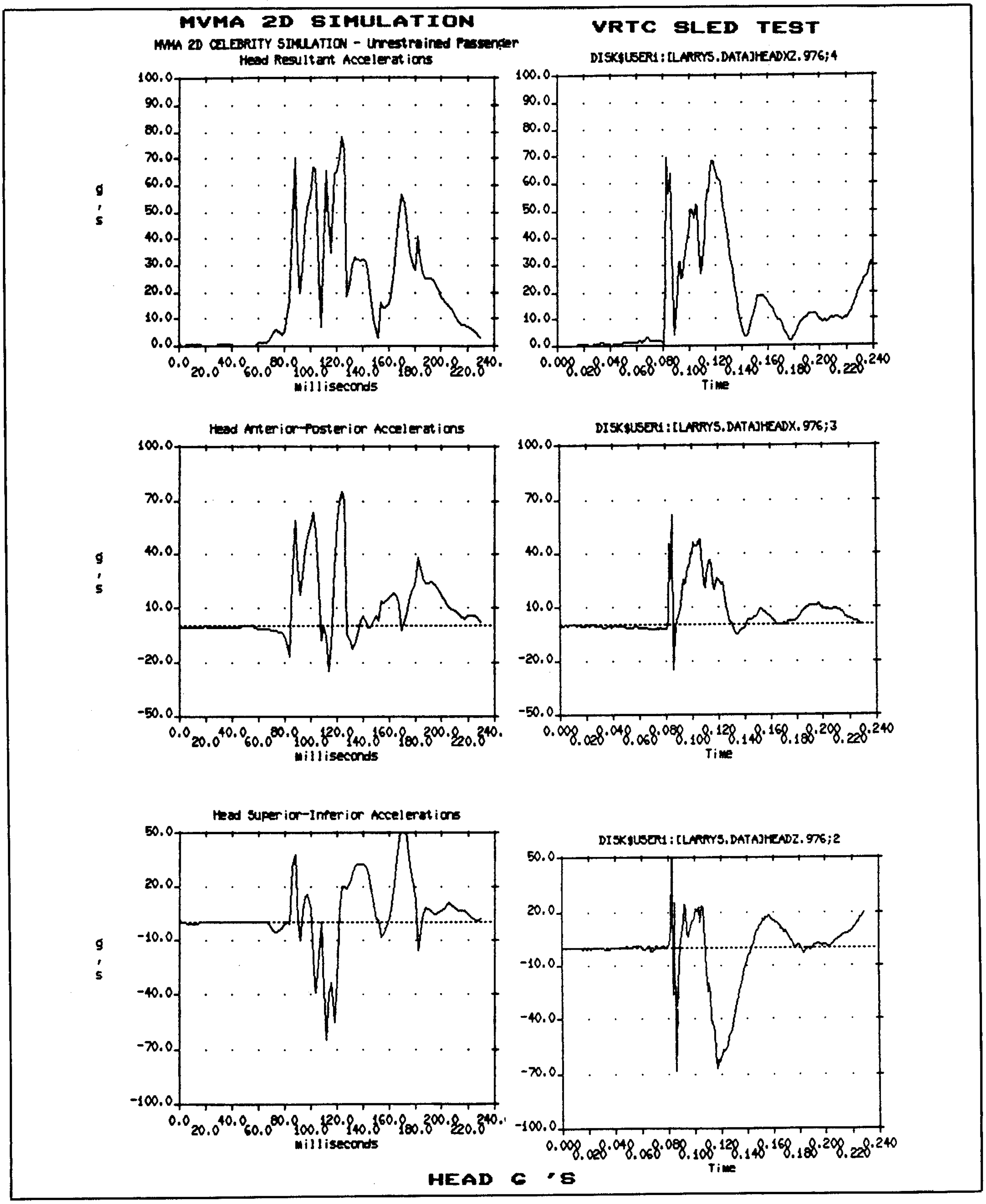
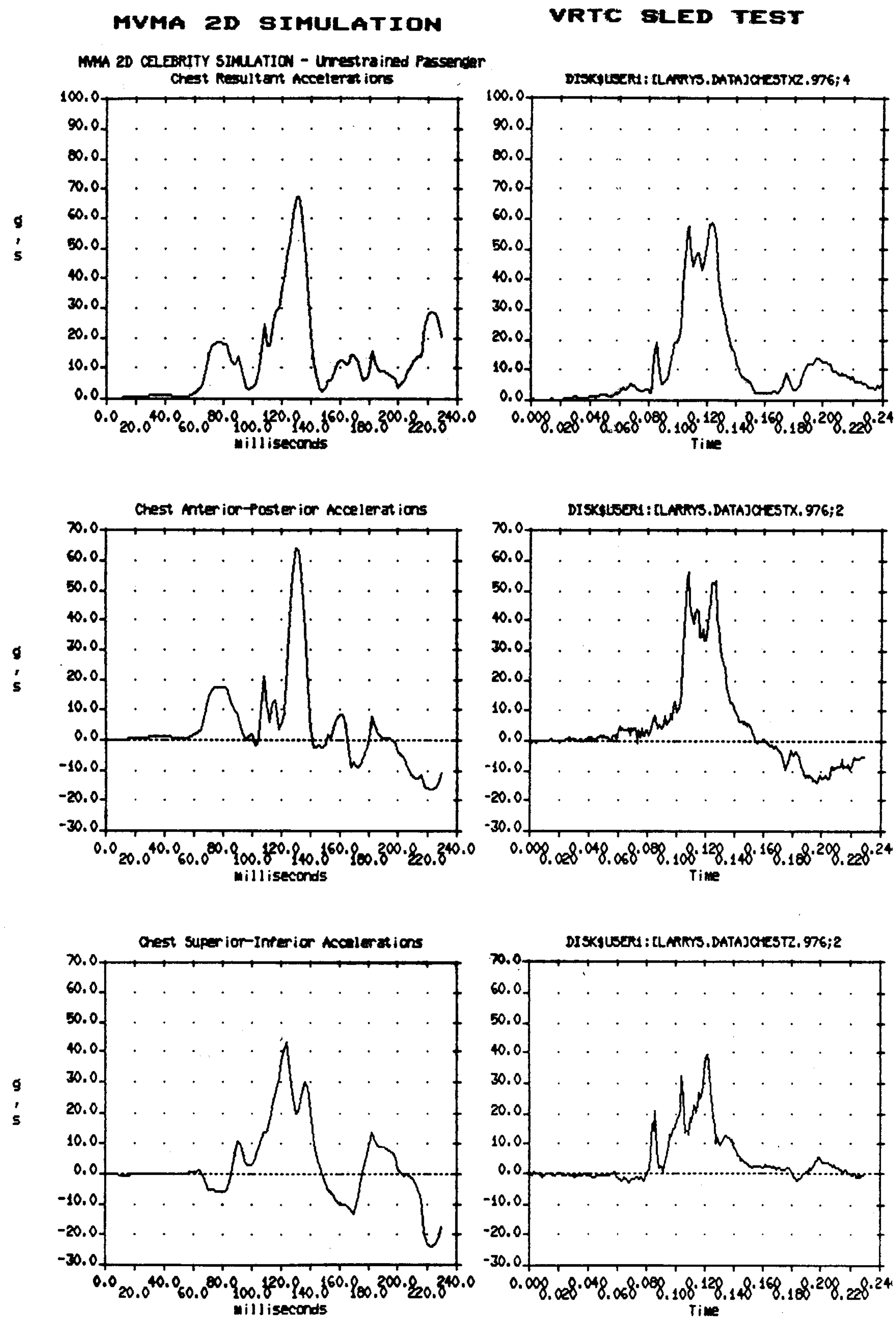
APPENDIX A

MVMA 2D SIMULATION OF CELEBRITY SLED TEST




MVMA 2D SIMULATION
VRTC SLED TEST
MVMA 2D CELEBRITY SIMULATION - Unrestrained Passenger
Chest Resultant Accelerations
DISK$USER1:[LARRYS.DATA]CHESTXZ.976;4
Chest Anterior-Posterior Accelerations
DISK$USER1:[LARRYS.DATA]CHESTX.976;2
Chest Superior-Inferior Accelerations
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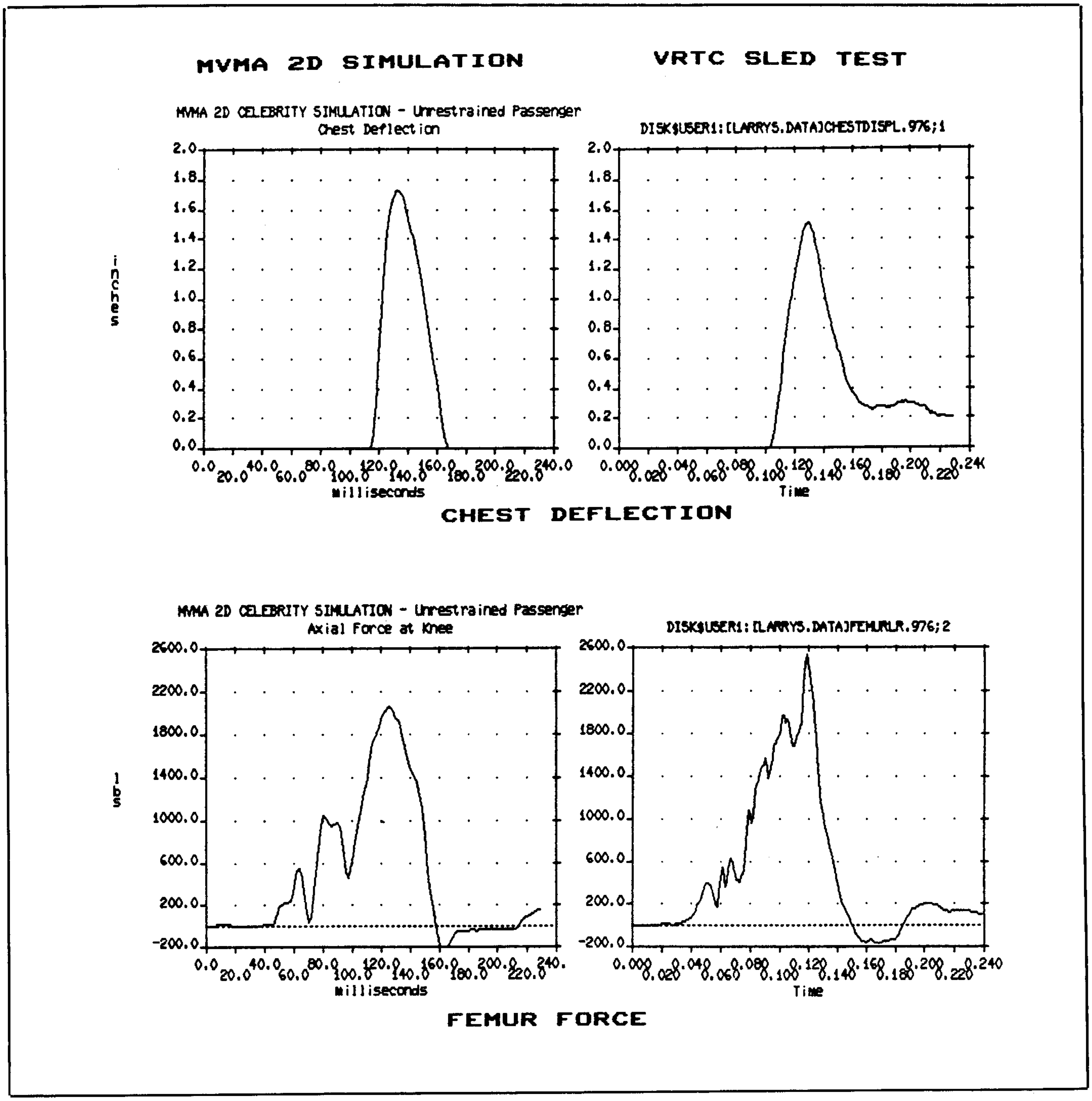
g's
milliseconds
Time

CHEST G 'S


MVMA 2D SIMULATION
VRTC SLED TEST
MVMA 2D CELEBRITY SIMULATION - Unrestrained Passenger
Chest Deflection
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inches
milliseconds
Time
CHEST DEFLECTION
MVMA 2D CELEBRITY SIMULATION - Unrestrained Passenger
Axial Force at Knee
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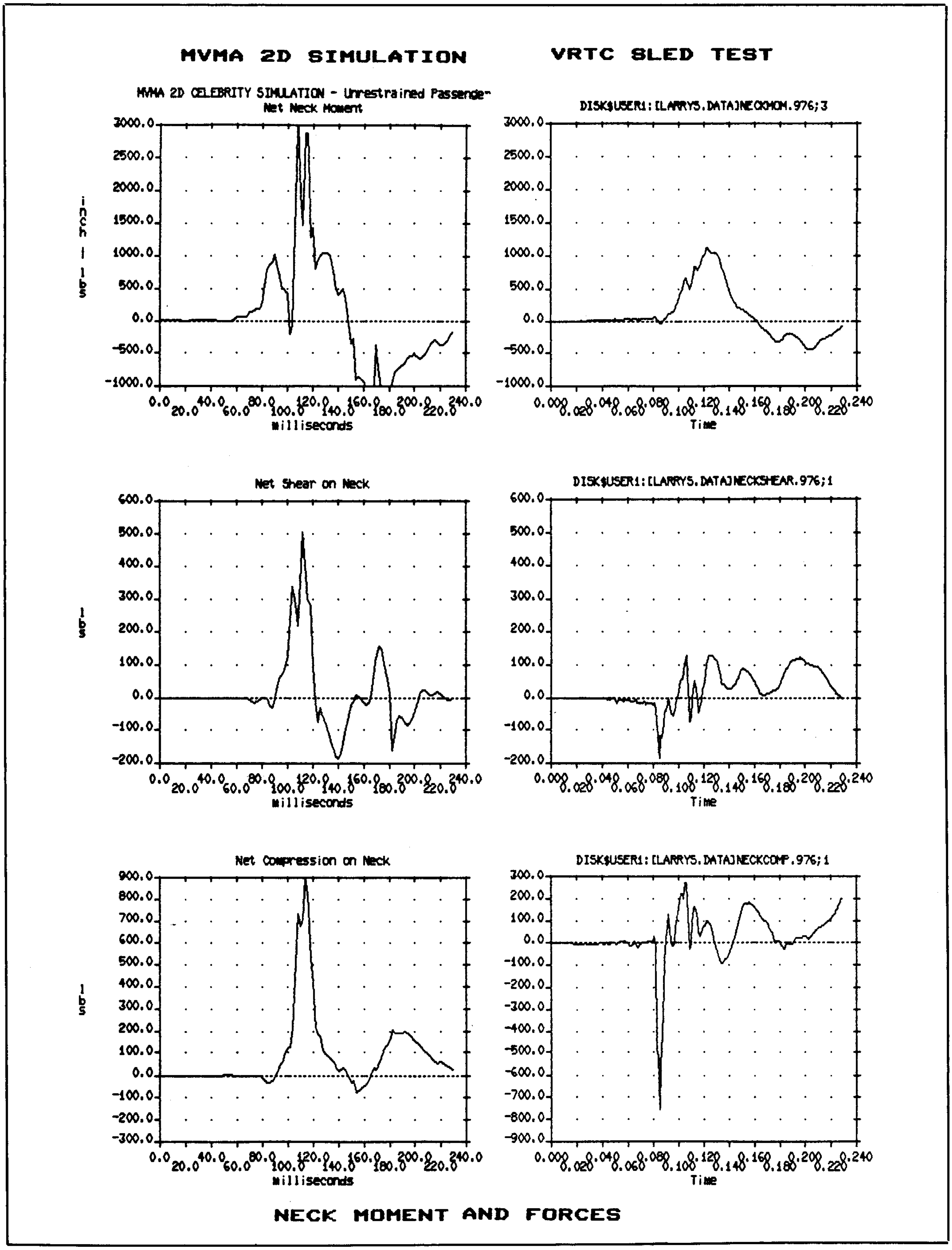
lbs
milliseconds
Time
FEMUR FORCE



MVMA 2D SIMULATION
VRTC SLED TEST
MVMA 2D CELEBRITY SIMULATION - Unrestrained Passenger
Net Neck Moment
DISK$USER1:[LARRYS.DATA]NECKMOM.976;3
inch - lbs
milliseconds
Time
Net Shear on Neck
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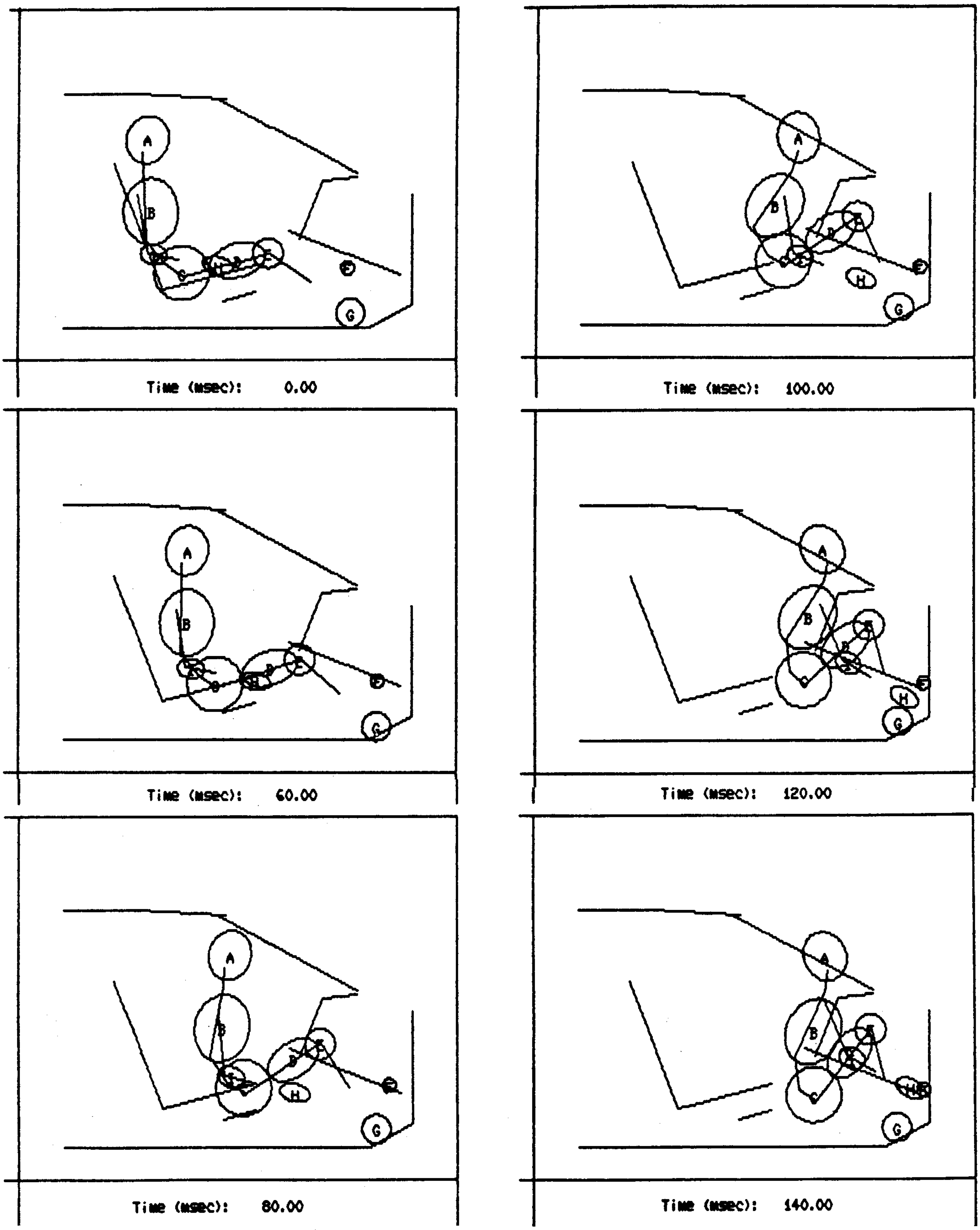
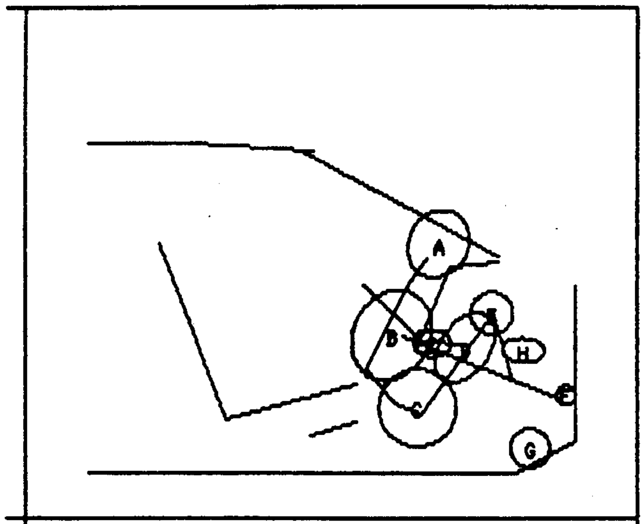
NECK MOMENT AND FORCES

OCCUPANT KINEMATICS





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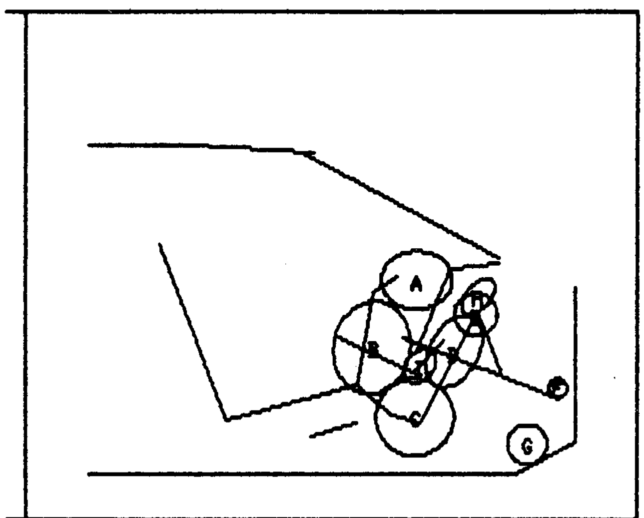

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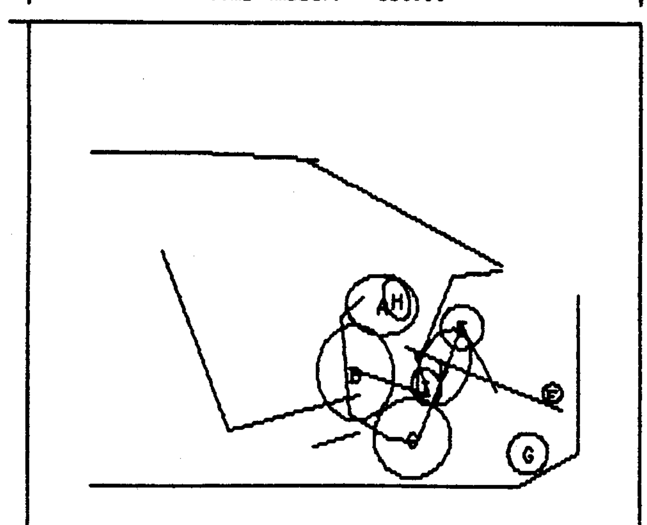

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Progress Towards Improving Car Occupant Protection in Frontal Impacts

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Abstract

Further improvements in the protection of car occupants involved in frontal impact accidents can be achieved only by a careful balance between what is needed for increased safety, what is technically possible and what can be afforded. Some steps in this direction, which have not yet been optimised, are presented in this paper and have been incorporated into the Demonstration Safety Car, ESV 87. A large reduction in intrusion has been achieved by developing an energy absorbing front structure and by reinforcing the passenger compartment to achieve compatible dynamic stiffnesses. It has been shown that even with small cars it is possible to absorb virtually all the impact energy in the frontal structure, with only slight deformation of the passenger compartment. Further improvements would require a reduction in forward movement of the occupants. Simulation studies have shown that this can be achieved by properly matched seat belt pre-tensioners. Fitted to the car is a safer steering wheel designed to reduce the likelihood of driver head injuries. A user acceptability study is reported which shows that most users liked many aspects of this design of steering wheel. Future work on head impact area testing is also outlined.

Introduction

Seat belt wearing rates, amongst front seat car occupants in the UK, are in excess of ninety percent. Consequently in frontal impacts, the majority of serious and fatal casualties which now occur have been involved in fairly high speed collisions of severity similar to or greater than a thirty degree angled barrier test at 60 km/h. In almost all cases, the severity of injury is exacerbated by the occurrence of intrusion into the passenger compartment. Improved protection may be provided by:

1. Reducing intrusion by providing for adequate energy absorption in the car's frontal structure and by strengthening the passenger compartment to ensure the preferential collapse of the energy absorbing structure.
2. Improving seat belt systems to reduce forward movement and seat belt loads.
3. Padding or weakening of interior structures, in particular steering wheels and other head impact areas, susceptible to occupant contact.

A modified small car, with improvements for pedestrian protection, was presented to the 10th ESV conference(1). This car has been further developed to incorporate improved protection for its occupants in frontal and side impacts (Fig 1). These improvements are discussed in this paper and the companion paper presented to the side impact session(2). Computer simulations and impact tests have been used to guide these modifications. This demonstration car, now named ESV 87, illustrates the principles involved. In new car designs, such improvements could be achieved in a number of different ways.

Accident Investigation

Accident Investigation has shown that, even for those wearing seat belts, car occupants face the greatest risk of serious injury from frontal collisions(3). In the majority of these collisions, the impact is concentrated on part of the width of the car with only one side of the car's main structure having to absorb the impact energy. Despite the presence of seat belt loading, belted casualties most frequently receive their serious injuries from contact with the car's interior. The occurrence of intrusion into the passenger compartment makes such contacts more likely and often results in the contacted area having a more aggressive shape or stiffness. The problem of intrusion in frontal impacts is greater with small cars where the shorter engine compartment tends to be less able to absorb the impact energy.

Because seat belts are efficient in reducing the probability of injury in frontal impacts, further significant advances require testing at higher speeds than was appropriate when unbelted occupants were the prime concern. A previous accident study related probability of injury to crash severity for belted front seat occupants of cars in frontal impacts(4). The best

Figure 1. Demonstration safety car—ESV 87

estimate of the probability of suffering an MAIS 3 or greater injury was given as 36 percent with a velocity change of 48 km/h (30 mph) and 67 percent with a velocity change of 64 km/h (40 mph). To determine overall injury risk, these probabilities have to be weighted by the frequency of occurrence of the different impact severities.

The Principles Behind Intrusion Control and Energy Absorption

The main structural strength of a conventional front engined car is provided by longitudinal box sections positioned at each side of the engine compartment. With some cars such as this one, an additional load path exists through the engine subframe. Although in a perpendicular block test the structure on both sides of the car absorbs energy, in most frontal impact accidents, the loading is taken on only one side of the car. Consequently the structure of cars designed to pass a perpendicular block test are easily overloaded in a partial engagement impact. As a result, insufficient energy is absorbed ahead of the passenger compartment and so it has to collapse, to absorb the excess impact energy.

Often an additional problem arises. As a result of the low transverse beam strength at the front of the car, loads are inadequately transferred to the main structure. This is illustrated by the way the bodywork collapses past the end of the main box section leaving it protruding (Fig 2). When this happens the energy absorption capacity of the car's front is reduced and again the passenger compartment has to absorb the excess energy.

A further problem arises when the loading of the main box sections is not axial. High bending moments are generated in the box section and it bends, usually at its junction with the firewall. This bending absorbs much less energy than would be absorbed if the box section were to collapse axially as in a perpendicular block test. Pre-failing the front part of the box section is unable to prevent this bending.

It should be possible to absorb all the energy in an impact without any passenger compartment collapse. For this to be achieved, sufficient depth for energy absorbing structure must be provided ahead of the passenger compartment. To ensure that this structure collapses, in preference to the passenger compartment, it must have a lower dynamic stiffness. In practice, the collapse depth of the frontal structure is limited by the space taken by incompressible underbonnet equipment such as the engine. In all cars which have small engine compartments, the depth available for energy absorption is small and so the dynamic stiffness of the absorbing structure must be high. Consequently the passenger compartment must be even stiffer to ensure that it does not collapse.

Modifications to ESV 87

The standard production car meets current European safety legislation, with its impact performance being typical of current small cars. However, as with all current production cars, its crash performance can be improved to give increased protection in more severe accidents.

Passenger Compartment Reinforcement

In the standard car, the main load paths into the passenger compartment are through the subframe into the floor, through curved box sections into the A-posts at waist level, through the engine into the firewall and through the front wheels into the sills. Each of these load paths was reinforced. Loads from the subframe mounts were taken through box sections extending the full length of the floor and outwards to the sills (Fig 3). The A-posts were supported by the doors which were strengthened with tall beams. Loads from the doors were taken through the B-posts to the

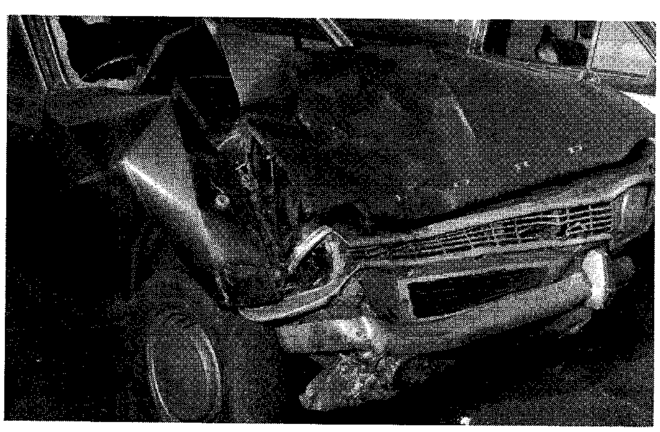

Figure 2. Accident-damaged car showing protruding main box section

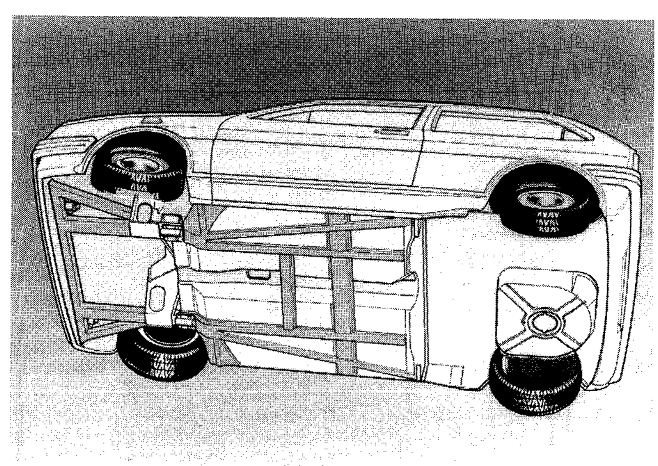

Figure 3. Structural modifications to the underside of ESV 87

rear wheel arches. From the firewall, loads were taken through a raised tunnel to the strengthened floor. The sills were strengthened by adding a fillet between the sill and the floor (Fig 4).

To reduce the possibility of the occupant "submarining" the front of the seat base was strengthened, and to prevent the seat breaking free from the floor, stronger seat runners and supports were used.

Provision of an Energy Absorbing Front Structure

The standard subframe is very strong, being designed to take engine and suspension loads. Because of its strength, the floor, against which it is mounted, collapses before it does. The subframe collapses by forming one or two plastic hinges each of which absorbs some strain energy. Similarly, the curved box sections absorb energy in forming plastic hinges but again the amount is limited by the number of hinges produced. In a partial engagement impact, or in one with an angled barrier, little energy is absorbed by the side of the car remote from the impact. Also the structure immediately behind the bumper, at the corners of the car, is quite weak. Because of this, little energy is absorbed in the early part of a partial overlap or angled barrier impact.

The following changes were made to increase the energy absorption capacity of the front structure (Fig 5):

1. The engine subframe was redesigned incorporating energy absorbing tubes and a strong front beam, the beam being positioned immediately behind the pro-pedestrian bumper.
2. The curved box sections were triangulated at their junction with the firewall to reduce the likelihood of bending at this point and so increase the number of plastic hinges formed in the impact.
3. An energy absorber was positioned between the engine and the firewall to load progressively the firewall prior to its contact with the engine.

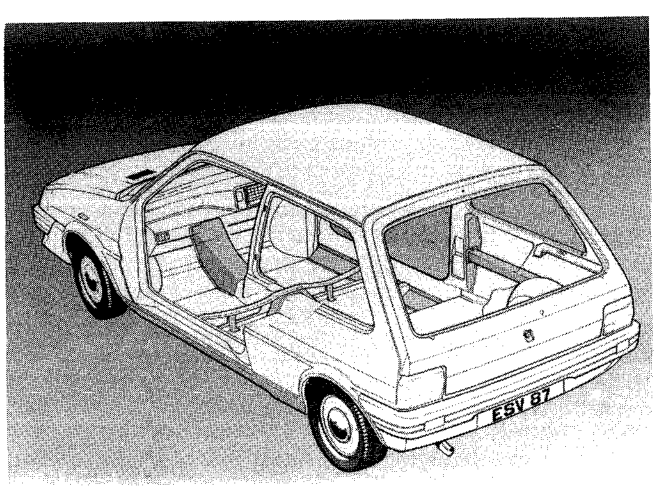

Figure 4. Structural modifications to the interior of ESV 87

Design of the Energy Absorbing Subframe

Four rectangular section steel tubes were incorporated in the subframe to absorb energy by crumpling. To reduce the size of the peak load, necessary to initiate failure, the front section of each tube was pre-failed. If the peaks were not removed, they could initiate early failure of other parts of the car's structure. To help accommodate non-axial loads the outer pair of tubes were angled outwards.

To maximise the available crush distance and commence energy absorption as early as possible, the front beam was positioned immediately behind the front bumper. The beam extended across the full width of the car with the intention that the loads from impacts at any point across its width would be transferred into the tubes on each side of the car.

Because the subframe is mounted low in the car and the front beam is positioned behind a bumper designed for pedestrian leg impact protection, this design should not pose compatibility problems in pedestrian impacts. For the same reason it is not thought that compatibility in side impacts would be reduced. Further work is required to resolve the different demands posed by different impact situations.

Results of an Angled Barrier Test

The car was impacted into a 30 degree angled barrier at 60 km/h. The barrier angle was such that the driver's side of the car struck first. Two instrumented OPAT dummies(5) were seated in the front of the car and were restrained with conventional inertia reel seat belts. The results of this test were compared with those of a standard car which had been tested in a similar way.

Intrusion Prevention

The modifications resulted in substantially lower levels of intrusion, despite the fact that the test was more severe than that specified in ECE Regulation 33, which calls for a perpendicular impact at only 50 km/h. The modified car easily passed all aspects of the requirement (Table 1) but with the standard car, rearward intrusion of the driver's footwell was greater than that stipulated and intrusion at the instrument panel was only just within the requirement.

The reduced intrusion levels are evidence that most of the impact energy was absorbed in the front structure. The overall deceleration was necessarily

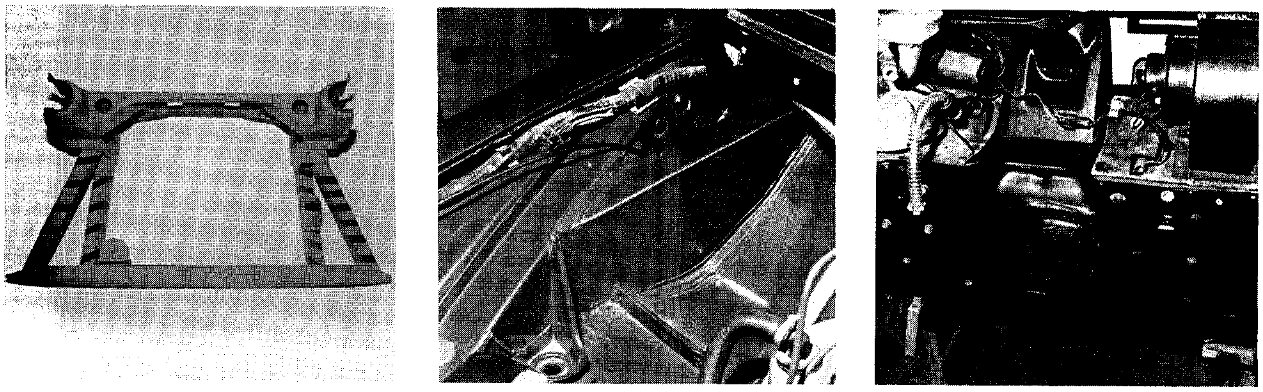

Figure 5. Engine subframe with front beam and energy absorbing tubes, box section triangulation and firewall energy absorber

greater than that of the standard car, as the modified car had to be stopped in a shorter distance. However the deceleration peaks were smaller and the loads remained more constant, particularly on the offside (Table 2). With the standard car the loads were seen to rise and fall rapidly, as additional parts of the structure started to fail. The more constant loading with the modified car was an indication that the subframe had performed as intended.

The transverse beam succeeded in transferring some of the load to the side of the car remote from the impact. This is illustrated by the fact that footwell intrusion on the passenger's side of the modified car was slightly more than on the standard car, despite its greater stiffness. Because the beam extended across the whole width of the car, corner loads were transferred to the subframe early in the impact. This resulted in the car's deceleration building up earlier in the impact.

The triangulation of the curved box section prevented bending at its root and may have increased the amount of energy absorbed by the box section. Unfortunately from this type of test, it is not possible to determine how much energy was absorbed by the different parts of the car's structure.

The energy absorber fitted between the engine and the bulkhead bent at its root being unable to cope with the non-axial load imposed by the engine. On the demonstration car a pyramid shaped absorber is fitted, though it has not at this stage been tested.

The static deformation of the car after modification was less than half that of the standard car, partly because of the reduced collapse of the passenger compartment. The rearward displacement of the A-post was reduced from 250 mm at its base and 140 mm at the waistline to 60 mm at both base and waistline.

Effects of the Modifications on Dummy Parameters

The results of the test were compared with those from an earlier test on a standard car. In some cases the data was not available in exactly the same format. Where possible, directly comparable data has been given in this paper.

The philosophy behind the design of the car allowed for the eventual incorporation of seat belt pre-tensioners with characteristics appropriate to the deceleration characteristics of the car. Such pre-tensioners should reduce seat belt loads and forward movement but were not available at the time of the test.

As expected, the more rapid deceleration of the modified car resulted in higher seat belt loads and consequently higher thorax acceleration (Table 3). The

Table 1. Occupant survival space (mm).

	Before Impact		After Impact				
			Standard Car		ESV 87		Reg. 33 Requirement
	Dvr	Pass	Dvr	Pass	Dvr	Pass	
Instrument Panel to R-point	610	630	455	565	545	610	>=450
Front of Compartment to R-point	930	920	585	860	780	850	>=650
Footwell Width(a)	610	610	540	570	535	490	>=250

(a) The footwells of ESV 87 were narrower because of the raised tunnel (The before test widths were: Driver 550 mm and Passenger 510 mm).

Table 2. Car acceleration and structural crush.

	Standard Car	ESV 87
Impact velocity (km/h)	60.3	57.9
Weight at test(a) (kg)	895	1040
OS B-post peak acceleration (g)	103 @ 62 ms.	65 @ 43 ms.
NS B-post peak fore/aft acc.(g)	45 @ 43 ms.	44 @ 46 ms.
Offside static crush (mm)	1190	480
Nearside static crush (mm)	50	20
Offside wheelbase shortening (mm)	489	190
Nearside wheelbase shortening (mm)	38	10
OS A-post base displacement (mm)	250	60
OS A-post waistline displacement (mm)	140	60

(a) The weight of instrumentation and cameras was different in each test.

Table 3. Summary of dummy data from 30 degree angled barrier impact at 60 km/h.

	Standard Car Driver	Standard Car Passenger	ESV 87 Driver	ESV 87 Passenger
HEAD				
HIC	954	1616	-(a)	597
Resultant acceleration				
3ms exceedence(g)	126	154	-(b)	80
80G exceedence(ms)	5.9	7.1	-(a)	2.8
CHEST				
peak acceleration (g)	44	46	59	59
3ms exceedence(g)	42	41	54	-(a)
Gadd Severity Index	302	333	518	-(a)
PELVIS				
peak acceleration (g)	75	47	63	62
LEFT FEMUR				
peak force (kN)	6.7	1.1	7.2	1.1
RIGHT FEMUR				
peak force (kN)	14.3(c)	3.5	1.7	2.0
SHOULDER BELT				
peak force (kN)	4.2	6.4	9.4	8.7(d)
LAP BELT				
peak force (kN)	3.2	5.1	8.4	8.5

(a) Instrumentation failed
(b) Peak fore/aft acceleration was 65 g.
(c) Prior to the dummy being damaged
(d) Prior to belt failure

seat belt loads were high on both dummies, the peak shoulder belt loads being 9.4 kN for the driver and 8.7 kN for the passenger. Lap belt loads were also high but provided the seat belt locates on the pelvis and not on the abdomen, such loads should not be a problem.

Later in the impact the passenger's seat belt broke. This was caused by the belt bunching at the B-post loop a phenomenon which has been seen before in tests, though not in accidents, and which is thought to be due to the presence of the seat belt load measuring transducer. The consequence of this failure was that a number of connections to the dummy were broken and some data were lost. In this test, the data from the passenger dummy are less important than that from the driver dummy, which is seated on the impact side. Despite the belt failure the passenger's HIC was below 600. In the standard car, the passenger's head hit the fascia top and recorded a HIC of about 1600.

Because of instrumentation problems, the HIC value and vertical acceleration of the driver's head could not be obtained. The driver's head hit the steering wheel, which was of the "safer design" discussed elsewhere in this paper. The peak fore and aft acceleration, which was recorded, was 65 g.

When incorporated in the car, a seat belt pre-tensioner should reduce forward movement and the likelihood of driver's head contact with the steering wheel. If contact did occur, it should be at a lower head velocity. It should also reduce the levels of seat belt loads and the risk of damage to the shoulder or rib cage.

In the test on the standard car, contact between the driver's knee and the intruding structure gave rise to a very high right femur load. A force of 14.3 kN was measured after which the dummy was damaged. In the modified car, the reduction in intrusion improved the situation considerably. All the femur loads were well below the target level. Improvements in the design of the fascia could reduce these loads further.

Computer Simulation of Seat Belt Systems

Computer simulation studies have been made using a modified version of the Calspan 3-D model(6). The effect of changing various seat belt parameters was evaluated in terms of their effect on forward movement of the head and its velocity when it crossed the plane of a typical steering wheel in its displaced post-crash position. The simulations clearly indicated that, in order to get a sizeable reduction in head velocity and forward movement without incurring high seat belt forces, pre-tensioning of the shoulder belt was necessary (Table 4).

Progress on Safer Steering Wheels

At the 10th ESV Conference a method for determining the aggressiveness of a steering wheel to a driver's face and cranium was described(7). An in-depth accident study(8) has shown that, from a total of 761 belted drivers involved in frontal impacts, 339 (45%) received injuries >=AIS 1 to the head. The steering wheel was identified as being responsible for causing injuries to the face in 139 (41%) of cases and to the cranium in 51 (15%) of cases. The distribution of injuries by location of impact on the steering wheel is given in Table 5.

Matched hospital inpatient and road accident data(9), for the period during which compulsory seat belt wearing was introduced, has been analysed. The analysis indicates that some 2000 drivers per year, in Great Britain alone, require hospital inpatient treatment for head injuries. Many of these casualties could benefit from a safer design of steering wheel. A safer steering wheel, made by Sheller-Clifford to production standards, was exhibited at the 10th ESV Conference. The wheel was fitted to a small car and subjected to a free flight upper torso (Blak Tufy) test(10,11) and the proposed European facial impact test. The safety wheel passed both tests.

The design of the Safer Steering Wheel, with its energy absorbing hub and modified rim and spokes,

Table 4. Potential improvements due to seat belt modifications from computer simulation.

Modification from standard belt(a) with 12% webbing(b)	Change in Max. Forward Head Movement (%)	Change in Head Velocity (%)	Change in Max. Shoulder Belt Tension (%)
4% webbing	- 16	- 18	+ 24
yield(c)	- 4	- 7	- 5
tensioner(d)	- 14	- 18	- 20
4% webbing+yield	- 15	- 21	+ 28
4% webbing+tensioner	- 27	- 33	- 18
4% webbing+yield+tensioner	- 34	- 38	+ 20

(a) standard 3-point inertia reel belt
(b) percentage stretch with 9.8kN (1000 kg) load.
(c) yielding lap belt
(d) shoulder belt pre-tensioner with load of 2.7 kN @ 20 ms.

Table 5. Head injuries to belted drivers related to impact location on the steering wheel.

Location of Impact	Facial Injury (>=AIS 1) No.	(%)	Cranial Injury (>=AIS 1) No.	(%)
Rim	40	(29)	16	(31)
Spoke	2	(1)	2	(4)
Rim and Spoke	11	(8)	7	(14)
Hub	37	(27)	9	(18)
Not Known	49	(35)	17	(33)
Total	139	(100)	51	(100)

was sufficiently different from current designs for its user acceptability to be questioned. It was therefore decided to carry out a user acceptability survey. For the survey, the size of the hub was reduced to improve visibility of the instruments. The wheels were fitted to Army, Police and other cars used in Government Service. Questionnaires were provided for completion at the end of each journey.

A total of 248 questionnaires were returned, from 189 men and 18 women. Ages ranged from 18 to 60 and mileage travelled ranged from one to 9000 miles. Ninety-six percent of users said they liked the wheel for driving and manoeuvring. Only ten said they altered the way they held the wheel. When asked questions about visibility of the instruments and the operation of the controls on the steering column, 93 percent said they liked the design. The most common adverse comment related to difficulty in seeing a warning light. Some 89 percent thought the wheel was comfortable in use and this could be related to the positioning of the rim reinforcement, which had been moved away from the driver to reduce facial loads. Of the four design features thought to be potentially less acceptable, only nine percent thought the rim was too thick, four percent thought it was too soft, four percent thought it was too slippery and ten percent thought the spokes were too wide. Each of these features could be changed in future designs. When asked for their overall impression compared with the standard design, 69 percent preferred the safety wheel and 11 percent preferred the standard wheel. Of the 20 percent who expressed no preference, the majority had made positive responses to earlier questions.

Head Impact Zone Padding

Even for those wearing seat belts, serious or fatal injury to the head is the most common problem in frontal impacts. These injuries are virtually all caused by head contact most frequently with the interior of the car. In some cases, the part contacted is outside the area covered by head impact test(12). This is particularly the case for occupants larger than fiftieth percentile.

Furthermore, because the test uses a rigid head form, no account is taken of any localised loading. An investigation is being carried out to test the feasibility of using a flesh covering over the headform or using a deformable headform to overcome this deficiency. The headform stiffness could then be matched to the nominal strength of the skull.

Conclusions

Even with the almost universal wearing of seat belts in Great Britain, frontal impact accidents are still responsible for causing most serious and fatal injuries to car occupants. This paper reports work intended to point the way forward. Intrusion into the passenger compartment is a frequent contributor to injuries. The Demonstration Car, ESV 87, illustrates how a car can be modified to reduce intrusion levels, although at some weight penalty. In a 60 km/h, thirty degree angled barrier impact test, dangerous intrusion was virtually eliminated. Even if intrusion were to be totally eliminated, forward movement would still be too large to prevent the possibility of occupant contact with the interior. Seat belt pre-tensioners should reduce this likelihood and the fitting of safer steering wheels should protect against facial injuries in the most frequent type of head contact, which is unlikely to be completely eliminated. New procedures for headform testing which check for localised loading of the head are still being evaluated.

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Steering Column Intrusion—Restrained and Unrestrained Occupant Effects

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Abstract

Research and compliance testing of steering assemblies has shown that new columns perform at or near their full energy absorbing potential. Accident data analysis, however, indicates that the EA units seldom exhibit compression which utilizes their full capability. The cause of the discrepancy between the test results and the accident data is unknown, although a number of explanations have been offered. Each accident is unique, however, and the column behavior in the varied accident modes is difficult to assess.

A frontal crash test matrix conducted by the National Highway Traffic Safety Administration provided an opportunity to examine the steering column performance under a variety of frontal crash configurations. The analysis presented in this paper was initiated with the objective of documenting the extent and nature of steering column intrusion for selected matrix tests. Static and photographic measurement methods were employed in documenting the steering column intrusion.

The results show significant differences between dynamic motions and residual displacements. They also show column performance being dependent upon both the vehicle and the crash configuration. The effect of these results on restrained and unrestrained occupant response is included in this paper.

Introduction

As early as the 1950's, the steering assembly was identified as the leading cause of injury in motor vehicle accidents. It continues to be, and is a reflection of the high percentage of accidents with frontal principal force directions, and the location of the steering assembly relative to the driver. In an effort to reduce the number of serious and fatal injuries produced by the steering assembly, considerable research has been conducted by private industry, and the U.S. and foreign governments. This research led to the development of energy absorbing (EA) steering assemblies, and Federal Motor Vehicle Safety Standards (MVSS) 203 and 204. FMVSS 203 and 204 established performance requirements for the steering column energy absorption and rearward displacement into the occupant compartment, respectively.

MVSS 203 and 204 have been cost effective in reducing fatal and serious injuries. It has been estimated that 1300 fatalities and 23,000 serious injuries are prevented annually, and that the rearward displacement of the column into the occupant compartment has been reduced by 81%(1). This has been accomplished for between \$10-\$12 (1978 dollar costs) life-time cost of the vehicle. Despite the improved column performance, the steering assembly continues to be the leading cause of serious injuries to the motor vehicle occupant. Accident data indicate that the highest ranking "body to vehicle" contacts in frontal crashes are the chest and abdomen into the steering column(2,3). The harm associated with steering column contact accounts for over 25% of the total societal costs due to motor vehicle accidents. The injuries which are observed in the accident environment suggest that the forces which are required to compress the EA units frequently are seriously injurious to the unrestrained driver. Until recently, the situation for the restrained driver could not be documented due to the low incidence of belt use. With the advent of state safety belt use laws, however, there appear to be indications that safety belts reduce the harm for restrained occupants, and the injury pattern is shifted from chest and abdomen to head and facial injuries caused by steering wheel impacts(4-6).

Research and compliance testing of steering assemblies produced since implementation of FMVSS 203 and 204 has shown that new columns perform at or near their full EA potential. Accident data analysis(7), however, indicates that the EA units seldom exhibit compression of over four inches—only half or less of their designed capacity for energy absorption. These findings are not inconsistent with the findings of other researchers(8-10), who have found that for steering assemblies which comply with FMVSS 203 and 204, there is no correlation between injury severity and steering assembly performance in both laboratory testing and accident analysis. Although EA units do not appear to be more effective in any particular crash condition, there are indications that they function better in distributed loading frontal crashes than in frontal crashes with concentrated loadings(7).

The cause for the discrepancy between the test results and the accident data is unknown. It has been postulated that crash induced damage to the steering column and/or occupant non-axial loading may be possible causes. Each accident is unique, however, and the column behavior in the varied accident modes is difficult to assess. Since most of the accident data regarding steering column performance is from unrestrained occupants, it is not easy to differentiate the column damage induced by occupant interaction versus structural intrusion effects. A crash matrix conducted by the NHTSA Vehicle Research and Test Center(11,12) provided an opportunity to examine the steering column performance under a variety of frontal crash configurations. This paper describes the extent and nature of steering column intrusion for selected matrix tests, and the effects on both restrained and unrestrained occupants.

Procedures

The complete crash matrix consisted of thirty tests involving car-to-car, car-to-barrier, and car-to-pole impacts in both offset and fully engaged conditions. A total of thirteen tests from the matrix (Table 1) were selected for static analysis of the steering column behavior. Tests with restrained drivers were selected for the analysis to avoid any steering column motion being induced by an unrestrained driver. Of the thirteen tests, nine crash tests in the matrix had passenger data which was not critical to the objectives of the testing. For those tests, photographic analysis was conducted, in addition to the static analysis, by adding a camera to the passenger side of the vehicle (Figure 1), and removing the passenger to avoid obstruction of the steering column camera view.

The onboard steering column camera was run at 1000 frames per second. Within its field of view was a reference point attached to the camera mount to allow documentation of the steering column vertical and rearward displacement relative to the vehicle compartment (Figure 2). Following each test with the steering column camera coverage, the column targets were digitized for graphical presentation of the column displacement. It should be noted that the film analysis was complicated in a couple of the tests due to movement of the reference point. There was also head and sometimes chest contact with the steering column from the restrained drivers, which may have had a slight effect on the column movement in some of the tests.

Each of the vehicles included in the analysis had a series of pre-test measurements taken. Prior to the test, the instrument panel was removed, and a matrix of reference dimensions made. The reference dimen-

Table 1. Tests used for steering column intrusion analysis.

					SUBJECT VEHICLE				CRASH PARTNER			
Test Number	Subject Vehicle	Delta V	Crash Partner	Crash Mode	Driver Dummy	Passenger Dummy	Driver Restraint	Passenger Restraint	Driver Dummy	Passenger Dummy	Driver Restraint	Passenger Restraint
841114	Honda	30	Honda	FF	HIII	P572	UR	UR	P572	HIII	UR	UR
850731	Fuego	30	Fuego	OF	HIII	NONE	R	NA	HIII	HIII	UR	R
850524	Omni	30	Omni	OF	HIII	NONE	R	NA	HIII	HIII	UR	R
850724	Honda	30	Honda	OF	HIII	NONE	R	NA	HIII	HIII	UR	R
850807	Honda	30	Fuego	OF	HIII	HIII	R	UR	HIII	HIII	R	UR
850814	Omni	30	Celebrity	OF	HIII	NONE	R	NA	HIII	NONE	R	NA
850425	Honda	30	Pole	CTR	HIII	HIII	R	UR	NA	NA	NA	NA
850621	Fuego	30	Pole	CTR	HIII	HIII	R	UR	NA	NA	NA	NA
850912	Honda	30	Pole	OF	HIII	NONE	R	NA	NA	NA	NA	NA
851031	Omni	30	Pole	OF	HIII	NONE	R	NA	NA	NA	NA	NA
850910	Fuego	30	Pole	OF	HIII	NONE	R	NA	NA	NA	NA	NA
850918	Fuego	30	FLCB	FF	HIII	NONE	R	NA	NA	NA	NA	NA

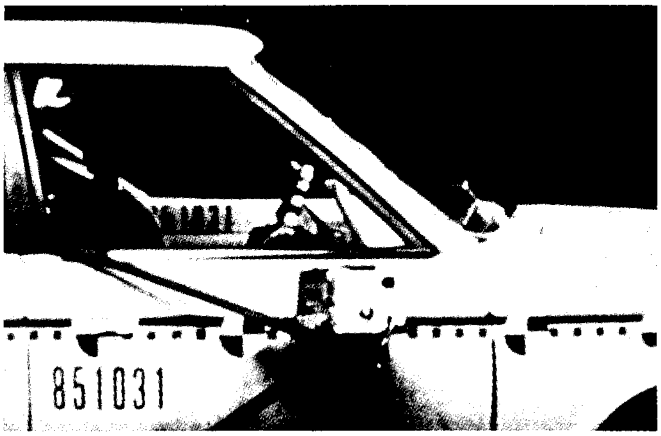

Figure 1. Steering column motion camera placement

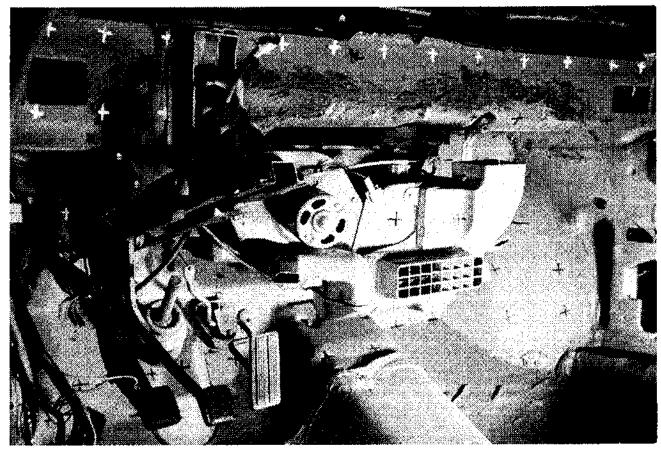

Figure 3. Static intrusion measurement locations

sions were made from either the rear tail light opening or the rear bumper. The points were located at four inch increments from the centerline of the vehicle at five elevations (Figure 3). The top two elevations (A and B) were located on the cowling (where the instrument panel and steering column were attached). The next two levels (C and D) were on the firewall, and the fifth level (E) was near the firewall/toeboard juncture. Following the crash test, these points were again located from the rear reference to determine the amount of rearward displacement.

Results and Discussion

The steering column motion and intrusion measurements are summarized in Figure 4 and Table 2. The horizontal, vertical, and angular dynamic motions of the steering columns are shown in Figure 4 as determined from the photographic film analysis. Although there are considerable differences in the column motions, they all tend to be characterized by a peak horizontal and vertical displacement which occurs between approximately 70 and 120 msec (roughly the time at which contact with an occupant would occur), and levels out to a residual displacement by 200 msec. The peak dynamic and residual (taken at 250 msec) displacements are summarized in Table 2, along with the peak static intrusion measurement (regardless of lateral position) from the five measurement levels.

Several observations are apparent from the data summarized in Figure 4 and Table 2:

- The importance of measuring steering column motion dynamically was clearly demon-

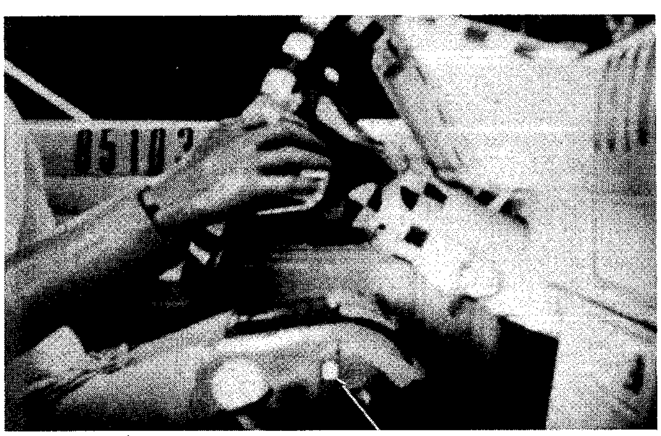

Figure 2. Steering column film analysis camera view

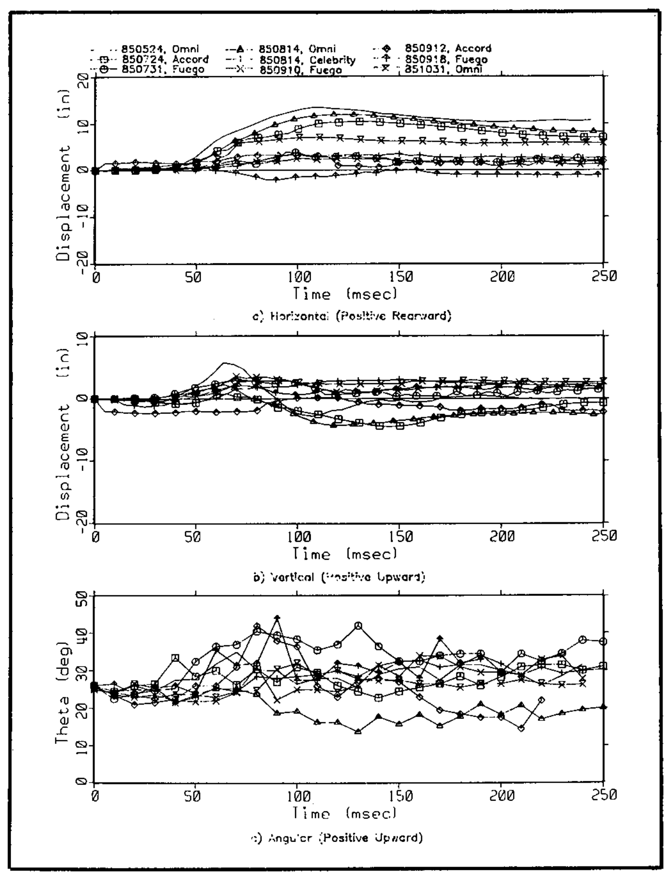

Figure 4. Steering column movement

Table 2. Steering column and intrusion analysis summary

Vehicle	Test No.	Test Configuration	Steering Column Film Analysis				Peak Static Measurement Level					SC Displacement Ratio (Residual/Dynamic)	
			Peak X	Dynamic Z	Residual X	Z	A	B	C	D	E	X	Z
Celebrity	850814	Omni/Celebrity OF	3.6	3.2	2.5	2.3	3.75	2.50	3.75	7.13	6.75	0.69	0.72
Omni	850814	Omni/Celebrity OF	11.9	-4.2	8.2	-2.6	7.00	8.80	13.25	11.62	8.78	0.69	0.62
Omni	850524	Omni/Omni OF	13.4	5.7	10.6	0.0	7.25	9.43	14.98	14.07	12.93	0.79	0.00
Omni	851031	Omni/Pole OF	7.0	2.8	5.8	2.6	4.63	6.55	10.02	12.06	12.86	0.83	0.93
Honda	850807	Honda/Fuego OF	NA	NA	NA	NA	13.63	13.88	18.06	17.25	16.50	----	----
Honda	841114	Honda/Honda FF	NA	NA	NA	NA	1.88	2.94	4.63	6.50	8.25	----	----
Honda	850724	Honda/Honda OF	10.4	-4.5	7.0	-0.7	7.63	7.88	11.00	11.50	12.25	0.67	0.16
Honda	850425	Honda/Pole CTR	NA	NA	NA	NA	2.50	2.75	7.56	8.44	10.00	----	----
Honda	850912	Honda/Pole OF	3.4	-2.3	1.8	-1.9	7.25	7.06	9.13	9.50	10.25	0.53	0.82
Fuego	850918	Fuego/Barrier FF	-2.1	2.1	-1.1	1.6	4.81	3.57	4.19	5.94	4.69	0.52	0.76
Fuego	850731	Fuego/Fuego OF	3.9	2.9	2.5	1.5	3.56	3.70	5.50	5.12	6.25	0.64	0.52
Fuego	850621	Fuego/Pole CTR	NA	NA	NA	NA	2.19	3.26	4.69	8.75	7.81	----	----
Fuego	850910	Fuego/Pole OF	2.5	3.5	1.3	1.9	3.44	3.81	4.69	6.44	6.32	0.52	0.54

strated. Note, for example, that the Omni/Omni OF test had the largest amount of dynamic upward motion, but zero residual.

- For the horizontal steering column motion, the residual position appears to be 50 - 80% of the peak dynamic motion. For the vertical motion, there does not appear to be a correlation between the peak dynamic and the residual motion.
- The horizontal movement of the steering column was rearward in all cases except the Fuego/FRB test. The forward movement in the Fixed Rigid Barrier test was the result of column rotation.
- The vertical steering column motion was predominately upward; four columns, however, moved downward significantly (two of those four also experienced upward movement). The maximum upward movement was less than 6 inches.
- The steering column rotation was predominately upward. The upward angular motion was generally less than 10°, although in three tests the rotation was 15 - 20° during the time interval that occupant contact would be expected to occur.
- The cowl intrusion (levels A and B) correlated well with the horizontal residual steering column movement (illustrated in Figure 5). This is not surprising since the column mounts to the cowl for the vehicles included in this analysis. The Honda/Pole OF test and the Fuego/Barrier test (in which the column rotated as discussed above) were exceptions to the correlation.

The static and dynamic results of Figure 4 and Table 2 indicate that the steering column can have considerably different responses, depending upon the vehicle and the crash configuration. The vehicle influence on the steering column response is illustrated in Figures 6 and 7. In both the small car/car offset tests and the small car/pole offset tests, the Omni steering column horizontal movement was greater than either the Accord or Fuego. For the horizontal motion, the Omni and Fuego represent performance extremes in the offset crash configuration for those vehicles in-

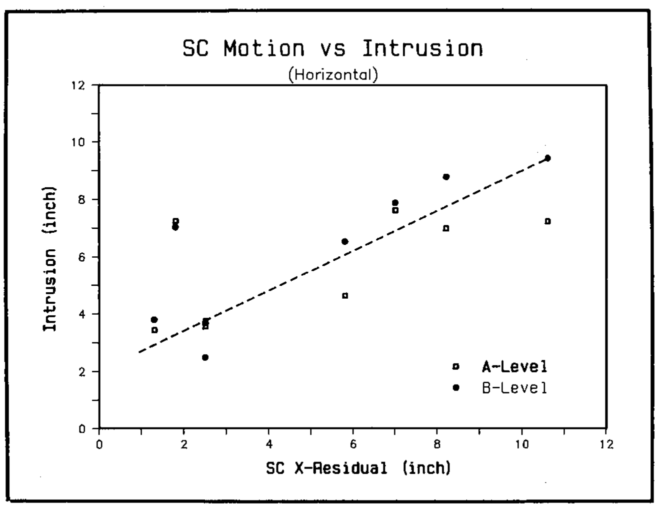

Figure 5. Residual steering column motion correlation to cowl intrusion measurement

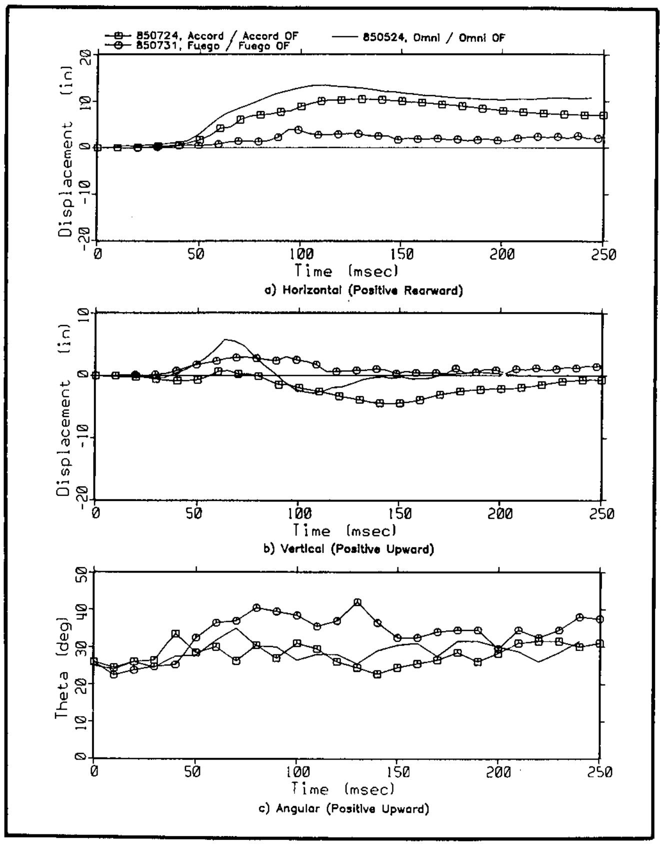

Figure 6. Car/car offset test steering column motion

cluded in the analysis. Although vehicle stiffness and crash configuration were considered in the test matrix development, it is possible that even greater performance differences could occur in the accident environment.

The crash configuration influence on the steering column response is illustrated for each of the small vehicles in Figures 8 - 10. Note, for example, that the car-to-car offset tests generally had a greater amount of steering column motion than the corresponding car-to-pole offset tests. The Omni vertical column movement was apparently quite dependent upon the struck object. In the Omni/Celebrity OF test, the column movement was the static downward extreme and nearly the dynamic downward extreme (Table 2); whereas, the upward extremes occurred for the Omni/Omni OF dynamically and the Omni/Pole OF statically.

A detailed analysis of the column mountings and design features was not included in this study. In a susbequent 30° oblique fixed rigid barrier crash test program with a VW Golf, however, the steering column was observed to have very little movement. Although the lack of column movement could have been due to the different crash configuration, it may also have been by design. The Omni and Fuego steering columns mount to the cowl with four bolts

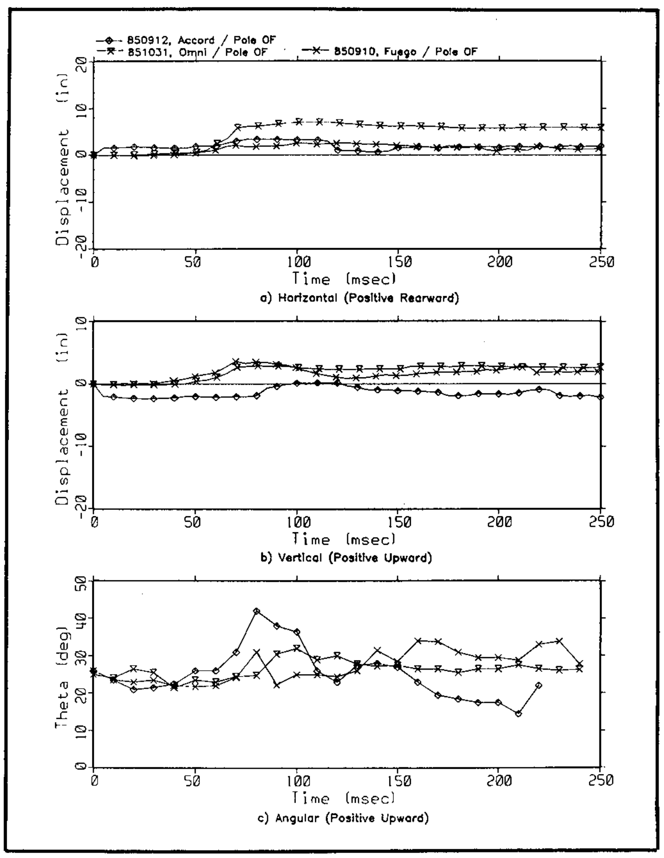

Figure 7. Car/pole offset test steering column motion

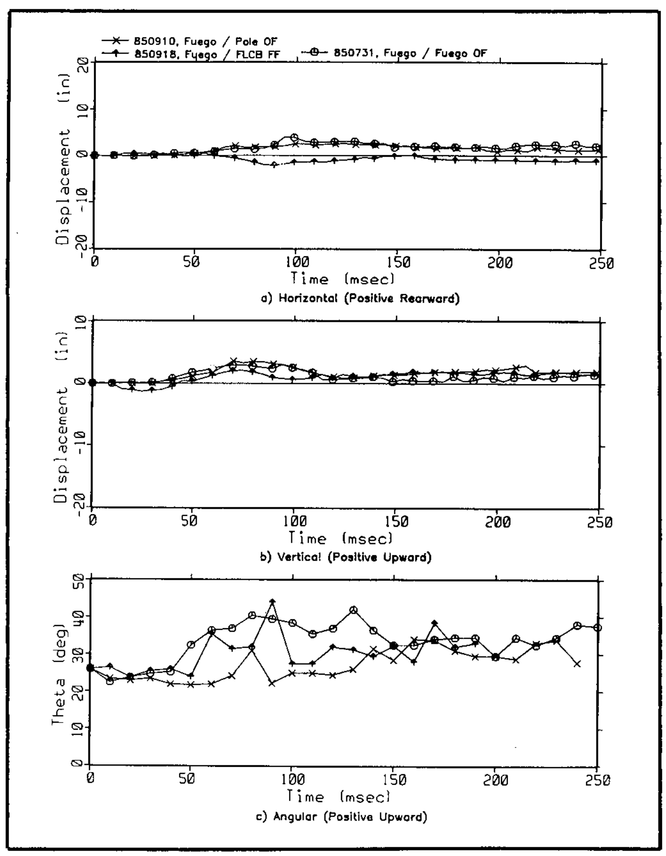

Figure 8. Fuego steering column motion

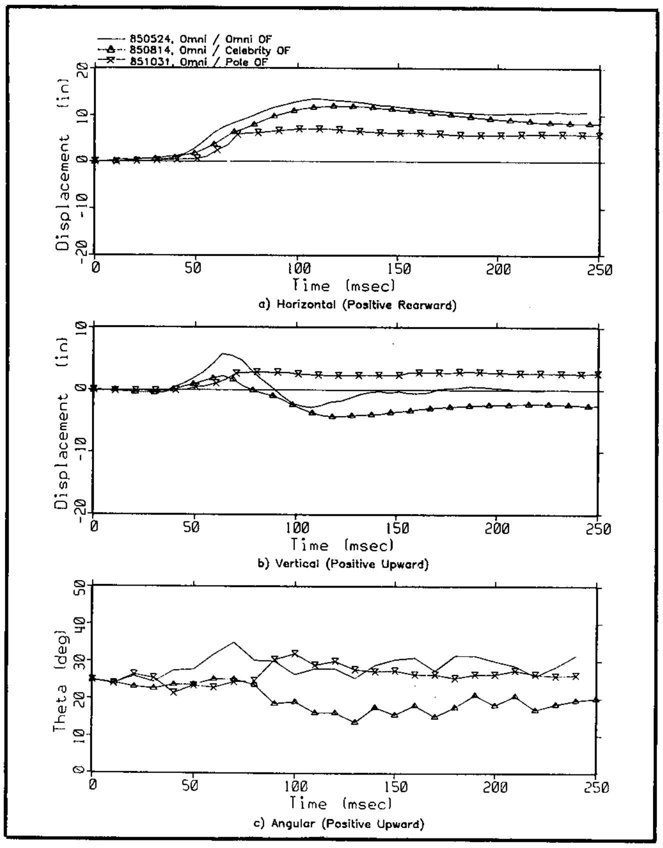

Figure 9. Omni steering column motion

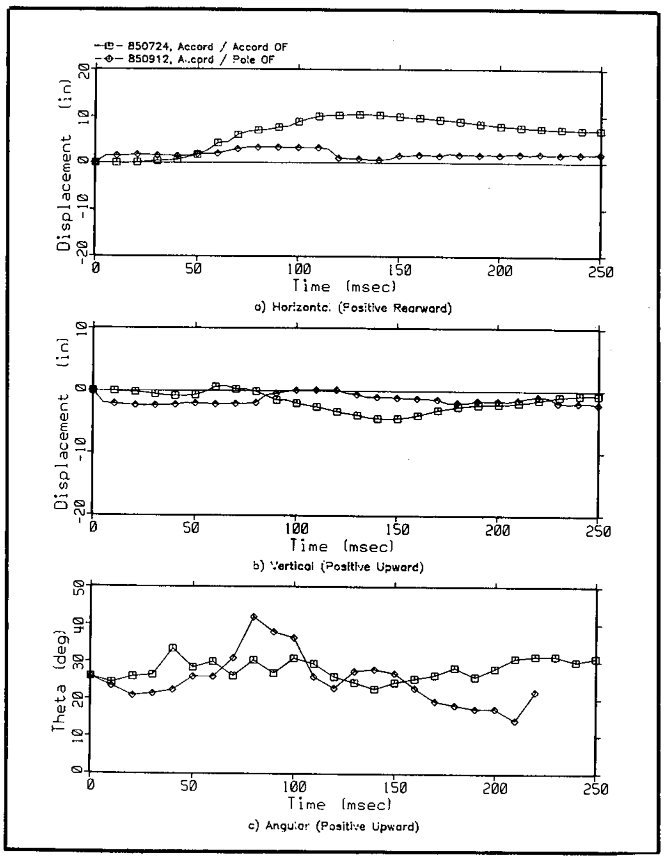

Figure 10. Accord steering column motion

(Figure 11 a, b). The Golf and Accord are mounted with only two bolts at the cowl (Figure 11 c, d). The lower end of the Golf column, and possibly the Accord column, appears to have been designed to allow rearward intrusion of the firewall relative to the cowl mounting, without movement of the steering wheel. In the Golf oblique barrier test, the relative movement was absorbed through deformation of the column tube webbing (Figure 11 d). In the Accord, it appears as though a nylon bushing would permit such movement. Further investigation and evaluation of these designs would be necessary to better understand their function. This cursory analysis, however, appears to indicate that limitation of the column motion is feasible in production steering column designs.

The ranges of dynamic steering column motion observed in these tests, and the vehicle and crash configuration influence on steering column response may help to explain the apparent lack of correlation between steering column performance in laboratory and compliance test results, and accident data. It also appears feasible to limit the steering column motion.

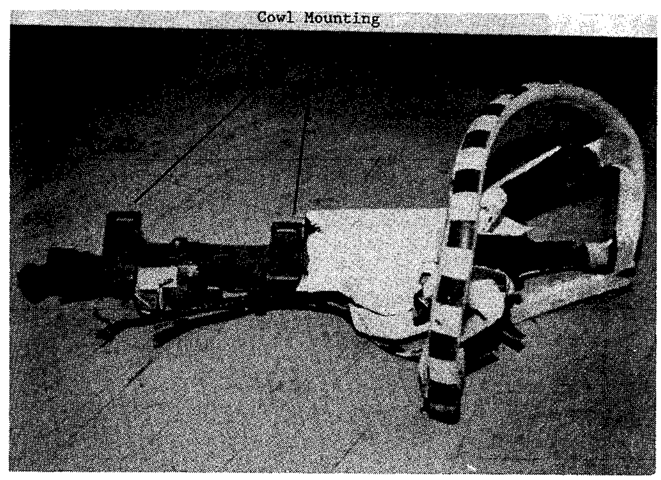

Figure 11a. Omni steering column designs

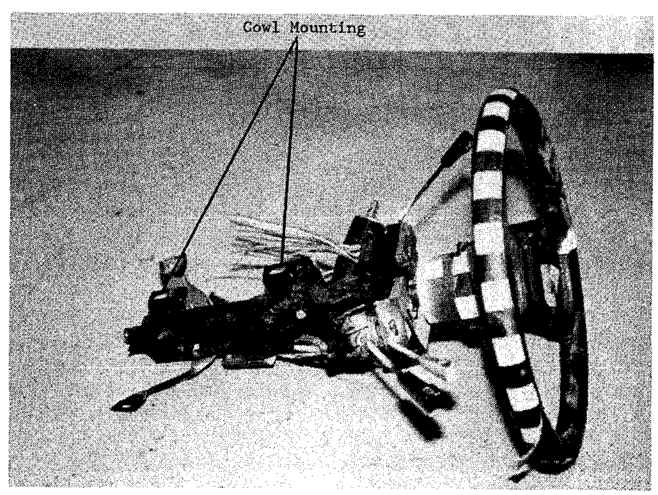

Figure 11b. Fuego steering column designs

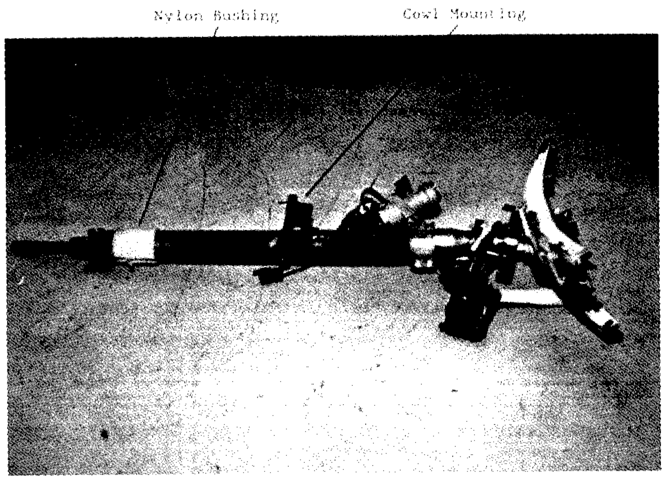

Figure 11c. Accord steering column designs

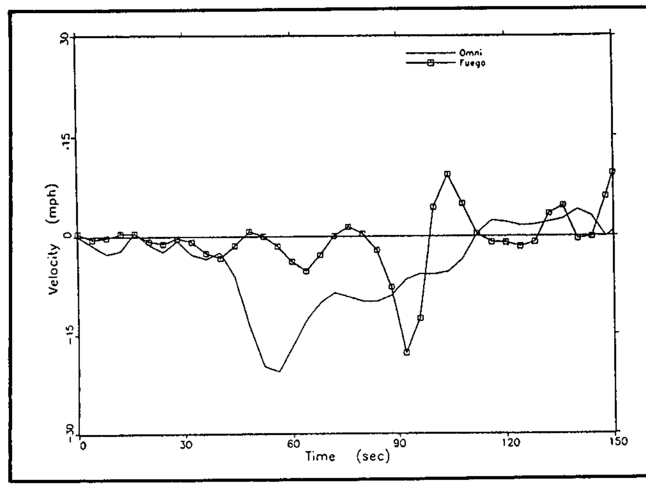

Figure 12. Steering column intrusion velocities for Omni (850524) and Fuego (850731)

Contact Velocity Calculations

Reduced compartment available internal distance due to intrusion, compartment geometry, vehicle crash pulse, and steering column dynamic motion all interrelate to determine the contact location and velocity for both the restrained and unrestrained crash occupants. Estimates of steering column contact velocities due to intrusion are made in this section.

Unrestrained Occupant Effects—For the unrestrained occupant, the horizontal intrusion would primarily affect the chest contact velocity, and the vertical intrusion would influence the vertical impact location. Steering column velocities relative to the compartment were obtained from the horizontal displacement-time histories of Figure 4 for the Omni (850524) and Fuego (850731) since they represent extreme cases for a given crash configuration, and are shown in Figure 12. Using these velocities, and neglecting occupant impact and compartment geometry effects, one-dimensional velocity-time diagrams were developed for both the Omni and Fuego (Figures 13 and 14).

The left compartment acceleration and the steering column velocities of Figure 12 were combined to produce these horizontal velocity-time diagrams. The initial dimensions from the dummy chest to the steering wheel hub was 14-15 inches for the Omni and Fuego. Using a 15 inch dimension, the contact velocity due to the horizontal column intrusion was estimated assuming that the occupant trajectory would have been directly forward. From Figures 13 and 14, the Omni and Fuego crash pulse differences alone would have had little effect on the unrestrained occupant chest contact velocity. The contact velocity ($\triangle V_1$) for the unrestrained occupant without intrusion would have been approximately 20 $\frac{1}{2}$ mph for both tests. With the steering column intrusion taken into consideration, the unrestrained occupant chest contact velocity ($\triangle V_2$) would have been 24 mph occurring at 70 msec for the Omni, and 30 mph occurring at 90 msec for the Fuego.

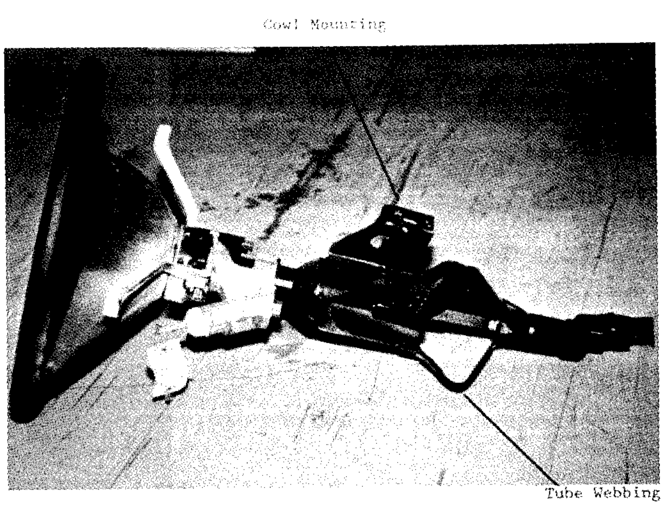

Figure 11d. Golf steering column designs

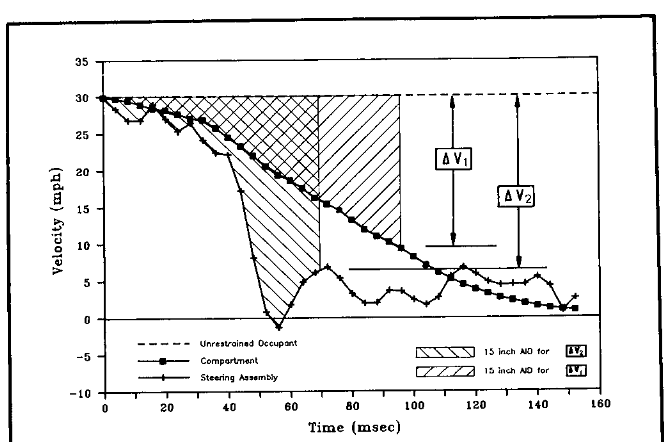

Figure 13. Omni (850524) velocity-time diagram for unrestrained occupant

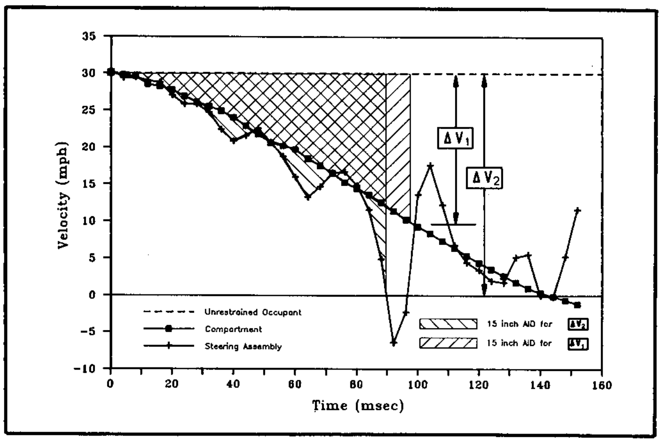

Figure 14. Fuego (850731) velocity-time diagram for unrestrained occupant

These extremes in steering column motion would indicate that greater intrusion of the steering column into the occupant compartment (13.4 inches for the Omni versus 3.9 inches for the Fuego) does not necessarily lead to higher unrestrained occupant contact velocities. Figures 13 and 14 illustrate that the lower contact velocity for the Omni steering column intrusion is a result of the intrusion occurring earlier in the crash event than in the Fuego test. The Fuego intrusion began at approximately 80 msec, whereas the Omni intrusion began at 50 msec leading to the reduced contact velocity and greater structural ride-down for the unrestrained occupant. This one-dimensional velocity-time analysis would indicate that the time of occurrence of steering column intrusion is a greater determinant of unrestrained occupant chest contact velocity than the amount of intrusion. It also illustrates the sensitivity of the intrusion timing. For example, if the unrestrained Fuego and Omni occupants had been a few inches out-of-position (closer to the steering column) at the time of the crash, the results could have been strongly reversed. That is, the occupant chest contact velocity could easily have been 30 mph (instead of 24) in the Omni, and 15 mph (instead of 30) in the Fuego.

Restrained Occupant Effect—For the restrained occupant, the vertical intrusion would most likely affect the head contact location and velocity; the horizontal intrusion would affect head contact location and, to a lesser extent, contact velocity. Quantitative determination of steering column intrusion effects on the contact velocity of the restrained occupant is more difficult than for the unrestrained occupant due to the greater complexity and interdependence of the restraint system, crash pulse, compartment geometry, occupant anthropometry, and consequent occupant kinematics. The steering column movement and time of motion are equally, or perhaps more critical to the occurrence of restrained occupant contact than for the unrestrained occupant.

Since the drivers from the tests of this analysis were all restrained, the effects of steering column motion on restrained occupants could be determined directly. As was indicated in Figure 4 and Table 2, both the horizontal and vertical movement of the column was dependent upon the vehicle and crash configuration. Film analysis was conducted to determine the time and location of the restrained driver contact with the steering assembly, and the column velocity was calculated with results as summarized in Table 3. Several points are apparent from Table 3.

- The steering wheel rim was the predominate steering assembly contact location.
- The driver head and face always made contact with the steering assembly. It should be noted that in a few cases the lower rim was also contacted by the chest or abdomen prior to the head/facial impact.
- The column velocity at time of occupant head/facial contact was substantially lower than the peak velocity for each case.

As was the case for the unrestrained occupant, these results indicate the higher contact velocities could have resulted if there were slight variations in any of the conditions. They also indicate that time occurrence of the column motion is critical to the restrained occupant contact.

Conclusions

The steering column performance from thirteen 30 mph ΔV crash tests of Celebrity, Omni, Accord, and Fuego vehicles was analyzed using static measurements, photographic film analysis, a simple, one-dimensional model, and dummy responses. It is noted that these vehicle steering assemblies comply with FMVSS 203 and 204. The procedures described in the analysis provide a methodology for documenting intrusion and steering column dynamic performance in crash testing. In addition, a limited post-test comparison of the Omni, Accord, and Fuego columns was made with that of a VW Golf which had experienced a 30 mph ΔV oblique barrier collision. The following conclusions were made from the analysis:

- The importance of dynamically measuring steering column motion was clearly demonstrated. Significant differences between dynamic motions and residual displacements measured post-test were observed.
- Although the magnitude of steering column intrusion into the occupant compartment did not correlate with driver contact velocity, the steering column intrusion timing was found to be critical to determining occupant contact velocity for both restrained and unrestrained occupants.

Table 3. SC velocity effects on restrained occupants.

Vehicle	Test Number	Test Configuration	Contact Location: Steering Assembly	Contact Location: Driver	Contact Time (msec)	Column Velocity (mph) At Contact X	At Contact Z	Peak X	Peak Z
Celebrity	850814	Omni/Celebrity OF	Upper Rim, Hub	Forehead, Face	109	-1.3	2.0	5.8	6.3
Omni	850814	Omni/Celebrity OF	Upper Rim	Nose, R. Cheek	82	9.2	-10.4	15.2	-11.1
Omni	850524	Omni/Omni OF	Upper Rim	Bridge of Nose	78	10.0	-15.2	13.6	-18.1
Omni	851031	Omni/Pole OF	Upper Rim	Bridge of Nose, Eyes	82	2.7	0.6	15.7	6.9
Honda	850807	Honda/Fuego OF	NA	NA	--	NA	NA	NA	NA
Honda	841114	Honda/Honda FF	NA	NA	--	NA	NA	NA	NA
Honda	850724	Honda/Honda OF	Upper Rim	Forehead	94	6.1	-2.4	9.8	-12.1
Honda	850425	Honda/Pole CTR	NA	NA	--	NA	NA	NA	NA
Honda	850912	Honda/Pole OF	Upper Rim	R. Cheek	87	-0.2	6.0	10.3	-13.6
Fuego	850918	Fuego/Barrier FF	Upper Rim, Hub	Nose, Forehead	87	-1.1	-4.2	-5.1	5.6
Fuego	850731	Fuego/Fuego OF	Left Rim	Nose, Mouth, Forehead	100	-0.9	-3.7	8.5	-6.7
Fuego	850621	Fuego/Pole CTR	NA	NA	--	NA	NA	NA	NA
Fuego	850910	Fuego/Pole OF	Left Rim, Hub	R. Cheek, Eye, Nose	95	-2.4	-4.3	6.7	5.4

- Unrestrained driver contact velocities estimated both with and without steering column intrusion indicated that considerably higher contact velocities can occur with intrusion.
- Post-test examination of the steering assemblies appeared to indicate that limitation of column motion due to intrusion is feasible in production steering column designs.
- The column performance was dependent upon not only the vehicle, but also the crash configuration.
- The residual rearward position of the steering column was 50-80% of the peak dynamic displacement. There was no correlation for the vertical displacement.
- The maximum upward movement of the column was less than six inches.
- The horizontal movement of the steering column was rearward in all cases except the Fuego/FRB test. The forward movement in the Fixed Rigid Barrier test was the result of column rotation.

The dependence of the steering column dynamic performance on the crash configuration, and the importance the column intrusion time occurrence has in determining the driver contact velocity may well be significant factors in explaining the discrepancy between test results and accident data. Application of the static measurement and photographic film analysis methodologies utilized in this study to other production vehicles and crash configurations are needed to better characterize steering column intrusion and its effect on occupant contact velocities.

Acknowledgements

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Reliability Considerations in the Design of an Air Bag System

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Abstract

An air bag may offer additional protection for vehicle occupants when used as a supplemental restraint system in combination with a seat belt system. Despite this important function, there is no way of demonstrating air bag readiness short of actually deploying it because it can be actuated only once. Therefore the air bag should be inherently highly reliable.

To help obtain high system reliability, an analysis was performed to evaluate the risk of the failure modes in each component and subsystem, and appropriate provisions were made for those possibilities. Then a set of high-reliability design concepts was established by looking at the common elements of all the improvements.

This paper describes the high-reliability design concept and some considerations in designing the air bag system. Then it discusses the features of an improved air bag based on these design efforts.

Introduction

An air bag system cannot provide satisfactory occupant protection when used by itself in all crash situations; however, if it is used as a supplemental restraint system, in combination with a seat belt, the air bag may offer additional protection for vehicle occupants in certain crash modes.

This restraint system potentially has two failure modes. One would be inadvertent deployment of the bag while in normal conditions. The second would be the failure of the bag to deploy when it should in crash.

Unlike other vehicle systems, we cannot verify the serviceability of the system by checking it before starting the vehicle because it is a so-called "one-shot device" which is actuated only once when necessary. It is not designed for repeated use. This one time operation makes it difficult to find system failures. For this reason, the air bag system has been designed to be inherently highly reliable.

High-Reliability Concept

To achieve high reliability of the air bag system, we first concentrated on developing several types of system and component constructions. Their reliability was then analyzed by such methods as the FMEA (failure mode and effects analysis) and the FTA (fault tree analysis). Furthermore based on these analyses, and the reliability technology used in the aerospace field, suitable remedies were provided for the automobile air bag system. Thus, the system and components were continuously improved by trial and error.

During this research and development process, we found some points obviously common to all remedies which logically led to the development of three high-reliability concepts:

Redundant Design

For relatively simple components, use of redundancy can be more effective than a complex design in improving the reliability of the system as a whole.

Simplified Construction

As the system or occupants become more complex, a combination of more than one fault is more likely to occur; therefore we can achieve higher reliability by making them as simple as is technically feasible.

Improvement of Major Components

The reliability of the air bag system as a whole can be efficiently increased by developing more reliable components for the main electric circuit which plays an important role in actuating the system when necessary.

Some Considerations in Designing the High Reliability System

Diagnostic Circuitry and Redundancy

A redundant design is one of the basic techniques to improve system reliability. In applying the redundancy to the air bag system, however, some components are physically difficult to duplicate such as placing two inflators in a steering wheel.

Instead of duplicating the same components, reliability can be achieved by a different setup. One way is to add an electrical diagnostic circuit which diagnoses the air bag system. This is an effective way to achieve a high reliability design.

However, since a diagnostic circuit is not foolproof, an evaluation was made for the following cases:

1. The air bag itself malfunctions because of the additional diagnostic circuit.
2. A system malfunction occurs simultaneously when a system failure is diagnosed.
3. A mechanical malfunction occurs which cannot be diagnose delectrically.

Construction of Crash Sensors and Inflator

The Honda air bag system uses four crash sensors. Its reliability depends mainly on how these sensors are connected to the inflator.

For instance, the position of the inflator in the circuit is closely related to the incidence of the air bag's inadvertent deployment (See Fig. 1). In circuit (a), a grounding short circuit on the negative voltage side of the inflator results inadvertent deployment of the air bag. While in circuit (b), a power short circuit on the positive voltage side of the inflator leads to such inadvertent actuation of the system. In circuit (c), however, the air bag does not deploy inadvertently unless there are two simultaneous short circuits: one on the power side of the inflator, and the other on the grounding site. By using the third circuit, there is a reduced likelihood of inadvertent system actuation in comparison with the other two circuits.

Now let us look at the relationship between the layout of the sensors and the probability of a system failure to deploy the air bag when necessary (See Fig. 2). The front sensors, installed outside the vehicle compartment, have long wire harnesses which may make them more liable than cowl sensors to a grounding short circuit. In circuit (d), a grounding short circuit of the front sensors results in a system failure to deploy the air bag; but, in circuit (e), it is not prone to such a failure.

Thus, the circuit configuration requires a thoroughgoing study because the system's reliability depends largely on the electric circuit design. Needless to say, we had to predict the incidence of different failure modes and then compare them quantitatively before making a final decision on circuit configurations.

System Failure at Crash

The condition of the vehicle can change catastrophically during a crash possibly resulting in an air bag system failure in some instances. Table 1 shows system components which may be affected by crash dynamics.

To eliminate the probability of the power unit becoming disconnected in a crash, we have installed a

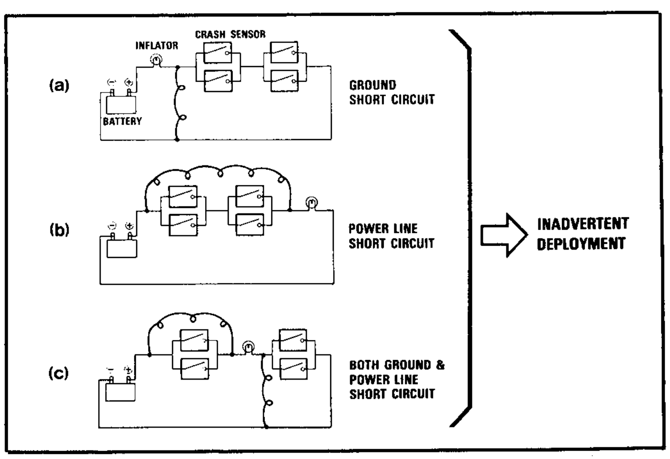

Figure 1. Inflator location and inadvertent deployment failure

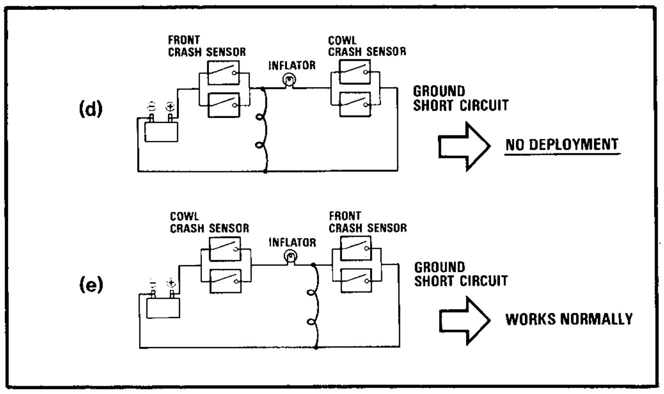

Figure 2. Sensor layout and no deployment failure

capacitor in the air bag system as a back-up for the power supply. Also to make the connector more reliable than a slip ring type, we have decided to use a direct connected cable reel design instead.

Construction and Operation of the Honda Air Bag System

Fig. 3 shows the system components and their layout. There are four crash sensors—one on each side of the foremost end of the front side frame and two cowl sensors in the main and diagnostic circuits in the vehicle compartment on the front end of the center tunnel.

As shown in Fig. 2(e), the four sensors are connected in parallel-series so that when at least one of the front and the cowl sensors is actuated simultaneously, an electric signal is sent to the inflator. This electric signal is transmitted through a separate independent main harness used only for the air bag system and then through a directly-coupled cable reel to the inflator.

When actuated the inflator generates nitrogen gases which inflate the air bag. The inflator and the air bag are attached to the retainer. During inflation, the air bag breaks through the thinner part of the module's cover fastened to the retainer, and further deploys toward the driver's head from the steering wheel.

The actuation of the sensors takes some 10-15 msec in a head-on crash against a barrier at 30 mph.

Table 1. Possible failure at crash

Failure Mode	Affected Component	Major Cause
Loss of Power Supply	Battery Power Unit Harness	Disconnected Terminal Broken Harness
Loss of Sensor Signal	Sensor Harness	Broken Harness Loosened Connector
Loss of Electrical Signal to the Inflator	Inflator Harness Slip Ring	Loosened Connector Corrosion, Chattering

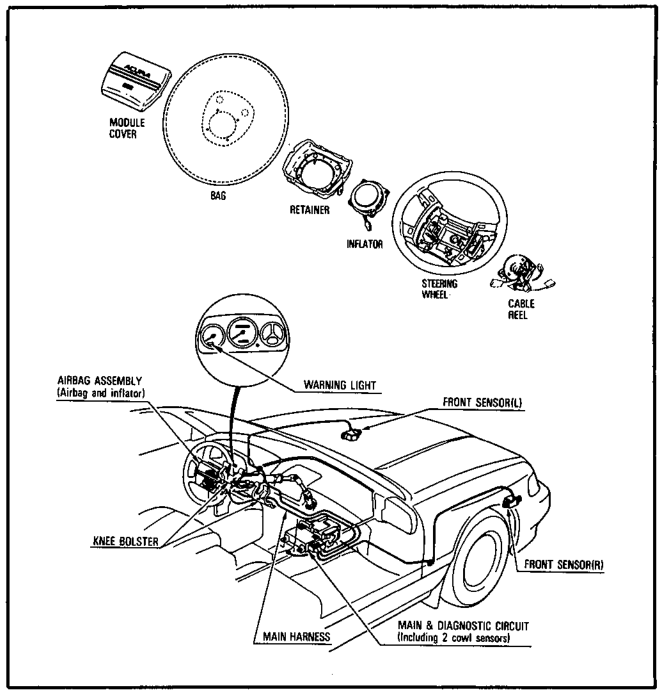

Figure 3. Airbag system construction

Subsequent operation of the system takes approximately 30 msec, after the sensor actuation and until the air bag is fully inflated.

High-Reliability Specifications for System Components

Crash Sensor

Basically the sensor consists of a sensing unit and a housing. The sensing unit (Fig. 4) uses the rolamite sensing technique: its contacts are gold-plated to provide more lasting electrical contact with less resistance over the life of the system. Joints of its components, including those parts where electrical continuity is required, are also spot-welded to increase their reliability. In addition, the band spring in the crash sensor always has electrical current flowing through it so that any problem may be detected immediately. And the sensing unit is put in a can

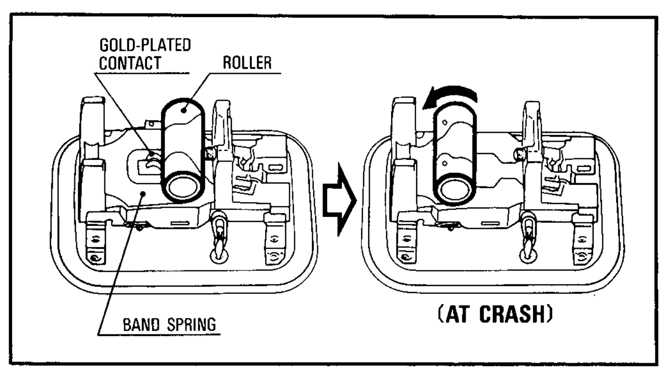

Figure 4. Crash sensor (sensing unit)

filled with inert gas which is made airtight by a hermetic seal.

The housing (Fig. 5) protects the front crash sensor from environmental damage such as water, dust, or other foreign substances. To manufacture this housing, silicone potting is applied to the conductor pin on the outside of the can while the can as a whole is further protected by epoxy potting.

For transmitting electric signals from the sensor, the system uses a cabtyre cable sheathed in corrugated tubing which directly connects the sensor to the main and diagnostic circuits without any intermediate connectors and which minimizes the number of joints and thus provides higher reliability.

Component Transmitting the Electric Signal

The connector (Fig. 6) providing electrical links between different system components uses the inertia lock method, and the terminals (Fig. 7) are set in the connector by a double lock. To reduce the stress on the leaf while the terminals are in contact, two leaves are provided for each female terminal, and their contact surfaces are gold-plated to increase the durability of electrical contact.

To achieve higher reliability, the connector is given a special feature to permit production line quality control checks. This feature is a series of ridges and grooves like a bar code on both the male and the female connector (Fig. 6). After joining the connector, the quality control inspector examines the ridge and groove pattern to ensure the fit of the connector (Fig. 8) and informs assigned workers of the connector fit quality. This method ensures that only the connectors with perfect fit will be produced.

The cable reel (Fig. 9) transmits electric signals from a stationary part to a rotary assembly. To achieve higher reliability of circuit continuity, this component uses a spiral of directly connected flat cable housed in a special case. The two ends of the

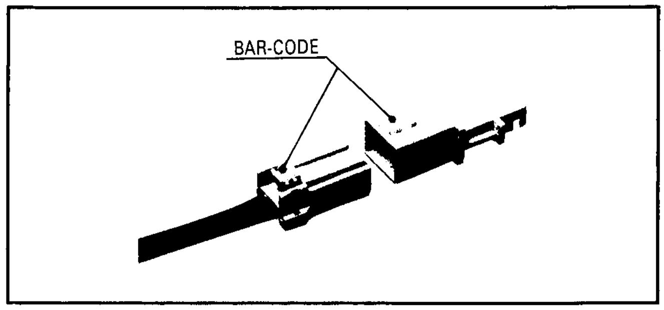

Figure 6. Bar-coded connector

cable are pulled out of the case and directly connected to the terminals by spot welding.

All wire harnesses are yellow to easily distinguish them on the assembly line, and all harness fixing clips are blue for easy inspection because of the color contrast.

Inflator

The inflator uses a propellant as described in Fig. 10. When ignited, it can inflate a 60-liter air bag in approximately 30 msec.

The system performance is significantly affected by excessive or insufficient propellant, by a missing filter, or by any other irregularities in the system fabrication. To preclude these problems, the weight of the inflator assembly is carefully checked by a computer system which rejects any assembly if the gross weight differs from the combined total weight of its components.

If the pyrotechnic unit is not loaded properly, the electric initiator may fail to work satisfactorily when receiving an electric signal from the sensors. All initiators are therefore closely examined by a radioscopic system under a strict quality control program.

Bag

The air bag (Fig. 11) itself is made of a nylon 66 fabric, the inside of which is coated with chloroprene

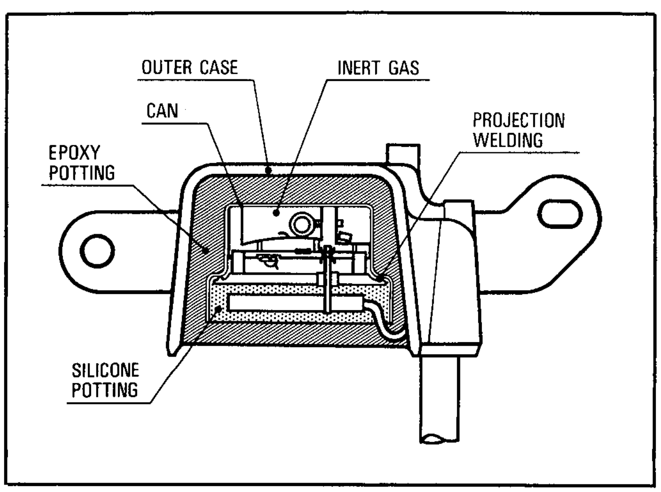

Figure 5. Front crash sensor (sensing unit & housing)

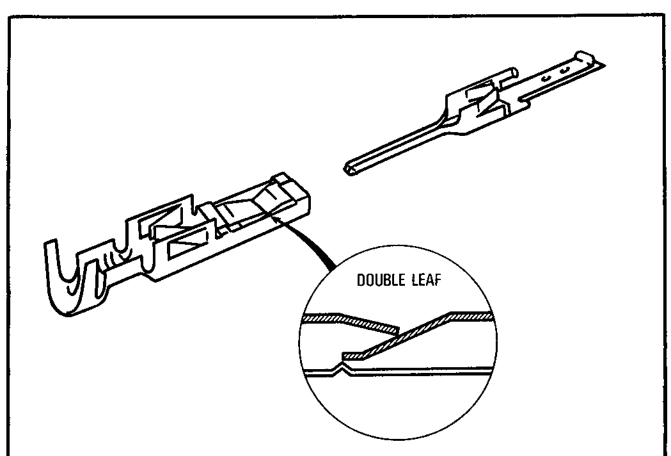

Figure 7. Gold plated connector terminal

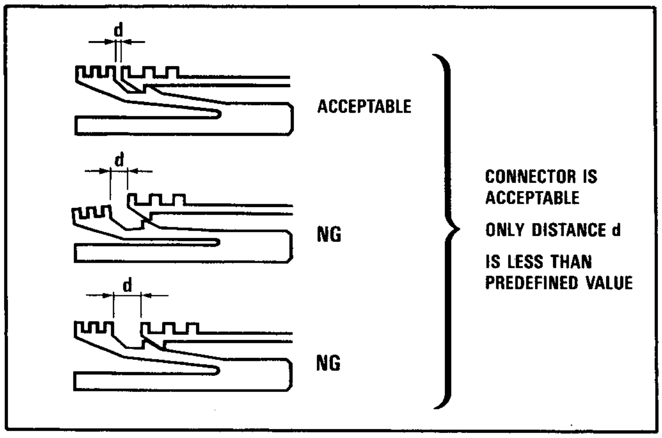

Figure 8. Bar-code connector to check the complete insertion

rubber. For higher reliability, the bag is double stitched.

The module cover (Fig. 12) is a urethane foam with an inserted polyester netting which prevents unnecessary tearing or bursting of the cover in any other part than the top center through which the bag deploys. The netting is fastened to the retainer with steel clamps.

Main and Diagnostic Circuit

The main unit (Fig. 13) has built-in cowl sensors to ensure that the system reliability will not be affected by an increase in the joints of cables and other components. It also features two other devices: one is a booster circuit which keeps the voltage stable during a fall in battery voltage; and the other is a backup power unit that supplies electricity during a crash if the battery becomes unserviceable because of the impact of the accident.

The diagnostic unit (Fig. 13) is separated from the main unit to avoid affecting it. This diagnostic circuit, consisting of widely-used elements now on the market, monitors the circuits of the air bag system and warns of malfunctions, if any, by an indicator on the instrument panel. The elements of the monitor unit are designed in such a manner that they can be easily attached to the printed circuit board by an automatic soldering system.

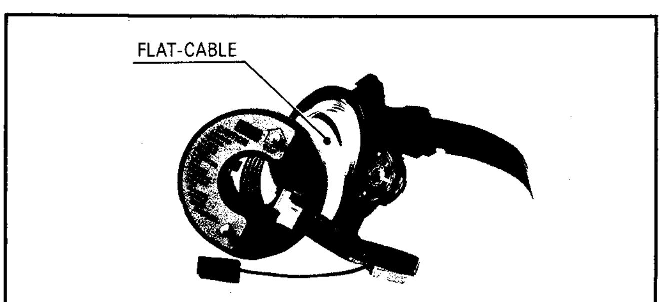

Figure 9. Cable reel

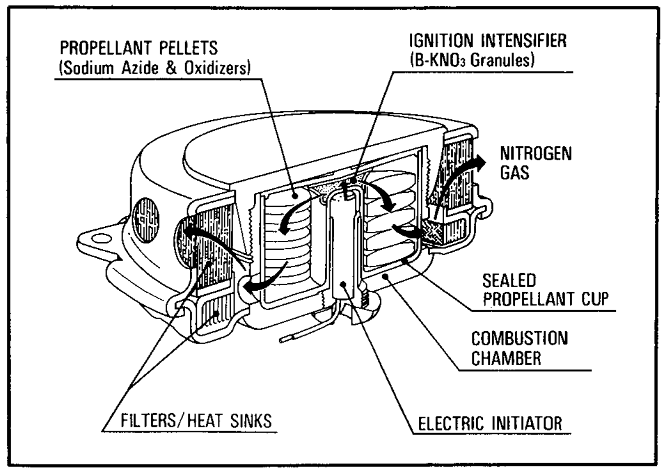

Figure 10. Inflator

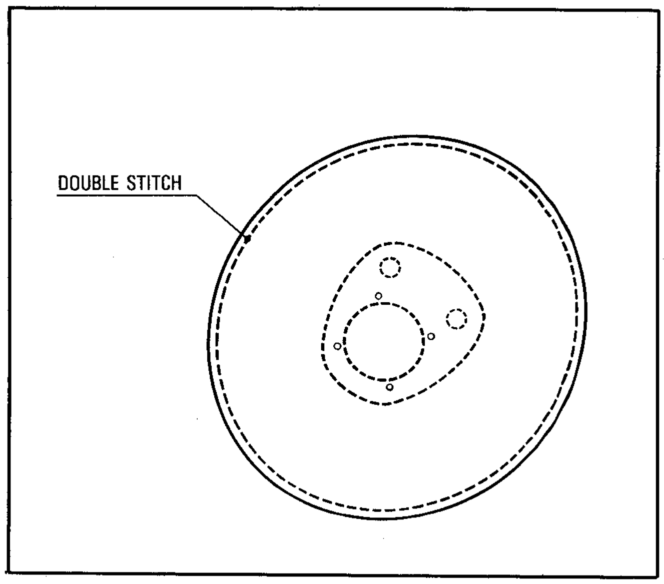

Figure 11. Bag

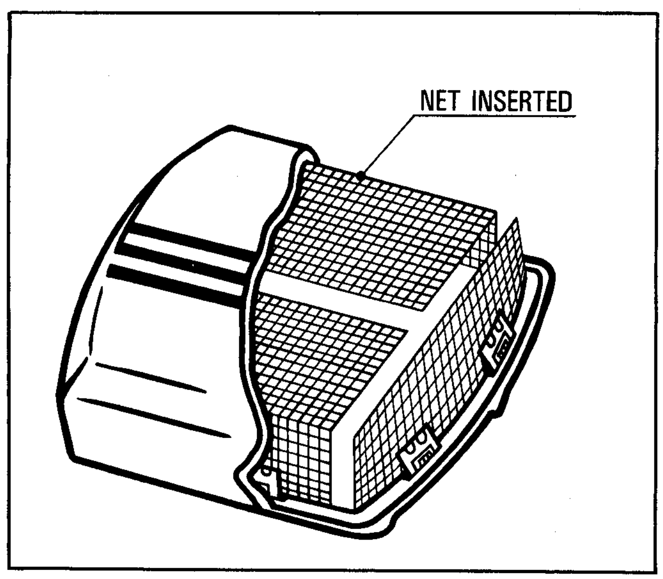

Figure 12. Airbag module cover

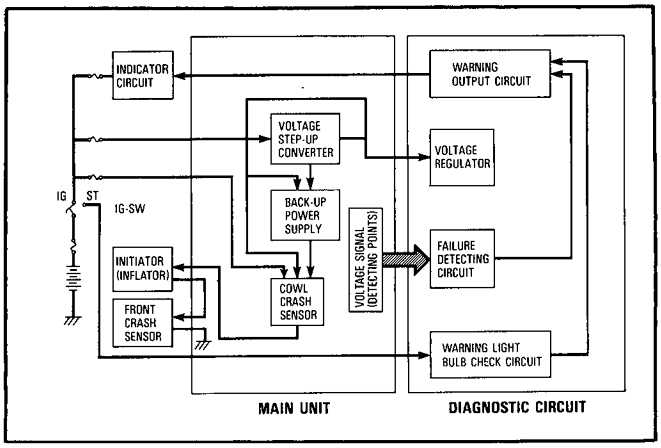

Figure 13. Main and diagnostic circuit block diagram

The foregoing specifications for system reliability are summarized in Table 2.

Table 2. High-reliability specifications for system components.

Component	High-reliability construction
Sensor	1. Dual circuit redundancy design with two each of front and cowl sensors in parallel-series connection. 2. Durable gold-plated contacts. 3. Anticorrosive hermetic seal with inert gas in the sensor case. 4. Airtight epoxy potting covering the front sensors entirely. 5. Continuous checkups on the band spring to find any damage by always flowing an electric current through them.
Harness	1. Reliable double cover.
Connector	1. Durable gold-plated contacts. 2. Installation of double locks in some parts. 3. Bar-code connector for automatically finding any errors in the assembling process.
Cable reel	1. Directly-connected spiral cable.
Diagnostic circuit	1. Back-up power unit to preclude a power failure during a crash. 2. Voltage step-up converter.

Conclusion

This paper has discussed the design and construction of the Honda high-reliability air bag system. In the future it is expected that many air bag systems will be introduced to the market by automobile manufacturers. Based on experience, further research projects could be carried out to develop further system refinements.

The Honda air bag is a supplementary restraint system, which by itself cannot offer complete or total occupant protection in all crash modes. It is intended to be used in combination with a seat belt.

Only limited information is currently available on the cost-effectiveness and customer acceptance of the air bag as an SRS system in the market. Some models of vehicles equipped with an air bag system could be examined to evaluate these commercial aspects of such an occupant restraint system in the years to come.

Designing a Passive Restraint System

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Abstract

The mandatory, phased introduction of passive restraint systems for the federal market has placed an increasing emphasis on the careful selection of suitable designs using a systematic approach.

Austin Rover Group's experience in the use of computer aided design analysis facilitates the use of such an approach and the implementation of the chosen design. This paper follows the progress of such a design process on a specific vehicle.

At the outset of the project, only belt based restraints were considered, but there were three possible approaches. Each was modelled and compared in different crash environments using computerized crash victim simulation (CVS). Attention was given to recognized injury criteria and the constraints imposed by other features of the vehicle concept.

From the model of the chosen system, worst-case loadings and energy absorptions were used by design areas to verify and develop specific items. In particular the fascia integrated knee bolsters and seat structure. In both cases finite element analysis methods were used.

For the bolsters the criteria involved maximum force levels and absorption of energy and for these a non-linear analysis with large displacements was applied. In the time available, an exhaustive study was not possible, but the computer analysis was used to highlight areas of strength and weakness by predicting the collapse mode. The bolsters were tested both individually and as part of the complete fascia to confirm the design performance. They were also tested as part of the full restraint system in a Hyge Sled Test using two dummies and realistic decelerations.

A full crash test confirmed the effectiveness of the system.

Introduction

The wearing of a seat belt has become a proven means of protecting a vehicle occupant from serious injury during a frontal impact. Much research has been put into the development and refinement of such systems to the extent that their performance is well understood. At the same time methods of analysis have developed such as sled testing and computer simulation, and these enable particular configurations to be appraised. Austin Rover has established methods based upon a mixture of testing and computer modelling for the design of restraint systems (Ref 1).

The US legislators proposed the phased introduction of passive restraint systems starting in 1987. There is a variety of types of possible passive restraint systems including the use of airbags and seat belts. Each has its merits and problems, but this paper will illustrate how the design philosophy established at Austin Rover has been applied to a belt-based passive restraint system. It will deal only with those aspects directly related to the dynamic performance in a vehicle impact environment. There is no reason why such an approach cannot be applied with equal effectiveness to an airbag-based system.

System Selection

The inclusion of a passive restraint system in an existing vehicle requires careful consideration of current components so that a minimum of change is introduced consistent with effective performance. At the outset it was stated that there should be as little change as possible to the basic interior package with particular reference to the fascia, and that design effort would be concentrated upon the options a belt-based system offered. Three possible systems were proposed as illustrated in Figure 1.

System 1 used a three-point belt with outer anchorages fixed on the door and an inboard retractor. To aid ingress and egress a motorized arm pulled the inboard webbing forward.

System 2 differed from the first by having retractors on the outboard ends and no motorized inboard arm.

System 3 utilized a single diagonal belt with a knee bolster and active lap belt. The outboard anchorage was fixed to the B-C post and motorized forward along the cant rail to aid ingress and egress.

Initial studies of comparative performance used computer simulations of each system in different crash conditions. The test data and crash victim simulation models (Calspan CVS Program) created during the design of the active restraint system, provided the basis for developing passive restraint models.

Initial preparations for these models used the BELTFIT computer program (Ref 2) to optimize the anchorage locations for each system for comfort. This program allows the user to define a set of anchorage locations and a seat outline and introduce a selected occupant shape within them. By finding the shortest possible belt path across the occupant's torso the program predicts the lie of the belt and from this makes some judgments about the comfort of the fit and its legal acceptability. The configuration can then be altered or a different size of occupant introduced at an appropriate seat location. Although this method only considers the comfort of the belt, it has also been found to yield good anchorage locations for dynamic performance. Typical examples are given in Figure 2.

Combining this information with the previously developed CVS models, three new models were set up to represent the optional designs. All models concentrated upon the restraint of a 50th percentile male (Hybrid II dummy) on the passenger side and all three models were set up with the same fascia/bolster stiffness taken from sled test experience with a rigid foam buffer. An example of the System 3 model is given in Figure 3.

Each model was subjected to three test conditions using data from crash test experience, to allow for intrusion and provide deceleration signatures. The test conditions were:

- 30 mph full frontal barrier impact.
- 35 mph full frontal barrier impact.
- 30 mph frontal impact with a 30 degree angled barrier.

The main questions to be answered were:

- To what degree does the complexity of including of a lap belt in the first two systems enhance performance?

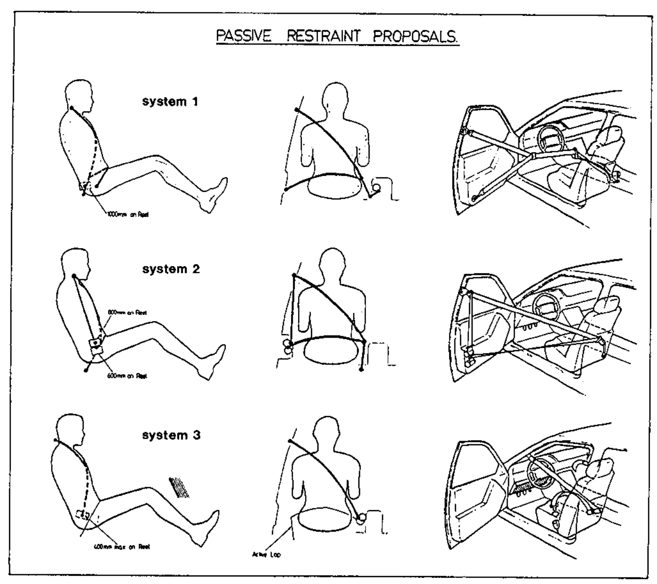

Figure 1

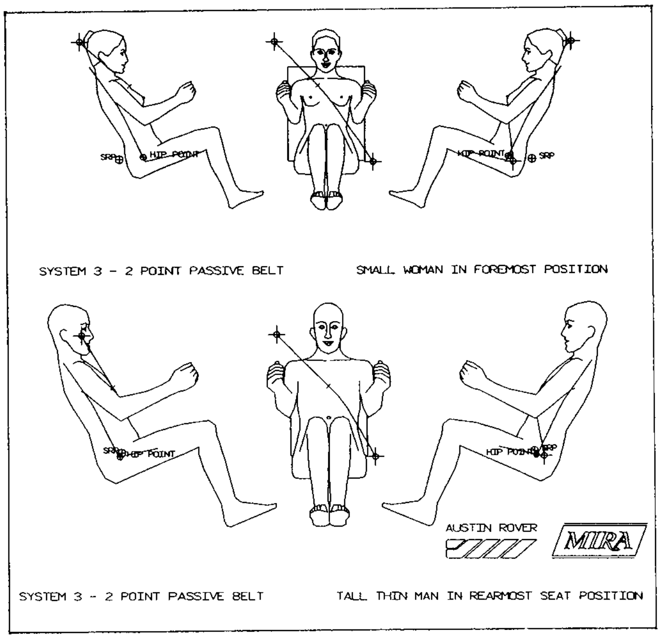

Figure 2. Typical BELTFIT examples

- Can satisfactory performance be achieved if the knee bolster surface on System 3 follows the line of the current fascia?
- Does the presence of a lap belt on the first two systems control lateral pelvis movement significantly better in an angled barrier impact?

The results are summarized in Table 1. All three systems performed in a manner which would have satisfied the FMVSS 208 injury criteria. The significant features were that System 3 produced less forward movement of the torso and lower head and chest accelerations but that the first two systems provided better pelvic restraint as reflected in the lower femur loads. The lap belts did not seem to contribute significantly to lateral pelvic control in the angled barrier impact.

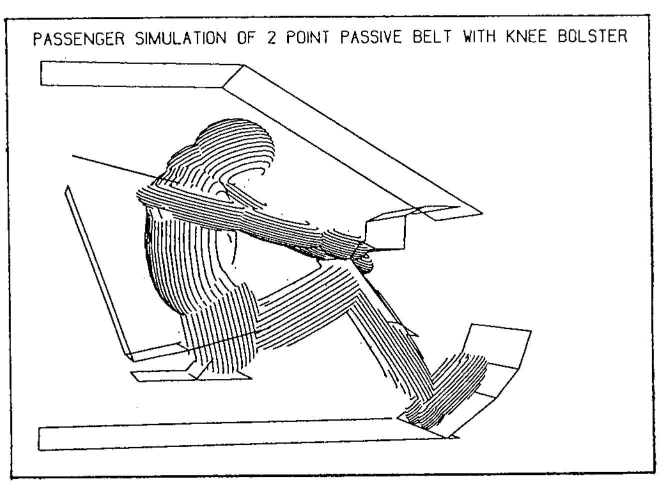

Figure 3. Typical CVS model

Table 1.

COMPARISON OF ALTERNATIVES

	HIC	CHEST G	FEMUR TO FASCIA LOAD (KN) R	FEMUR TO FASCIA LOAD (KN) L	FEMUR TO FASCIA ENERGY (J)
35 MPH FULL FRONTAL					
SYSTEM 1	743	41	6.4	6.5	419
SYSTEM 2	897	51	8.1	8.0	604
SYSTEM 3	452	44	8.4	8.4	723
SYSTEM 3 + LAP BELT	586	44	7.6	8.1	589
30 MPH FULL FRONTAL					
SYSTEM 1	410	35	4.2	4.1	118
SYSTEM 2	646	40	5.0	4.9	185
SYSTEM 3	275	39	5.6	5.8	287
SYSTEM 3 + LAP BELT	266	36	5.0	5.2	194
30 MPH 30 DEG. FRONTAL					
SYSTEM 1	203	37	4.6	4.6	154
SYSTEM 2	407	35	4.1	4.2	122
SYSTEM 3	182	36	5.7	6.0	312
SYSTEM 3 + LAP BELT	198	38	4.7	5.3	180

Closer examination of the results showed that the energy absorbed by the fascia was not greatly different for the three systems, especially at the higher speed where there was more intrusion. This arises from the location of the outer lap belt anchorage of Systems 1 and 2 on the door. This being a four door car, meant that the anchorage was well forward relative to the occupant and the belt had a near-vertical initial lie. It could provide little rearward restraint until the torso had moved forward enough to make the angle shallower. The presence of retractors directly associated with each lap belt also served to soften their effect through webbing being 'spooled' off under load.

Conversely the better torso restraint provided by System 3 resulted from a combination of better anchorage location and no outboard retractor.

It was felt that the results indicated that the knee bolster for System 3 could follow the surface of the current fascia provided its stiffness was optimised.

For the reasons given above and taking into account production and cost factors, System 3 was chosen for further development for this particular vehicle concept. As a final check the model was re-run with the active lap belt included. The results are included in Table 1 and show satisfactory performance with femur loads reduced but torso and head results slightly worse.

Hardware Development

The design criteria for the restraint system components were established from the computer studies and the sled tests. They were a maximum seat pan deflection of 30 mm for a distributed load of 10 kN, and an energy absorption for the bolster of 300 J/leg within a maximum femur load of 10 kN.

A distributed load of 10kN was applied over the anti-submarining pan of a previously developed finite element (f.e.) model of the seat. The magnitude of this load was obtained from occupant modelling of frontal barrier tests.

This exercise confirmed that the seat design was acceptable, in that the deflection due to the load applied by the occupant was within predefined limits (30mm) and caused no structural failure (see Figure 4).

The knee bolsters and their support system were to be constructed from spot welded sheet steel, and designed so that they would fit into the standard fascia moulding (see Figure 5).

The design philosophy was that the bolsters would deform on impact, but that the support system and fascia would remain relatively intact.

Finite element models of the knee bolsters were analysed in isolation from the rest of the system using the ABAQUS program. This allowed the effect of varing bolster dimensions, material gauges and types to be assessed, prior to the manufacture of prototypes. The design criteria was to achieve a target energy absorption within a predefined intrusion of the occupant knee into the bolster (see Figure 6).

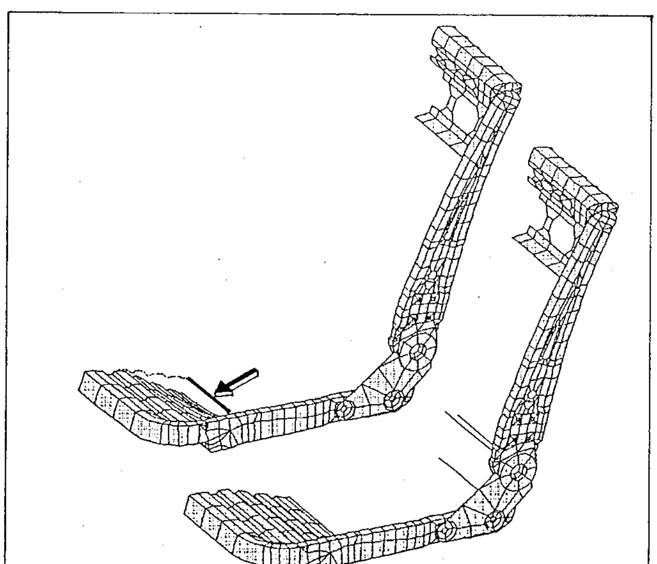

Figure 4. Finite element seat model

Prototype bolsters were then built and tested statically to compare their load/deflection characteristics with those of the model. Subsequently the prototype bolsters were modified using the data gathered from the analyses, and re-tested until the design target had been satisfied.

Attention was now turned to the bolster support rail. This had to provide support for the bolsters and fit to the standard fascia 'A' post mountings. A prototype system had already been manufactured, but f.e. analysis showed that this was unlikely to support the impact loads.

The f.e. model was then reinforced and local stiffnesses (obtained from a NVH Department f.e. facia model) were added. An analysis of the reinforced design gave favourable results, so a prototype was modified and assembled into a fascia along with the knee bolsters for further testing.

These tests were conducted using the in-house sled rig facility. Wooden knee forms mounted on a trolley were fired into the bolsters with the velocity and kinetic energy determined from film and CVS analyses. Individual load cells measured the loads which would be transferred to the occupant's femurs, and these were below the design criterion of 10kN (see Figure 7).

Development Sled Tests

To check performance, practical tests were carried out using dummies and a prototype system mounted on a Hyge sled. The bolster system and fascia were mounted to a stiff frame via brackets fabricated out of sheet steel. This allowed flexibility of mounting of

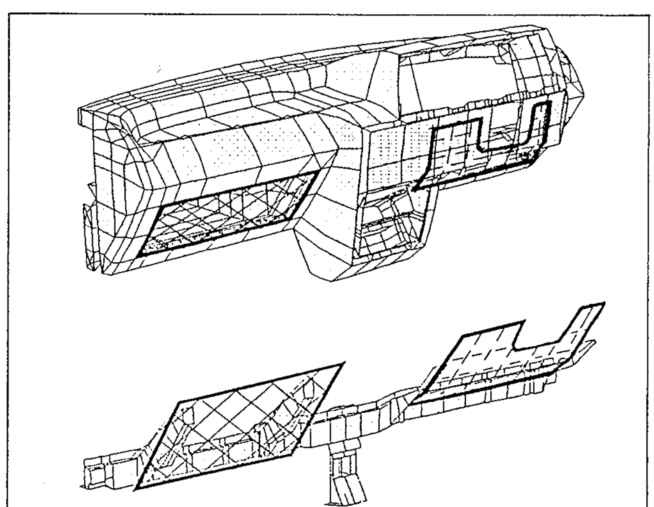

Figure 5. Finite element fascia & bolster system model

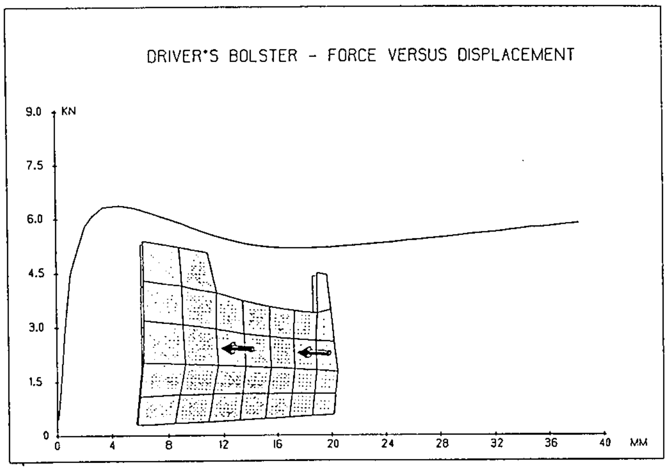

Figure 6

similar order to that expected on a vehicle. However, it was recognized that this could not represent a full crash test condition, as it omitted the components contained in the fascia and the dynamic intrusion which would be expected to affect bolster performance. A steering wheel and column were incorporated onto the frame at their working position. To load the bolster system correctly both driver and passenger were tested at the same time.

Two tests were carried out and Figure 8 shows the configuration of the first which simulated a 30 mph impact. For the second a 35 mph impact pulse was used and the seats were moved forward to give a rough representation of intrusion at the higher speed.

The results given in Table 2, are all within acceptable limits. The passenger figures are very close to the findings of the initial computer studies, although the bolster system is softer than the original foam buffer. The driver results show good restraint with the head striking the steering wheel lightly only at the higher speed. The performance of the bolsters followed that of the component tests. The overall conclusion was that the effectiveness of the system was confirmed.

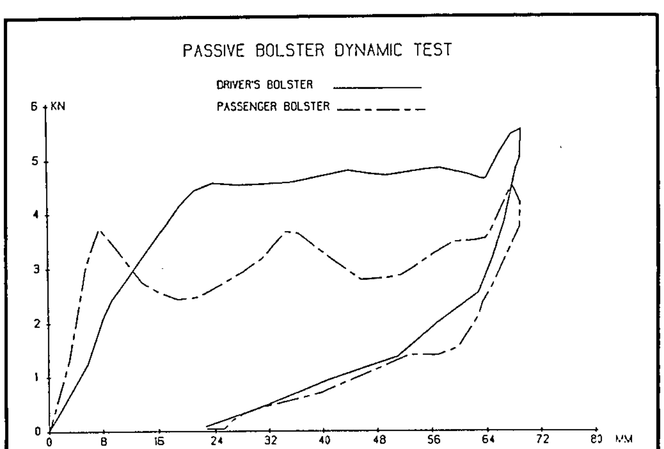

Figure 7

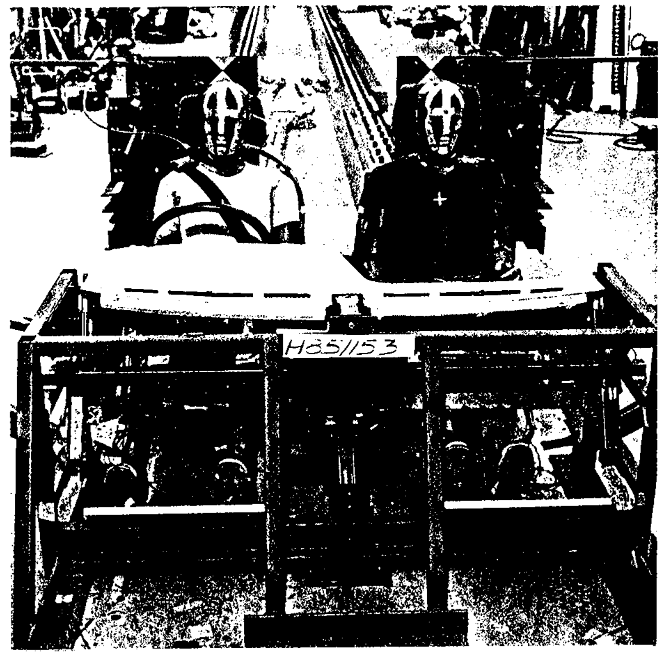

Figure 8. Sled test configuration

Final Verification

The final test of the restraint system was carried out in a vehicle 30 mph frontal barrier impact to FVMSS 208 conditions. The results are given in Table 3. Both driver and passenger results all lie well within acceptable limits. The passenger head and chest results are very close to those from the sled test. The femur results indicate that the intrusion of the fascia and the hardware within it have made the bolster stiffer.

The driver dummy results show mild contact with the steering wheel and the higher femur loads indicate that the movement of the column and local hardware has affected the bolster stiffness.

Table 2.

DEVELOPMENT SLED TEST RESULTS

35 MPH FULL FRONTAL

	HIC	CHEST G	FEMUR LOAD (KN) R	FEMUR LOAD (KN) L	TO FASCIA ENERGY (J)*
DRIVER	659	36	6.3	5.2	553
PASSENGER	610	43	2.0	3.9	450

30 MPH FULL FRONTAL

	HIC	CHEST G	FEMUR LOAD (KN) R	FEMUR LOAD (KN) L	TO FASCIA ENERGY (J)*
DRIVER	370	32	4.3	3.8	351
PASSENGER	259	31	2.6	4.0	275

(* ESTIMATED FROM FEMUR LOADS AND MOVEMENT ANALYSIS)

Table 3.

COMPARISON OF 30MPH DEVELOPMENT RESULTS

	DRIVER				PASSENGER			
	HIC	CHEST G	FEMUR LOAD (KN) I/B	FEMUR LOAD (KN) O/B	HIC	CHEST G	FEMUR LOAD (KN) I/B	FEMUR LOAD (KN) O/B
CRASH TEST	422	-*	4.4	8.0	269	28	5.4	6.5
SLED TEST	370	32	3.8	4.3	259	31	2.6	4.0
CVS ORIGINAL SIMULATION	-	-	-	-	275	39	5.6	5.8

(* DATA CHANNEL FAILURE)

Figure 9 and Table 3 show that the predictions of the original computer CVS model give good correlation with the results of the crash test.

Discussion

This paper has illustrated how a combination of computer analysis and practical testing has enabled an effective passive restraint system to be evolved within a limited time frame. This has been achieved by making use of the attributes of computer modelling to select a system, define its requirements and apply this data in the development of components. Practical testing was guided by the experience gained from the analyses and used to co-ordinate the parts into a working system. The effectiveness of this system has been confirmed in a full vehicle crash test.

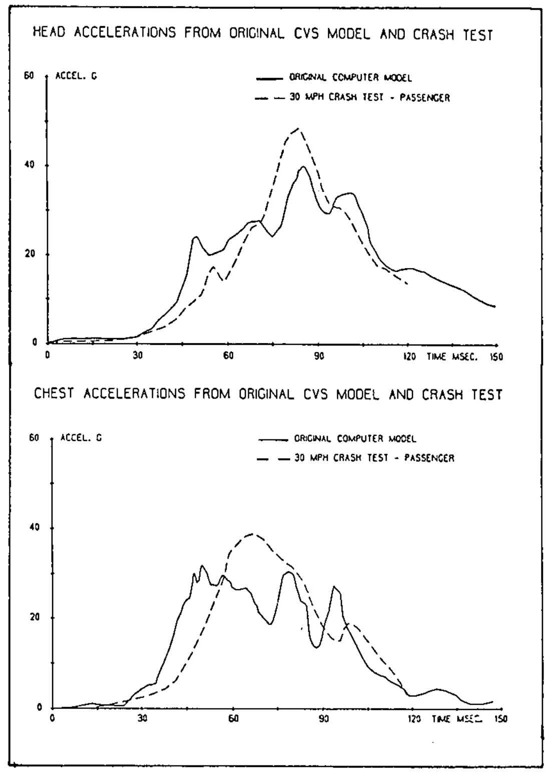

Figure 9

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2. "Development of a Computer Program to Design Seat Belt Layouts giving the Greatest Comfort, Safety and Convenience"—MIRA Report No. K52012/1

Using Computer Analysis Techniques in Designing Safer Steering Wheels

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Abstract

There is a world-wide increase in the wearing of seat belts, brought about by laws regulating their design and use. With the improved safety this brings, there is also a change in the pattern of injuries which arise, in particular, from the steering system.

The wearing of a seat belt changes the most likely injury site from chest to head and both research and proposed legislation recognize this fact (Ref 1).

Austin Rover uses computer-aided design techniques extensively in the design of its vehicles. Crash Victim Simulations can predict the nature of potential head/wheel impacts and rely upon design data to represent the wheel correctly.

Headform pendulum impacts are used to determine the stiffness characteristics of existing steering wheels.

By using a non-linear, finite element model, of a steering wheel, it is possible to predict its performance under pendulum impact conditions. The operation of crash victim simulation with this computerized finite element analysis allows an acceptable design to be developed prior to prototype construction. The technique allows material, construction and mass details to be varied in this process and can also be used to assess other properties such as strength and NVH.

One proposed form of legislative test adopts a pendulum impact but including a measure of impact

pressure. Work on the finite element modelling of pendulum impacts with steering wheels is continuing so that this feature can also be simulated.

Introduction

A world-wide increase in the wearing of seatbelts, brought about by laws regulating their design and use, has meant a change in injury patterns. This is particularly so for the driver, whose interaction with the steering system is now more likely to cause injury to the head than to the chest. Both research and proposed legislation recognize this fact (ref 1).

The following factors influence the severity of injury to the driver:

- Degree of intrusion of column and wheel into the occupant compartment
- Head trajectory dependant upon the restraint system
- The characteristics of local padding at wheel rim and hub
- The structural collapse of the steering wheel

This paper concentrates upon methods relating to the last of these and how they can be combined with the occupants trajectory to predict likely injury patterns.

Austin Rover uses CAD techniques extensively in the design process. Figure 1A illustrates how Crash Victim Simulation (CVS) is used in appraising the driver's restraint system and the effects of steering wheel design. The first part of this paper illustrates the development of this process.

The Calspan CVS program (Ref 2) is used in a modified form by Austin Rover for the analysis and development of restraint systems. Other analysis methods concentrate upon the structure of the vehicle and from these and practical tests, information on the crash environment is taken for the CVS models.

The steering system can be allowed to move within the CVS model but is usually placed in its final position because it has largely come to rest before the occupant interacts with it. However, it is essential to have an adequate representation of the steering wheel structure in order to predict the potential risk of occupant injury for each design.

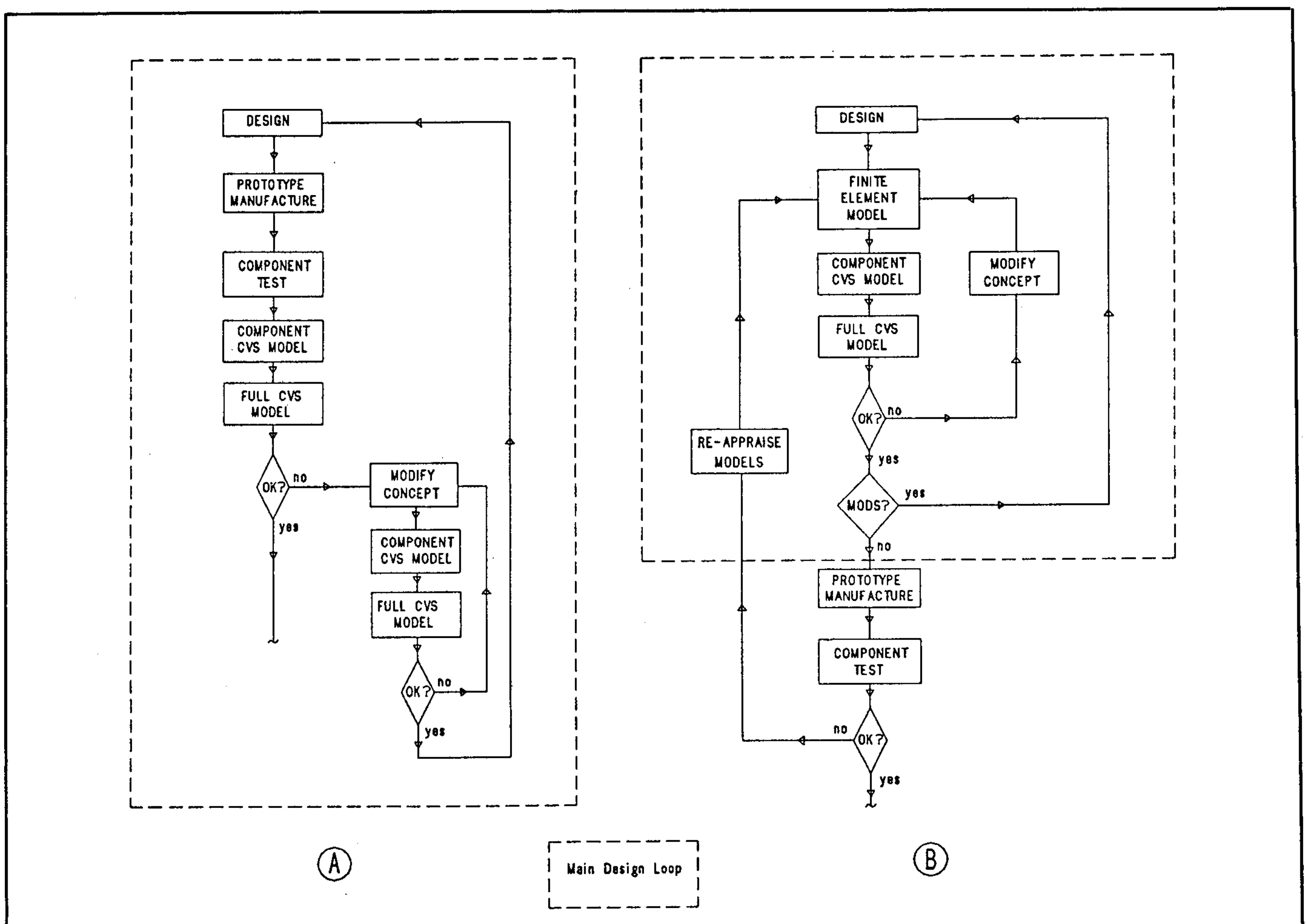

Figure 1. Current (A) and modified (B) design process

Component Impact Tests

Initial steering wheel collapse data was obtained from a series of pendulum impact tests. Three types of wheel were used (see figure 10):

- A two spoke A-frame wheel with semi-rigid, skinned foam covering.
- A three spoke wheel with semi-rigid, skinned foam covering (style 1).
- A three spoke wheel with soft, skinned foam covering (style 2).

Three important impact locations at which head and chest impacts could occur were selected as follows (See Fig 2):

1) Centre of upper rim
2) Centre of lower rim
3) Hub impact at 60 degrees

The steering wheels were attached to a steering column held rigidly by a rig with the column axis horizontal. The pendulum consisted of a hard rigid spherical headform of equivalent mass 6.8Kg attached to an arm of length 1.5M.

The pendulum impact speed, corresponding to the highest head impact velocities seen in crash tests, was 9M/S. The hollow headform contained an accelerometer mounted at the centre of mass.

Fig 3 shows the pendulum test rig setup for impact on the upper rim (location 1). Impacts on lower rim (location 2) were similar except that the steering wheel was rotated 180 degrees axially.

The headform trajectory was inverted when compared to that of an occupant's head in an accident. This was because of the expected rim deflection at the chosen impact speed. The more correct trajectory would have led to the pendulum arm striking the hub.

However, initial tests on the upper rim were unsatisfactory with the spherical headform due to the large deflections and consequent rideover (Fig 4a). To avoid this, for the upper rim impacts only the headform was replaced by a semicylindrical impactor (fig 4b). However, at the extremes of travel the rim was put into tension causing unrepresentative high accelerations. Further tests will include lower speed headform tests with the correct trajectory (fig 4c).

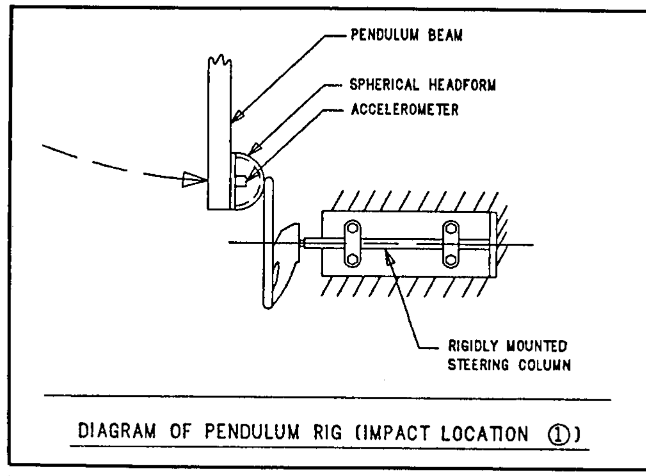

Figure 3

CVS Steering Wheel Model

The results from the pendulum tests were used to develop simple CVS simulations to act as pre-processors for full CVS models. These simulations treated upper and lower rims as separate masses jointed at the outer spoke locations about which the wheel rims were observed to bend (see Figure 5). The joint characteristics used were a constant friction torque representative of a plastic hinge together with some viscous damping.

Initial values for joint torques were estimated from the test results. By running this model several times and adjusting values it was possible to evolve a representation of the wheel which gave comparable results to the tests. The aim was to achieve a close fit to the initial acceleration peak and a good balance of subsequent average force and total deflection.

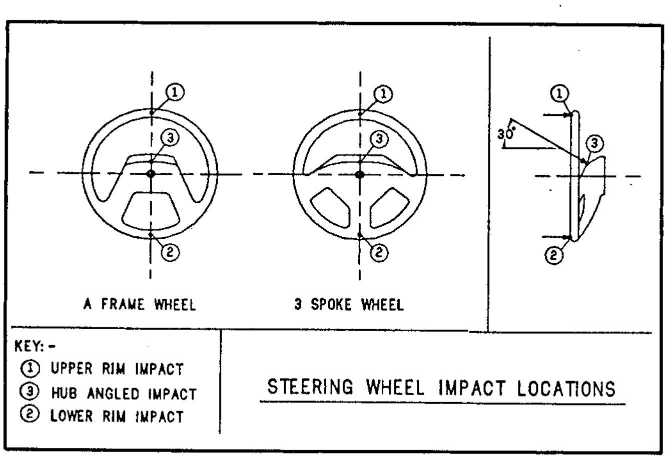

Figure 2

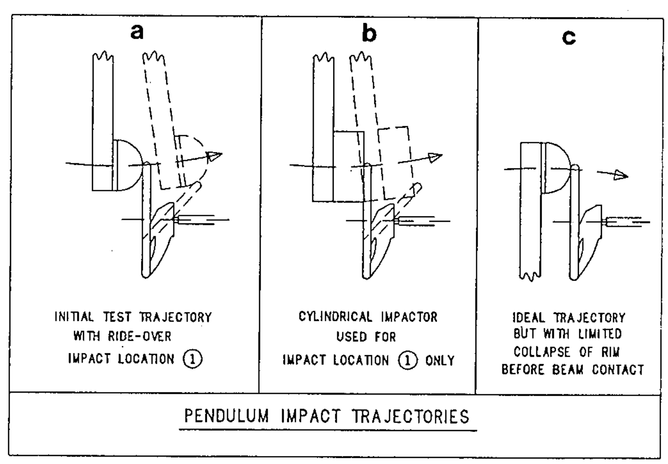

Figure 4

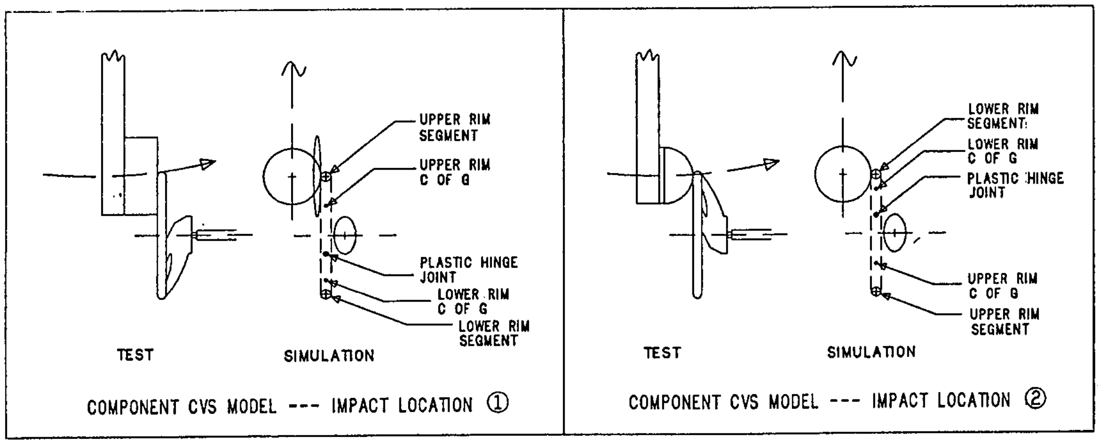

Figure 5

Figure 6 shows acceleration traces from test and simulation for an impact with the upper rim of the A-frame wheel. The simulation results show a good correlation of initial peak followed by an acceptably close pattern of acceleration. The high test accelerations later in the impact are a result of the impactor tensioning the rim as described earlier.

Figure 7 shows force-displacement traces from test and simulation of an impact with the lower rim of the 3 spoke style 2 wheel. The correlation is good.

Full CVS Model

The 3 spoke style 2 steering wheel representation derived from the simple pendulum simulation model was inserted into a full Crash Victim Simulation model. Included within this was the pendulum impact stiffness for possible hub contacts. Local rim compression stiffness was also included by combining data from a low speed pendulum impact of the rim against a rigid surface, and the stiffness of the material covering the dummy head.

Allowance for pre-alignment of the upper rim due to chest contact with the lower rim, was made using pendulum test data. In the future it may be possible to create a more complicated representation which automatically takes this into account. Figure 8 shows the full CVS model format.

This model was validated against crash tests and correlation was good. Typical head acceleration results obtained from the model are shown in Figure 9.

Alternatives to Practical Testing

It was realized that the limiting factor in the above process for steering wheel design appraisal was the need for practical testing of an actual steering wheel (see figure 1A). Clearly this limits the opportunities to effect changes in the design as a result of the CVS modelling because the creation of wheels for testing takes a significant period of time. Replacing the testing by an analytical technique would shorten the main design loop considerably (see figure 1B). This would mean that the loop could be traversed several times and allow alternatives to be considered before the design was committed to prototype manufacture. For this reason the Finite Element method was considered.

The benefits of applying the linear finite element (FE) technique are well proven and expounded even for very complex structures and loadings. However, a pendulum impact causes large displacements and non-

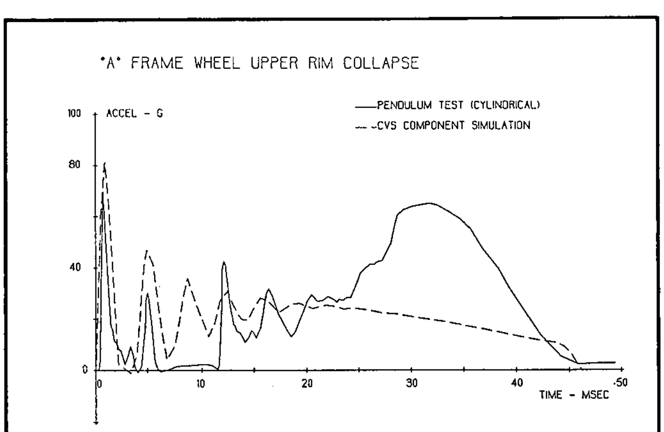

Figure 6

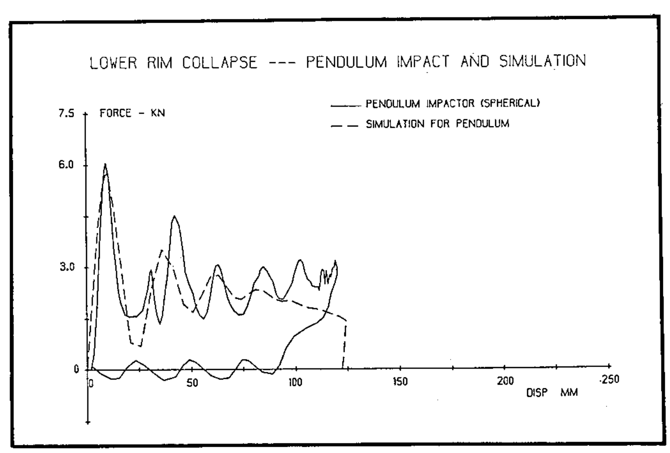

Figure 7

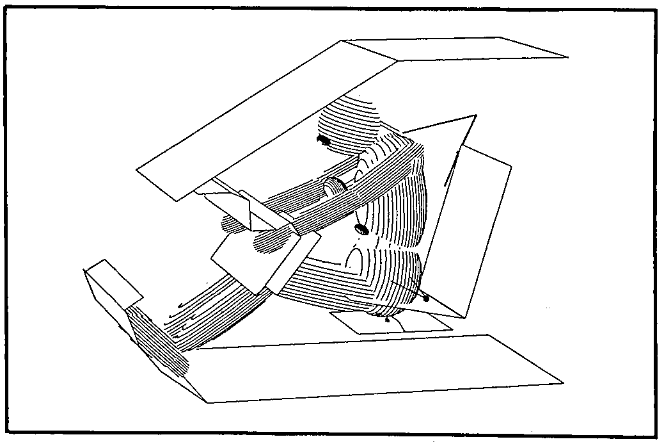

Figure 8. Full CVS model

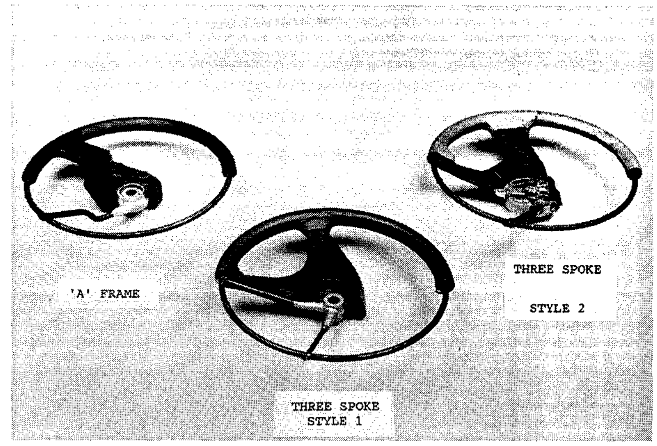

Figure 10. Steering wheel types tested

linear material responses. Application of the non-linear FE technique has to be done with care especially for complex shapes. Non-linear dynamic impact modelling in particular is not easy but it was felt that the steering wheel would easily lend itself to such an analysis as it is of a regular shape and form.

Application of the technique at an early stage in the design process not only enables data suitable for CVS to be obtained, but also highlights any poor aspects of the design in general as the model may also be used for general design analysis such as NVH prediction (wheel 'shake').

Finite Element Steering Wheel Model

All three types of steering wheel shown in fig 10 were modelled. The main construction of all the wheels is similar and comprises a solid bar rim armature, with solid bar spokes welded to it and a central diecast hub. In each case the whole assembly is covered with a self skinning foam.

The finite element model that was used comprised a series of beam elements representing the armature and spokes, and relatively stiff beam elements representing the hub.

To provide an initial check of the model, static bending tests, on an A-frame wheel, were carried out loading the structure well into the plastic region. Although exact material data could not be obtained the general form of the models behaviour was good (see figure 11). The load discrepancy at high deflections of the upper rim arose because the model took no account of any structural effect of the foam covering present on the test sample.

The pendulum impact tests were modelled assuming an impactor with a linear motion normal to the plane of the rim. This was taken as acceptable for the initial period of the impact up to a deflection of 150mm. No account was taken of the foam other than its mass. The graphs for the acceleration of the impactor compare well with the experimental data for the impact period of interest for both the A-frame and the three-spoke, style 1 wheels (figs 12, 13 & 14).

It was observed during trial runs of these models that the mass of the rim had a considerable effect on the acceleration of the impactor. To investigate this effect a model was constructed using a tube of equivalent stiffness to the standard rim bar, this reduced the mass of the rim by 50 percent. The effect of this on dynamic impact performance was to reduce

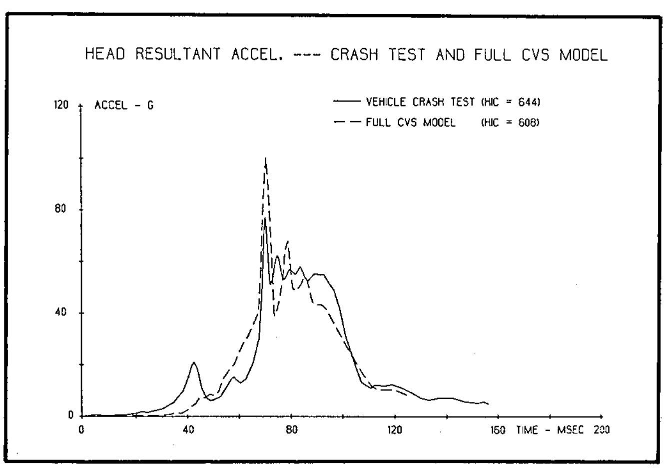

Figure 9

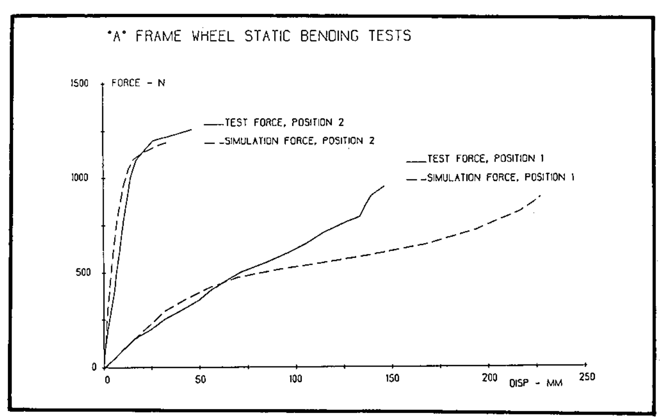

Figure 11

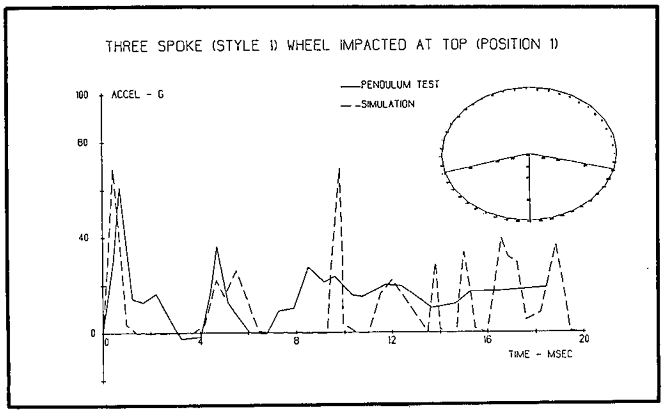

Figure 12

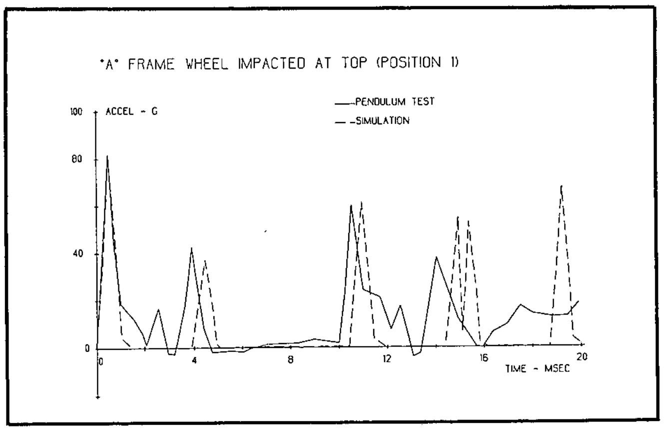

Figure 13

the initial peak acceleration by 50 percent. Other runs indicated that reduction of the rim mass is considerably more effective than altering its bending stiffness.

The results for the impactor acceleration on the three-spoke wheel style 2 do not compare so well (fig 15). The reason for this is the comparatively thick

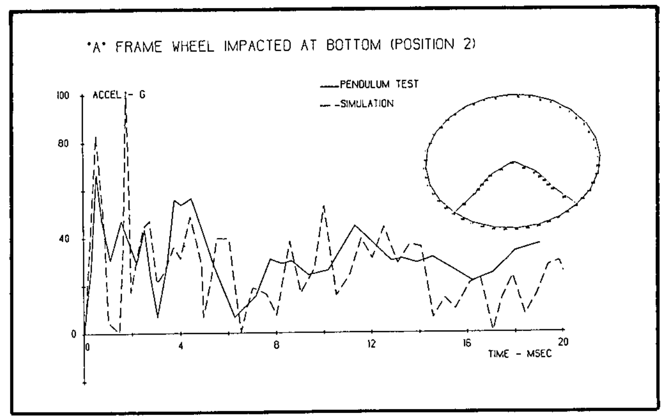

Figure 14

layer of 'soft feel' foam on this wheel, whereas on the other two wheels it was a comparatively thin layer of hard foam. Thus for this wheel the foam provides a significant amount of energy absorption and impact cushioning. This was confirmed by testing an uncovered armature which gave closer results to the corresponding simulation (fig 15).

It would be possible to represent the foam as solid elements but this would complicate the model. In order to keep the model simple, it was attempted to represent the foam as a spring-damper system. The results obtained were disappointing, and so it would seem necessary to model the foam by a more complex spring-damper system or by a finite element mesh. The model will inevitably become more complex, as it is adapted to represent the 'soft face' impactor tests envisaged to simulate potential facial injury.

Summary

Austin Rover has used Crash Victim Simulation methods to predict the likely nature of contact and

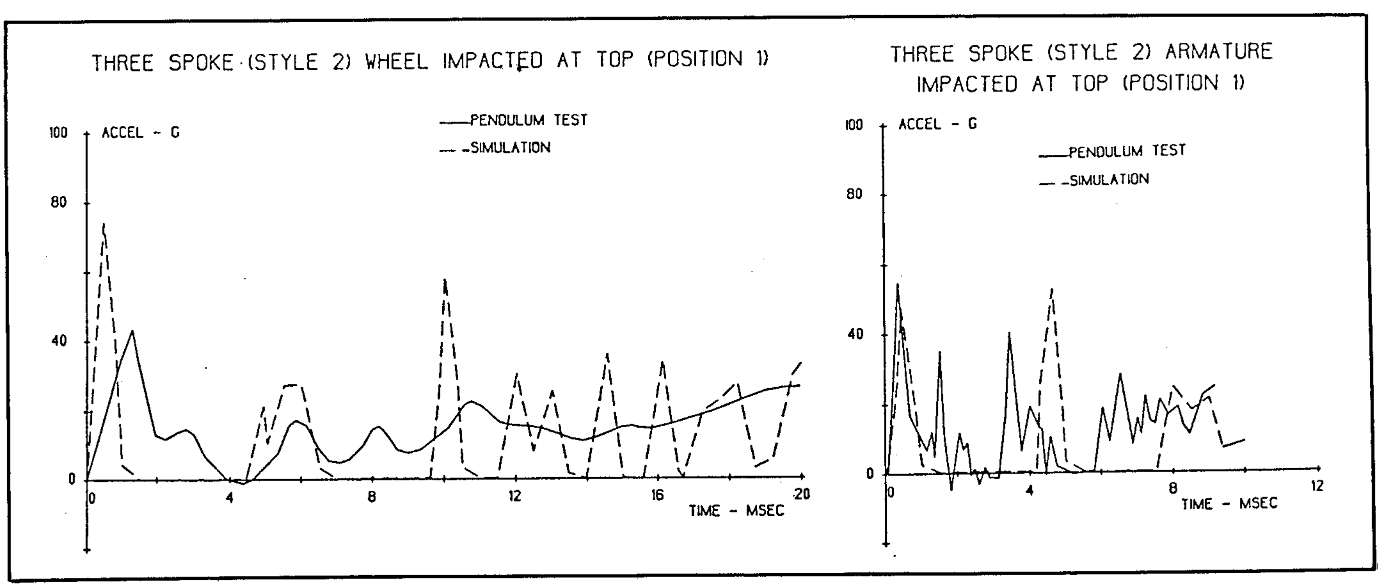

Figure 15. Three spoke (style 2) results

potential injury from the steering system of a vehicle.

To enable this, data has been obtained from pendulum impact tests on steering wheels and a satisfactory representation created on a simple, pre-processor CVS model. This then forms part of the full CVS model used to appraise the design in a vehicle impact. However the time required for prototypes to be created and tested limits the advantages of using the CVS technique.

With the application of the nonlinear FE technique suitable data can be obtained at an early stage in the design process, before any wheels are available. The main design loop can then rely entirely upon analytical methods for the evolution of the design features. As such, it can be invoked repeatedly over a relatively short period of time. A further advantage to be explored is that the same FE model can also be used to investigate other design parameters such as static strength and NVH performance.

Some types of foam covering have a significant effect on impact performance, and it will be necessary to model the foam completely in the FE model. This is especially important for extending the model to predict performance in the 'soft face' impactor tests.

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Facial Injury Occurrence in Traffic Accidents and Its Detection by a Load Sensing Face

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Abstract

Head and facial injuries are becoming less frequent and less severe in modern day cars. Volvo has conducted accident research with respect to frequency, location and severity. Comparisons between belted drivers and front seat passengers are made. Only about 6 percent of the passengers that sustained some kind of injury had a facial injury. The corresponding figure for the driver was 10 percent. With respect to fracture and contusion, the following facial areas emerged as the most often injured: nasal region, forehead and mandible.

The fracture and contusion type injuries can be detected by using a new load sensing face with piezo electric sensors. This face was subjected to some P572 calibration tests with results similar to those obtained with a standard Hybrid II head. The face was mounted on a Hybrid II dummy and subjected to sled testing. The kinematics were not affected by the umbilical cables etc. Consequently, the face can be used in normal testing without significantly affecting other measured safety parameters such as the HIC. Also, for future biomechanical research to establish injury criteria, the load sensing face promises to be a helpful tool.

Introduction

Facial injuries are becoming more important in car safety design. This does not mean that the number or severity of injuries in the face are increasing. In fact, they are both decreasing as a consequence of improved restraint systems, improved car interior design, the increasing number of seat belt wearers, etc. However, it has not been possible to totally eliminate facial injuries. The proportion of the less severe injuries in the facial area is therefore growing, which must be accounted for by new test methods in product safety development.

Head injuries have traditionally been divided into two different groups depending on the injury mechanism, i.e. acceleration and pressure induced trauma, figure 1.

Hitherto, most safety research has concentrated on the severe head injuries. These could be detected by means of acceleration levels measured at the center of gravity of the head. Federal Motor Vehicle Safety

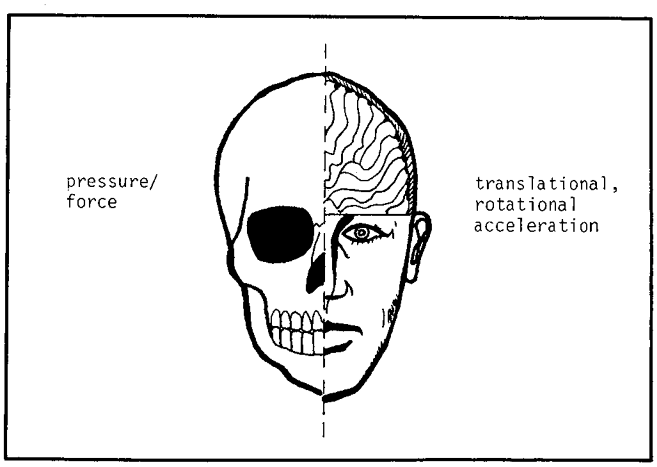

Figure 1. Injury mechanisms

Standard (FMVSS) 208 utilizes this method(1,2,3,4 and 5). Another method used is the rotational acceleration measurement.

The less severe head injuries, for example fractures of different facial bones, have until recently not been possible to measure. Here, the force or pressure applied must be evaluated. Various attempts have been made; deformable foam systems and strain gage force transducers are only two of several methods(6, 7 and 8). However, these methods have proved insufficient.

A load sensing face that utilizes thin piezoelectric pressure sensitive films was developed and presented by Volvo in 1986(9), figure 2. The dummy facial area is covered by 52 of these sensors, each measuring the force applied to the different facial segments. An on-board data acquisition system stores the data from all sensors. After the test, a host computer retrieves the data and presents the force—time histories from each individual sensor. The load sensing face can be used to associate the applied force with facial bone fractures and other connected injuries such as contusions.

This paper presents the load sensing face and two studies that were carried out in parallel. The first part of the paper deals with the Volvo accident material and the information we have obtained from in-depth accident studies. Laboratory use of the load sensing face, including calibration and some dynamic testing, is discussed in the second part.

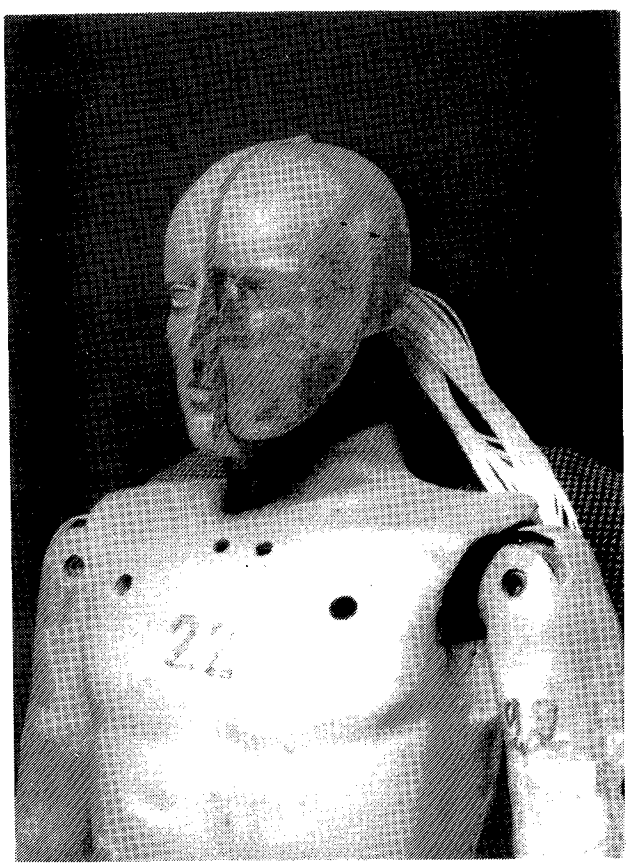

Figure 2. The load sensing face

Accident Knowledge

An accident study was made to try to answer the following questions regarding facial injuries:

- what is the proportion of severe to moderate injuries?
- which facial regions are most often injured?
- is there a difference between driver and passenger?

The studied sample was selected from a database of accident material, sampled according to a cost repair criterion(10), i.e. tow-away accidents. In the material both injured and uninjured persons, altogether 15,000 occupants, are represented. The sample mainly contains belted occupants. The unbelted frontseat drivers and passengers only add up to 6-7 percent. A statistically significant conclusion is thus not possible to draw from this unbelted group. The impact direction chosen to study is frontal to oblique/frontal (11 to 1 o'clock). The structure of the material chosen for the accident study is shown in Appendix 1.

Driver Accident Analysis

Volvo's analysis showed that approximately 10 percent of the belted drivers that were injured in frontal to oblique/frontal accidents sustained injuries in the facial area.

In order to obtain a proportional assessment of driver facial injuries, the severity of the injuries was assigned to an appropriate level in the Abbreviated Injury Scale (AIS). The result is shown in table 1. It can be noted that the majority of the facial injuries was coded AIS 1 and to some extent by AIS 2.

The driver facial injuries were then studied to establish which region had been injured. The injuries were divided into groups; laceration, abrasion, contusion, fracture and others. We found that the less severe lacerations dominated the number of injuries. The most complicated of these injuries, the fractures and the contusions, are presented from a severity, a

Table 1. Facial injury proportion of belted injured drivers and passengers.

AIS		Proportion (%)	
		Driver	Passenger
1		77	85
2		17	11
3		5	4
4		1	0
5		approx. 0	0
6		0	0
	Total	100	100

locational, and a proportional point of view in figure 3.

Passenger Accident Analysis

Investigation of passenger statistics shows that less than 6 percent of the belted passengers injured in frontal to oblique/frontal accidents sustained injuries in the facial area.

Table 1 shows that the facial injuries of the passengers did not exceed the AIS level of 3 at all, and only 4 percent of them exceeded AIS 2.

An investigation to study facial regional differences, corresponding to the driver study, was performed. It showed that also among the passengers, the less severe lacerations were dominating. Fractures occurred in even fewer cases than among the drivers. Figure 4 shows the distribution of the contusions and fractures.

Comparison Driver and Passenger

Compared with the drivers, the front seat passengers sustained less frequent and less severe injuries. This can be explained by the absence of a steering wheel, and thus a longer available deceleration distance for the passenger.

While the steering wheel can be said to be the main impact area for the driver, the instrument panel is the region most frequently hit by the passenger. This probably explains the reversed ranking between the frontal and the nasal area for driver and passenger respectively.

Injuries lower down the face, such as the mandible area, are as frequent among the passengers as among the drivers regarding the percentage. Counted by numbers, however, these injuries occur as rather unusual for the passenger category. This is probably due to a lesser likelihood of any impact at all because of the limited vertical extension of the instrument panel compared with that of the steering wheel.

Injury Coding

The analysis revealed that the majority of facial injuries is coded by the lowest AIS levels. Some examples of injury coding for different AIS levels are as follows: AIS 1 is ascribed to a fractured tooth, a smaller hemathoma in the forehead etc. The facial injuries coded AIS 2 include a fracture through the nasal bone without displacement. Mandible fractures are coded AIS 1-2. Bilateral fractures of the zygomatic bones or the maxillary bone in accordance with Lefort III are injuries which can have an AIS value of 3.

The AIS scale(11) is mainly a means of estimating the threat to life risk. The AIS in its present form, however, does not adequately measure the level of disability or the actual harm sustained by the individual. A disability scale that would complement the AIS and provide the link between injury assessment and societal costs is being pursued.

Injury Detection

Two of the injury types described above, laceration and abrasion, can be detected using chamois skin. Lacerations of the skin covering the dummy face are judged with respect to the number of lacerations, depth and length. The outcome of this is a figure which should correspond to an actual laceration pattern in real life. The other two injury types, contusion and fracture, can be measured by the load sensing face.

Injury Thresholds

The various areas of the face are significantly different with respect to anatomical design, strength of the facial bone and the overlying soft tissue. Combined, this presents a complicated problem, i.e. the injury thresholds differ substantially. The thresholds are also dependent upon how the force is

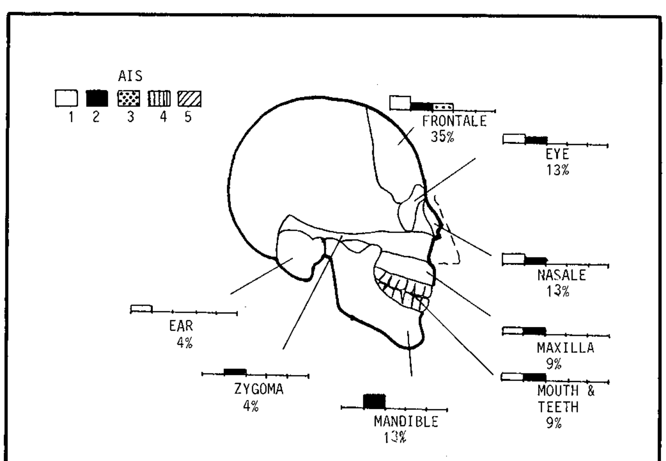

Figure 3. Driver facial injuries; fracture and contusion

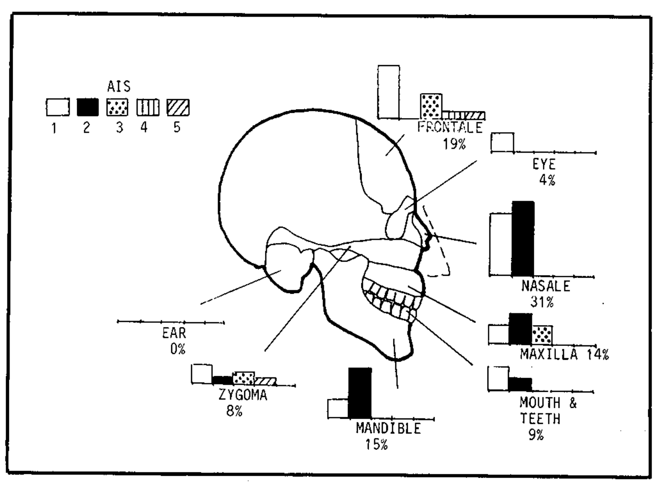

Figure 4. Passenger facial injuries; fracture and contusion

applied. One should take into account both the impacted surface characteristics, as well as the impact direction, the specific pressure and the duration of the force subjected to the face.

The effect of the impact speed has not been fully examined, most biomechanical researchers have used impact speeds typically below 10 m/s. The reported research has also concentrated on impacts on various facial bones, but the effects from an overall pressure has seldom been studied. The impacted surfaces have mostly been of a rigid, sharp design that is not representative of steering wheels, instrument panels or other interior parts of today's cars.

Research by Nahum (12), Schneider (13), Tarriere (14) and others have provided us with injury thresholds for different facial segments. However, these values are not correlated to a pressure or force registered by some kind of recorder, such as a dummy head, used for measuring car interior impacts during crash testing. We believe that the load sensing face is an excellent tool to evaluate injury thresholds and that it can find extensive use in future biomechanical research. This correlation between biomechanical data and measured parameters such as force or pressure to the facial segments, is necessary to make it possible to understand how to further improve the interior safety design.

Laboratory Experience With the Load Sensing Face

The prototype load sensing face is derived from a Hybrid II head(9). The attachments of the piezo-electric pressure sensitive sensors, the umbilical cables etc, were all chosen not to significantly change the characteristics of the head. In order to evaluate the possible differences between the load sensing face and the standard Hybrid II head, it was subjected to various tests.

Calibration and Validation

The FMVSS 208 P572.6 and P572.7 procedures were used as a basis for studying the suitability of the load sensing face as a test instrument.

Mass. The mass of the prototype head including skin and a 3 axis accelerometer is 4.93 kg. This can be compared with the 4.54 + – 0.04 kg that a standard Hybrid II head weighs. The excess of the load sensing face, 0.39 kg, can be lowered in future head designs by using other, lighter materials and thus bringing the weight within acceptable levels.

Center of Gravity. The electrical cable from each sensor is lead to a junction box positioned in the skull cavity shown in figure 5. This slightly alters the center of gravity for the head. The c.o.g. in the horizontal-vertical plane is moved 9 mm forward and 12 mm upward. This can be considered an acceptable change of the c.o.g.

Table 2. Head drop tests.

Test	Resultant acceleration (m/s^2)	Pulse duration at 981 m/s^2 (ms)	Unimodal curve	Lateral acceleration (m/s^2)
Load sensing face				
Test 1	2509	1.28	Yes	- 51.8
Test 2	2683	1.15	Yes	- 56.9
Test 3	2686	1.15	Yes	- 95.5
Standard Hybrid II				
Test 1	2682	1.28	Yes	-168.9
Test 2	2733	1.28	Yes	- 34.2
Test 3	2762	1.15	Yes	67.8
Requirements	2060 to 2551	0.90 to 1.50	Yes	- 98.1 to 98.1

The Polar Moment of Inertia. The polar moment of inertia of the head with skin cover was also measured to check the mass properties. For the lateral y-axis the polar moment was measured to be Iy = 0.0243 kg m2. Published values for an average Iy of 0.0233 kg m2(15) for cadavers indicate that the load sensing face has good inertial equivalence.

Head Drop Tests. Matched head drop tests were performed with the load sensing face and a standard Hybrid II head as a reference. Both heads used the same PVC skin to be able to compare only the skulls themselves. Three tests were made with each head. The results are summarized in table 2.

As can be seen from table 2, the load sensing face meets the requirements except for the resultant acceleration in two tests, where the values are slightly over the specified range. However, since the standard Hybrid II head we used exceeded the limit in all three tests, the load sensing face can be regarded as fairly close to the requirements. Also, with some allowable modifications of the skin-skull friction coefficient, this would probably move the measurements into the desirable range.

Neck Pendulum Test. Neck pendulum tests were performed with the load sensing face and a standard Hybrid II head. The neck and the PVC skin were the

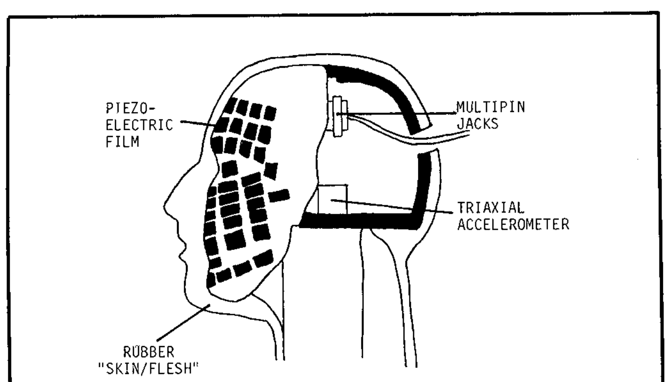

Figure 5. Schematic view; the load sensing face

Table 3. Neck pendulum tests.

Test	Res. acc. (m/s2)	Max. Rotation (degrees)	Time at max.rot (ms)	Max.chordal displacement (mm)
Load sensing face				
Test 1	287	65	52.6	135.5
Test 2	279	65	53.0	134.6
Test 3	286	66	53.2	135.2
Standard Hybrid II				
Test 1	293	62	53.8	128.4
Test 2	293	62	53.5	127.1
Test 3	287	62	53.4	127.0
Requirements	max. 255	63-73	53.2-66.8	127.0-152.4

same for both heads, only the skulls were shifted. Both neck and PVC skin were new and had not been used in earlier tests. Three tests were performed with each head. The results are summarized in table 3.

Table 3 shows that the load sensing face meets the requirements concerning maximum rotation and maximum chordal displacement. The resultant acceleration shows higher values than are required, but these values are even higher for a Standard Hybrid II head. The requirements will possibly be met for both dummies if the neck is changed.

The higher values of the chordal displacement for the load sensing face will allow the straight line motion of the head's center of gravity relative to its initial point to be longer than for the Hybrid II head. Thus, when interpreting the results of a test, it should be born in mind that the degree of severity will appear somewhat higher for the load sensing face in its present form than for the Standard Hybrid II.

Temperature Sensitivity. The load sensing face is designed to provide accurate readings in the 22 +- 11 degrees Centigrade temperature interval. Since the FMVSS 208 requires the stabilized temperature of the test equipment to be between 18.9 and 25.6 degrees Centigrade for Hybrid II testing, the face should work well in the laboratory. This is also the case in Hybrid III tests, where the allowed temperature is between 20.6 and 22.2 degrees Centigrade.

Testing

In future routinely performed crash testing one can distinguish at least three different ways of using the load sensing face: barrier crash testing, sled testing and component testing. The outcome from two of these test methods, the sled and the drop test, will be discussed below.

Sensor Calibration. Before testing, the face has to be calibrated. This is performed inside a hydro dynamic chamber. The face is protected by a rubber boot and placed inside the chamber. All sensors and a reference strain gage are subjected simultaneously to a steep pressure pulse, supplied by inert gas. The data acquisition system compares each individual sensor with the reference pressure and correction factors are computed. The corrections are due to zero offset, charge leakage and scale factors. After calculation they are stored in a file of individual sets of calibration factors for each sensor to be used later for evaluation of impact data(9).

Dynamic Testing. The aim of the first test runs was to provide answers to the following questions:

- what are the effects on the dummy kinematics due to the umbilical cables attached to the dummy head?
- how does the onboard MDAS (Modular Data Acquisition System), designed to withstand high g-levels, function during the crash simulation?

A Volvo car body, from the 700 series, was mounted onto the sled. A Hybrid II dummy, equipped with the load sensing face, was positioned in the passenger position. It was restrained by a three point belt. The sled was then subjected to a simulated 30 mph barrier crash. High speed films were used to check the dummy trajectory. The umbilical cables were routed out of the back of the dummy skull (figure 7). The connections to the multipin jacks inside the skull cavity were secured by means of a steel plate that was fastened by two bolts. Enough slack in the cables was provided to ensure that these did not limit the movement of the dummy. An extra amount of cables was positioned behind the back of the dummy to allow for upper torso displacement during the crash event.

Pre-test preparation also included examination of the MDAS box to establish what precautions were necessary in order to prevent any damage resulting from high g-levels. The power supply unit, which is

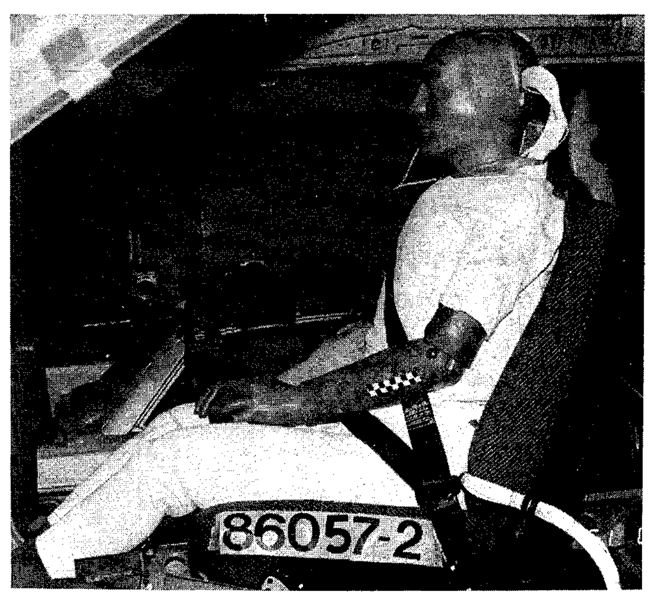

Figure 6. Pre test; sled testing

the heaviest part of the MDAS box, was secured inside the MDAS shell by means of sheet metal support and rubber padding. The MDAS unit was dismounted after the two tests and checked for any damage.

The MDAS box itself was positioned in the longitudinal direction of the car in order to have the lowest possible strain to the MDAS internal electronics. The box was attached to the floor by means of rubber cylinders (figure 8). The whole unit was mounted in an area behind the front seat.

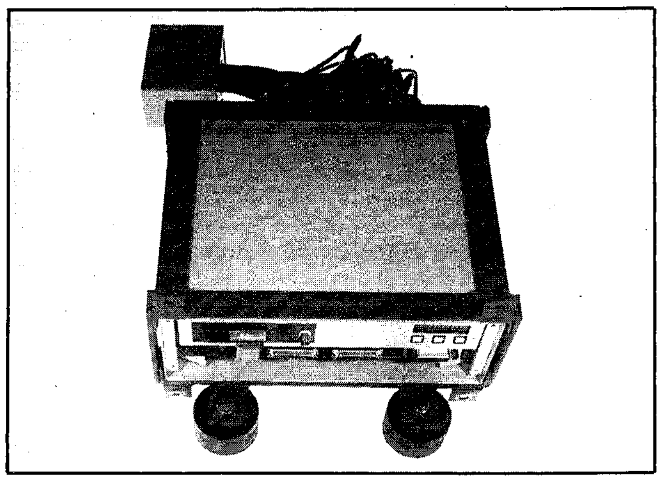

Figure 8. MDAS protection box

Sled Test Evaluation. Tests were made with a dummy complete with the load sensing face and its cables. A series with a standard Hybrid II head was also performed for comparative reasons.

Although few tests were made, it seems that no major difference in dummy trajectory at the c.o.g. of the head will appear. This will be further studied in future sled testing.

For future applications the cables from the load sensing face will be improved in the following ways:

- the thickness and stiffness of the cables can be reduced to some extent
- the cables will be routed out of the base of the skull to be parallel to the vertical axis of the neck

Both countermeasures will even further lessen the resulting effect from the cables. Then the acceleration signal from the center of gravity of the head will not be affected, which means that the HIC calculation can be compared with tests utilizing a standard Hybrid II dummy head.

The MDAS box sustained some damage in the latter part of the test series. To avoid this, and also to be able to run tests at higher speeds, the g-force resistance preparation has to be improved.

The load sensing face can also be used in component testing in a drop test device, figure 9.

This method is applicable when testing steering wheels, instrument panels or other interior components. It enables the test engineer to have a simple and inexpensive method to make a coarse evaluation of the safety performance.

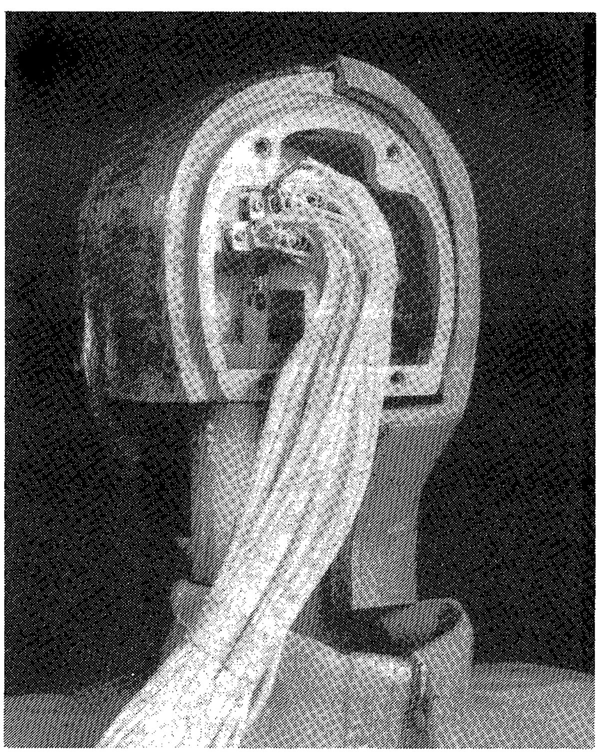

Figure 7. Cable attachments; back plate removed

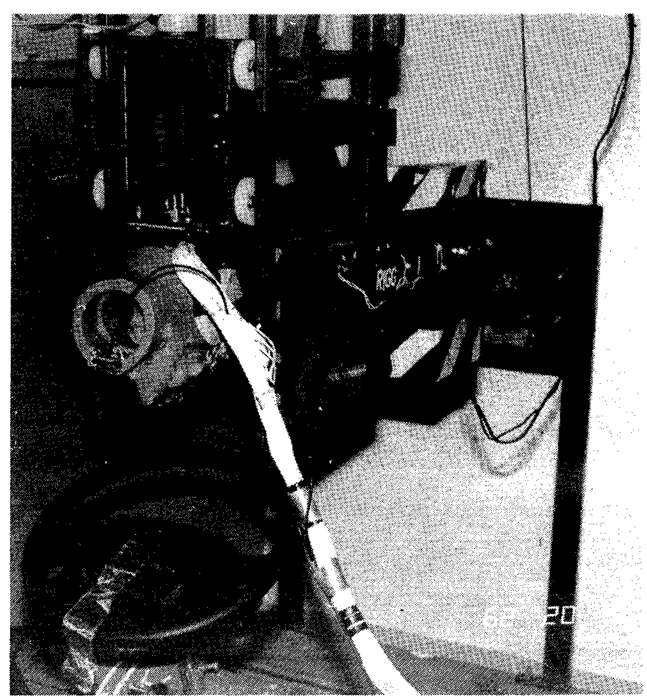

Figure 9. Component testing; impact against steering wheel

Hybrid III Prototype

In the first prototype of the load sensing face 52 sensors covered the facial region. These sensors were mounted rather uniformly over the dummy face.

From preliminary testing with the head form we have found that the number of sensors, 52, involves unnecessary long computer time and that after testing, the time for analysis by the test engineer has increased. To be able to use the load sensing face in routine testing, the number of sensors should be decreased. This, however, must be reached without losing the detailed picture of the sustained facial violence.

From our accident survey, we know that four facial areas are of special interest: the nasal region, the forehead, the maxilla and the mandible. A new sensor configuration should at least cover these areas.

A Hybrid III prototype with only half of the number of sensors used in the previous prototype has therefore been designed. The new sensor location is based on the principle that neighbouring areas with equal resistance to impact and with similar curvature can be covered by the same sensor.

The conclusions above are derived from accident statistics obtained mainly from frontal and oblique/frontal impacts (11 to 1 o'clock). Consequently, if, for example, side impacts were to be studied, the sensor location would probably be somewhat different. If the face is to be used in side impact testing, the temple region should be covered with sensors to a greater extent.

Future Development

The load sensing face is currently undergoing further testing.

The hardware equipment, such as the MDAS unit and the umbilical cables will be further improved. The aim is to withstand higher g-forces and also to limit the effect on dummy movement.

Computer software will be developed in order to make the facial forces easy to survey.

An attempt will be made to simulate the nose and the structure underlying it. This could be obtained by using a frangible insert for one-time use only which can be changed without removing flesh and head from the dummy.

Summary

Even though the facial injuries in the accident statistics are few and minor, there is clearly a need for a device that can measure facial injuries. The load sensing face developed by Volvo detects fracture and contusion type injuries.

Accident research done by Volvo includes injury types and injury severity for both belted drivers and front seat passengers. Of the occupants sustaining some sort of body injury, only 10 percent of the drivers and 6 percent of the passengers sustained a facial injury. Of the facial injuries the majority was coded AIS 1-2. Some small differences were seen between the driver and the passenger, both with respect to injury frequency as well as to facial location.

The face can be used in both crash and sled testing with complete dummies as well as in component testing. Calibration tests of the head, including drop-test, mass properties, etc., together with some limited dynamic sled testing indicates that the load sensing face does not particularly alter the Hybrid II head characteristics. The extra equipment—umbilical cables, etc.—do not significantly affect the dummy kinematics.

For future biomechanical research to establish injury criteria the load sensing face promises to be an excellent tool. Correlation between injuries and forces measured by the face is necessary, so that knowledge can be gained to further improve interior safety design.

Acknowledgements

The work presented here is the result of the research and deliberations of many persons. We would particularly like to acknowledge Prof. Charles Y. Warner of Collision Safety Engineering and Dr. Milton G. Wille of Brigham Young University who made the development work on the load sensing face. We also wish to thank Agneta Ebbesson for her contribution to the calibration and comparison of the load sensing face and the Standard Hybrid II head. The authors would also like to thank Johnny Korner for his help in the accident study.

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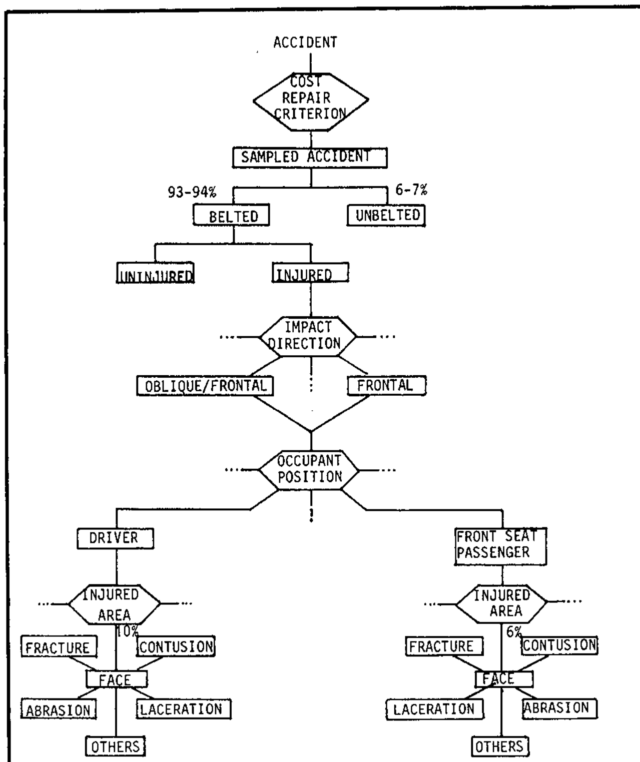
Appendix 1



Flow chart, accident sample

The Child in the Volvo Car

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Abstract

The objective of this report is to describe Volvo's development work in the field of child safety. Experience from car accidents involving children is used to describe different modes of travel for children of different age groups, the effectiveness of different child restraint systems and problems of misuse.

The problems of differences in the requirements of child safety legislation are discussed.

This experience combined with experience gained from laboratory tests constitutes the basis for development work on child safety systems in Volvo cars.

Volvo's new child safety program covers all age groups of children and needs for different ways of travel.

Volvo Safety Design Philosophy

To Volvo, safety has always meant safe transportation in a real traffic environment. Volvo Safety Design Philosophy can be illustrated by a circle as in Figure 1.

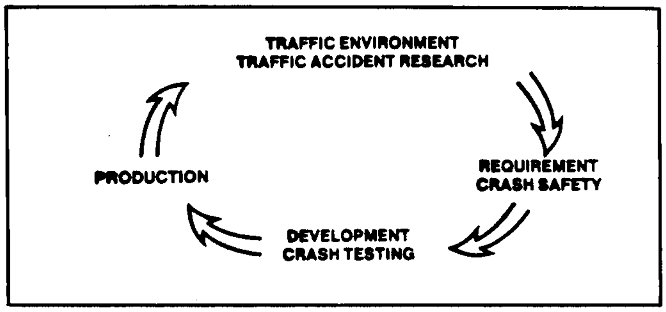

Figure 1. Volvo safety design philosophy

For many years the Volvo Traffic Accident Research Team has been extensively engaged in investigating accidents and increasing its know-how about the crashworthiness properties of complete vehicles and their various design systems, and about the various occupant injury mechanisms. This knowledge is used both for short-term and long-term feedback in the development of future vehicles.

This feedback is one important source of information for establishing the safety property requirements of a car. These requirements then provide the basis for design and development work.

Traffic Accident Experience

Experience gained from the Volvo accident material described below gives us knowledge of the way children travel in cars and injury risks for different modes of travel.

Background Data

Each year, Volvo compiles information on about 2,000 of the most serious road accidents in Sweden involving Volvo cars. The accident material is based on a repair cost criterion where all cars of a repair cost of 15,000 SEK or more are selected. In addition, in-depth studies are made of between 150 and 200 serious road accidents, in which a Volvo car was involved as well as a number of special studies.

The accident study material for the years 1976-1986 covers approx. 1000 serious road accidents in which at least one child was an occupant of the car. The material involves Volvo cars of the 140/160, 240/260, 340/360 and 740/760 models.

Mode of Travel

There were a total of 1601 children involved in the accident material. In the case of 1463 children, it is known whether or not they were travelling restrained. Only these 1463 children are covered in the following analysis.

Figure 2 below shows how children split into different age groups have been travelling during 1976-1986.

Comments. Of the childen in the age group 0-11 months, 6 travelled in an infant seat and 4 in a child seat. All seats facing rearwards.

28 of the children between 0-11 months travelled in a carrycot. The reason for this is that up to 1984 it was recommended in Sweden that infants should travel in a carrycot. In 1984, however, infant seats were introduced in Sweden and today—1987—approx. 90% of the infants travel in infant seats.

Since only rear facing childseats are recommended in Sweden all except two seats in the accident material were facing rearwards.

In figure 3 the percentage of children in different age groups using some type of safety equipment is shown. In this report safety equipment is recognized as infant seats, child seats, booster cushions and seat belts.

As we can see from figure 3, restraint use is highest amongst children from 0-1 years (53%) and for children from 1-3 years (46%).

MODE OF TRAVEL	AGE 0-11 months	1-3 years	4-6 years	7-10 years	11-14 years	TOTAL
RH front seat-belted		2	5	14	15	136
RH front seat-unbelted			1		6	7
Outboard rear seat-belted		7	34	46	69	156
Outboard rear seat-unbelted		41	108	170	213	532
Centre rear seat-belted		4	8	13	5	30
Centre rear seat-unbelted		24	42	56	37	159
Child seat/Infant seat	10	82	5			97
Booster cushion		22	51	36		109
Carrycot	28	6				34
Luggage compartment-unrestrained	2	4	9	8	4	27
Other modes*)-unrestrained	7	64	53	32	20	176
TOTAL	47	256	316	375	469	1463

*) Other modes of travel include lying or standing on the rear seat, sitting on the lap, sitting between occupants

Figure 2. Mode of travel for different age groups during 1976-86. (Volvo accident material)

Children between 4-10 years have the lowest restraint use (31%) while restraint use for the older children is 40%.

However, the restraint use for children in all age groups has increased rapidly from 1981 and onwards, as shown in figure 4.

As we can see from figure 4, the percentage of restrained children increased from 22% in 1976 to 72% in 1986.

The group of children that accounts for the growing restraint use mainly consists of children using booster cushions and seat belts. This primarily means children between 4-10 years, but to some extent also includes children between 1-3 years.

The increased restraint use amongst children from early 1980 can be explained by many factors. During the beginning of 1980s many campaigns were carried out in order to provide knowledge of the importance of using child safety equipment when travelling in cars. Some years later intensive campaigns dealing with the use of seat belt in the rear seat started.

In 1984 some county councils in Sweden started loaner programs of infant seats directed towards people with newborn babies. These loaner programs, which today cover all of Sweden, have meant that approx. 90% of all infants travel in an infant seat today. The infant seats have to a great extent replaced the carrycots as a way of travel for infants. In 1986 a law making rear seat belt use also compulsory came into force in Sweden. Though the law excluded children under the age of 15, it has probably had an effect on increasing public awareness of the need of being restrained when travelling in cars.

Injuries to Children

Injury frequency. In this section the accident material described above is used to analyse the injury risk for restrained and unrestrained children. The injury risks for restrained and unrestrained children are compared. Furthermore, calculations of restraint effectiveness are made.

As mentioned earlier, restrained children are those who use a two/three-point seatbelt, a booster cushion or a rearward facing child seat. Unrestrained children are those who travel in another way, except for children travelling in a carrycot. These children (carrycot) are excluded from the comparisons in this section.

The restraint effectiveness (e) is defined as the injury rate reduction attributable to restraint use, given as a percentage of the injury rate without restraint:

$$e = \frac{\text{injury rate, unrestrained} - \text{injury rate restrained}}{\text{injury rate, unrestrained}} \times 100$$

We will first compare the injury rate for all restrained and all unrestrained children. The injury rates are given for three levels of AIS. Totally there are 528 restrained and 901 unrestrained children between 0 and 14 years of age.

We can see from figure 5 that there is a restraint effectiveness at all AIS-levels. A restraint effectiveness of 50% can be calculated for injuries of a severity level AIS 2-6.

We then divide the restraint group into three subgroups. These subgroups are seat belt, booster cushion and childseat according to the previous definition.

From figure 6 we can see that the injury rate for children using rearward facing child seats is extremely low compared to other modes of travel. For children using child seats and for children using booster cushions there are no injuries more severe than AIS 3.

Injuries more severe than AIS 1 for children using child seat and booster cushions are described in the appendix.

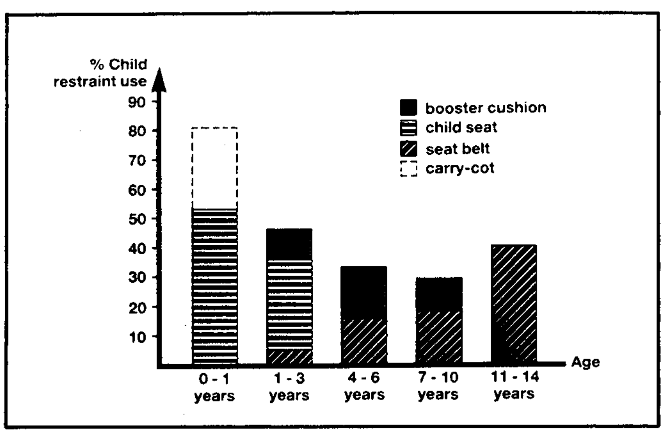

Figure 3. Percentage of children in different age groups using some type of safety equipment 1976-1986 (Volvo accident material)

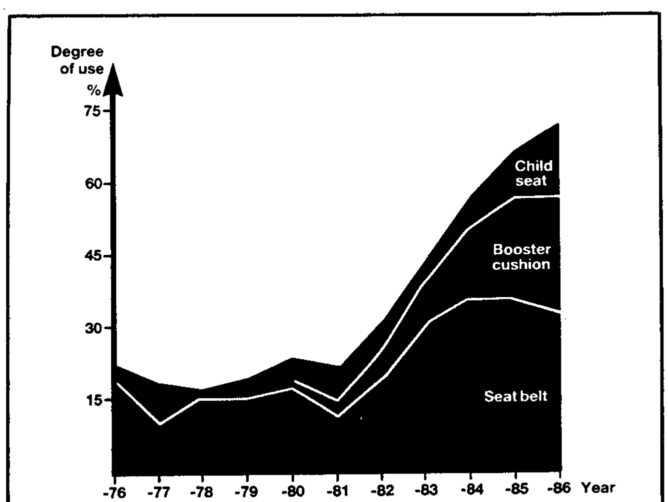

Figure 4. Percentages of children using restraints from 1976 to 1986 (Volvo accident material)

Restraint	Injury rate (5) AIS 1 %	AIS 2-3 %	AIS 4-6 %
Yes	25.9	4.4	0.8
No	30.9	8.3	2.0
Restraint effectiveness	16%	47%	60%

Figure 5. Injury rate and restraint effectiveness at three AIS-levels. Restrained and unrestrained children in the age group 0-14 years

When estimating the effectiveness of the different types of restraints, it is important that each restrained population (e.g. child seat user) does not differ too much from the unrestrained population in variables than can influence the injury rates.

There is a difference in age distribution between the groups of children presented.

To reduce the risk of drawing faulty conclusions we compare the effectiveness for seatbelt and for booster cushion only for children in the ages between 3 and 10 years and the effectiveness of rearward-facing child-seat in the age group 1 to 4 years.

The injury rates for all children (0-14 years)(figure 5) and children in the age group 3 to 10 years (figure 7) do not differ much:

From the figures in table 7 we can calculate the effectiveness in reducing AIS 2-6 injuries for seat belts to 58% and for booster cushion to 63%.

If we compare children in the age group 1 to 4 years depending on whether they have travelled in a child seat or travelled unrestrained we can calculate a very high effectiveness for the child seat in reducing minor as well as severe injuries. The effectiveness in reducing AIS 2-6 injuries is about 90%.

Among the children who used a child seat there is only one injury more severe than minor (see case 1 in appendix).

Types of injury. In figure 9 the numbers of maximum AIS 2-6 injuries for each body region are presented

Mode of travel	Injury rate (%) AIS 1 %	AIS 2-3 %	AIS 4-6 %	Total No.
Seat belt	29.8	5.3	1.2	322
Boosters	29.4	3.7	-	109
Child seat	9.3	1.0	-	97
Unrestrained	30.9	8.3	2.0	901

Figure 6. Injury rate for three types of restraints for children age group 0 to 14 years. Three AIS-levels

Mode of travel	Injury rate (%) AIS 1 %	AIS 2-3 %	AIS 4-6 %	Total No.
Seat belt	31.3	3.5	0.9	115
Boosters	31.1	3.9	-	103
Unrestrained	30.0	8.6	1.9	536

Figure 7. Injury rates for unrestrained children and children using seat belts or booster cushions in the age group 3 to 10 years. Three AIS-levels.

for different types of restrained children and for unrestrained children.

The most common type of injury of this severity (AIS 2-6) for each group is the head injury.

Of the 97 children travelling in a rearward facing child seat, there was only one injury more severe than minor, an AIS 2 injury to the head which is described in the appendix (case 1).

The AIS 2-6 injuries sustained by children using a booster cushion are described in the appendix (cases 2-5).

For children in child seats and children using booster cushions none sustained neck or abdominal injuries of level AIS 2-6. For belted children one neck injury case was reported. This case was a severe sideswipe accident with a heavy truck. The belted driver was killed. The 12-year old belted girl in the right rear seat sustained a compression fracture on one of the cervical vertebraes (AIS 2) and a minor bruise on the abdomen.

From these injury figures we can see that the risk of severe neck and abdominal injuries caused by belt use is very low.

Misuse

Misuse can be defined as partial misuse or gross misuse. Partial misuse means, for example, child not properly restrained, wrong size or age of child, restraint too old. Gross misuse means for example incorrect mounting or no mounting of child restraint.

Some surveys(1)(2) have shown that a great percentage of child safety equipment is being misused. However, the consequences of misuse from the injury point of view are relatively unknown and depend on the type of system being used.

Mode of travel	Injury rate (%) AIS 1 %	AIS 2-3 %	AIS 4-6 %	Total No.
Child seat	7.2	1.0	-	97
Unrestrained	26.8	10.6	1.5	198

Figure 8. Injury rates for children, 1 to 4 years of age, using a child seat or being restrained. Three AIS-levels

Child seats. According to accident data from the USA, a correctly used safety seat reduces the fatality risk by 71% while a partially misused seat reduces fatality risk by 44%(1).

To obtain information on misuse in the Volvo accident material some questions are put concerning mounting of the seat and restraint of the child.

In short this material shows that most of the child seats were mounted correctly. In none of these cases was it indicated that the seat came loose from its attachments. Only in 2 cases was the child not properly restrained in the seat.

This indicates that the misuse frequency of rearward facing child seats is low. This is also confirmed by the high injury-reducing effectiveness (90% for AIS 2-6 injuries).

Booster cushions. In the 109 cases with booster cushions, 105 children used the booster cushion together with a lap-shoulder belt, 4 children together with a lap belt. In some cases children used a booster cushion without being restrained in a seat belt. These children are considered as unrestrained in this report.

From the accident material it has not been possible to draw any reliable conclusions about the mounting of the booster cushions.

The best method of measuring the misuse of booster cushions is probably to make on-the-road inspections of the mounting of the boosters. Such inspections have been made in Sweden and these indicate that approximately 40% of all booster cushions are misused in some way. This type of misuse is mainly partial misuse, where the safety belt is not properly attached to the seat belt guide on the cushion (2).

Although the partial misuse of booster cushions is relatively high, there is a clear effectiveness (68%) in reducing more severe injuries (AIS 2-6).

However, more information is needed about the extent to which the misuse influences the injury outcome.

Mode of travel Body region	Seatbelt No. %	Cushion No. %	Childseat No. %	Unrestrained No. %
Head, face	15 4.7	4 3.7	1 1.0	70 7.8
Neck	1 0.9	-	-	4 0.4
Neck frontal	-	-	-	1 0.1
Chest	1 0.3	1 0.9	-	9 1.0
Abdomen	-	-	-	9 1.0
Pelvis	-	-	-	3 0.3
Vertebrae	1 0.3	-	-	1 0.1
Upper extr.	2 0.6	1 0.9	-	20 2.2
Lower extr.	2 0.6	1 0.9	-	17 1.9
N =	322	109	97	901

Figure 9. Injury rate (AIS 2-6) body region vs type of restraint and unrestrained children 0-14 years of age.

Test Experience

Each year, Volvo performs about 1000 tests in the Crash Safety Centre. Out of these tests about 80 are fullscale tests, 400 are crash simulations on a Hyge sled and 500 are component tests. Both frontal, rear end, oblique, offset and laterial collisions are tested and simulated. When the development of a new part or an accessory is discussed, relevant crash tests in different cars are planned. It is mostly a question of crash simulations but as far as possible tests in ordinary fullscale collisions are made. For instance, when the new child safety programme was under development almost 300 tests were carried out in the different Volvo cars to make sure that the child seats would behave in a proper way according to our own requirements.

Test Methods

Almost every country in the world has its own national requirements concerning child safety. This creates problems for car manufacturers and others when developing and/or manufacturing child safety equipment intended for different countries. One example of this is that some countries do not permit the transport of children under a certain age in the front seat, even if they are properly secured in a child seat. This means that it is difficult to manufacture a rearward facing child seat as those seats are normally installed in the front seat, leaning against the dashboard.

In North America another problem arises with the rearward facing seat. When certifying a child seat according to the federal regulations, it is only permitted to use a standard bench and a safety belt. It is currently not permitted to use something to lean a child seat against, for instance a dashboard or a front passenger seat. Therefore it is not possible to get approval for a rearward facing toddler seat.

The Swedish authorities, however, encourage both producers and users of child safety equipment to transport children up to approximately 4 years in rearward facing devices.

A European regulation, ECE 44(4) has been adopted by almost all the countries in Europe. Manufacturers of child restraints may choose whether they want to apply for the ECE 44 approval or the national approval in the relevant countries.

What ECE 44 is to Europe, FMVSS 213(5) is to the USA. An overall comparison between the two requirements is made in figure 10. Some expressions have been used in that comparison which may be described as follows:

Classes

- the integrated class, in which the belts and the seat are completely integrated with the child restraint

- the non-integrated class, in which the adult belt is used to restrain the child

Categories

- the universal category, in which the device is connected to the lower seat belt anchorages
- the semi-universal category, in which an extra anchor-point is used
- the specific vehicle category, in which any type of attachment to the car can be used.

It should be noted that FMVSS 213 requires a set of American p572c dummies, 6 months and 3 years old. The 6 month old dummy is uninstrumented while the 3 year old dummy has accelerometers in the head and in the chest.

ECE 44 requires another set of dummies called the TNO-dummies, 9 months, 3, 6 and 10 years old. They are all instrumented with an accelerometer in the chest.

Our experience is that there is a big difference between the two sets of dummies. The 3 year old TNO dummy is, for instance, more sensitive to submarining than is the 3 year old p572c dummy. The 10 year old TNO dummy moves in an unrealistic way. The chest seems to be very stiff which together with a weak 'lumbar spine' means that the dummy has a tendency to slip out of the shoulder belt very easily, i.e. jackknifing. This problem has also been mentioned in other reports, for instance in (3).

Even though we are convinced that a rearward facing child seat offers a better overall protection than does a forward facing one, we have designed the combined child seat to be used in both ways. The reasons for this are:

- Forward facing child seats are generally used in countries outside Scandinavia. With this seat, we want to give people an opportunity to try the rearward facing seat as it is "included in the price" when they purchase it as a forward facing seat.
- To raise the usage rate of child restraints for this age-group as the child seat is useful from infancy up to approximately 4 years of age.

We made some sled tests in complete car bodies with existing forward facing child seats before we started the development work on our new child safety programme. The seats were all approved according to different regulations, for instance ECE 44, FMVSS 213 and F.

No child seat fulfilled our own requirements as regards effect in the car. For instance, submarining occurred with some European child seats. With some American child seats the dummy displacement was so big that the head of the dummy hit the back of the front seats. The reason why these things happened will be discussed below.

One of the most important parameters in the matter of forward facing child seats is the way the seat is secured to the car. FMVSS/CMVSS 213 requires that the seat is secured to the car with the ordinary seat belts while ECE 44 gives the designers free hands to choose whether they want to use the ordinary seat belts or extra fittings.

It is easier to obtain good crash performance with two extra straps that are bolted at the ordinary lower belt fixation points. This is also the most common solution in Europe. There are two reasons for this:

- The designers can choose the location of the straps on the child seat. This is very important as the angle of the straps decides how swiftly the child seat is restrained during a frontal impact. The more upright the angle, the greater is the rotation of the straps before they start to refrain the child seat. This results in a longer displacement of the child, which means an increasing risk of head contact with the interior, especially in smaller cars.
- All cars are not equipped with safety belts in the rear seat.

ECE 44	FMVSS 213
Classes	
Integral Non integral	"Integral"
Categories	
Universal Semi-universal Specific vehicle (1)	"Universal"
Age groups	
Group 0 < 10 kg Group 1 9-18 kg Group 2 15-25 kg Group 3 22-36 kg	"Infants, 0-6 months (-7.5 kg)" "Toddlers, 6 months-4 years (7.5 kg - 18 kg)"
Other items	
Special rig with a simulated dashboard.	Special rig, no dashboard.
30 mph frontal collision 20 mph rear-end collision	30 mph frontal collision
It is possible to use extra fittings, the dashboard etc.	The CRS has to be fitted into the car with a lap belt only. It is possible to use one extra fitting, but it is necessary to meet the regulations without the extra strap. It is not permitted to use the dashboard.
Main requirements	
Chest res. < 55g/3ms Chest vert. < 30g/3ms Head displacement No submarining	HIC <1000 Chest res. < 60g/3ms Head and knee displacement

Figure 10. A comparison between FMVSS 213 and ECE 44. 1) It is possible to make the test either in a fullscale crash test or in a special vehicle body on the test-trolley

Use of the ordinary seat belts, as in all American child seats, creates the problem of the difficulty in tensioning the belt across the child and the child seat itself. This means that it is much harder to devise a good "force taking" angle of the lap belt. As many child seats today are designed in accordance with the federal regulations, another problem arises, i.e. that neither ECE 44 nor FMVSS/CMVSS 213 requires a complete safety belt when the crash test is carried out. These regulations do not require the use of a safety belt with a lock and locking tongue which in turn influence the fitness and the behaviour of the child seat in a proper car. As not all cars are the same, one can find cars with very high locks (1.5-2 dm) as well as cars with very low ones. The high locks may mean that it is very difficult to install the child seat as the lock often is very stiff. During a crash, this may result in bending of the lock which may destroy it. Furthermore, the efficiency of the restraint may be lower since the belt tension is reduced. This may also mean that the child seat becomes unstable during normal driving conditions. The big advantage of securing the child seat with the seat belt is of course that it is easier to use.

As mentioned before, tests of certain European approved forward facing child seats resulted in submarining problems in frontal impacts. This is a problem that does not exist in either rearward facing child seats or in North American forward facing seats. The reason why it does not occur in rearward facing devices is obvious. In North American child seats the reason is to be found in the design of the harness.

The ECE 44 regulation requires that if the child is secured by a 5-point harness, then the crotch strap or any other strap passing between the child's thighs must break or disconnect from its fitting at a static load of not more than 50 N. On the other hand, FMVSS/CMVSS 213 requires that the crotch strap shall be fixed and not give way under any circumstances. The reason why ECE 44 requires a releasable crotch strap is the belief that the crotch strap may result in crotch injuries. Our opinion however is that the fixed crotch strap is much better as it keeps the lap belts low down on the pelvis which means that the hips will not move forward and cause injury in the genital area.

We have mounted a high-speed camera during a couple of tests to see what really happens in that region. No problems occurred as the fixed crotch strap held down the lap belts over the hips. What may happen with the releasable crotch strap is that the shoulder belts will pull the lap belts upwards towards the stomach as the shoulder belts are connected with the lap belts. This means that the risk for abdominal injuries increases as submarining may occur. It is, however, possible to design a forward facing seat the "European way" without the submarining problems, which is what we have done. We believe, however, that the European system is rather sensitive to changes in crash-pulses, stiffness of the seat cushion etc. This means that a "European" forward facing child seat ought to be car-specific or semi-universal rather than for universal use.

This judgement is made with the North American and the British markets in mind as no information on crotch injuries is available from the accident statistics. The British regulation permits the use of the fixed crotch strap.

Conventional Infant Seats

It seems that almost all countries believe in rearward facing systems for infants. The conventional pattern for this system requires only the safety belt, lap belt or lap/shoulder belt to be secured in the car; see Figure 11.

This is of course very convenient when installing the child seat in the car, but the simple installation sometimes creates some disadvantages which cannot be neglected.

- The child seat becomes unstable, which can cause problems.
- If an accident should occur the safety belt will only absorb forces in one direction. This means that the infant seat will be well restrained during the beginning of a frontal impact. However, the safey belt cannot prevent the infant seat from rotating towards the seat back of the front/rear seat during the rebound of a frontal impact or during a rear end collision. This means that the infant's head will contact some part of the seat back. Even if the seat back is mostly made of soft material there is a chance that the

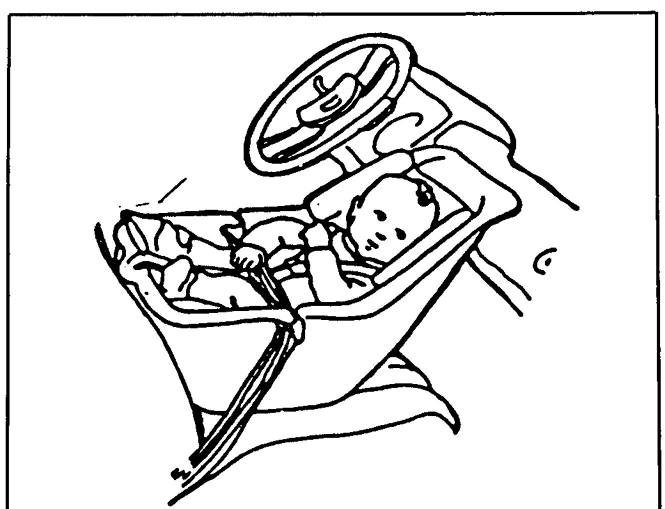

Figure 11. A conventional infant seat.

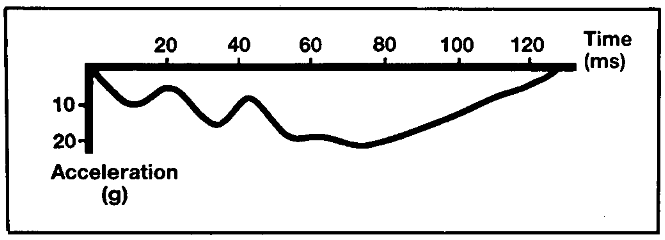

Figure 12. Sled acceleration pulse. Frontal crash simulation.

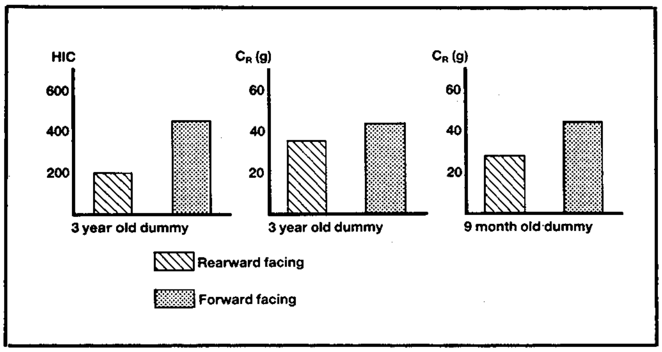

Figure 13. The staple diagrams show the difference in injury criteria between the forward and rearward facing position. A 3 year old American p572c dummy and a 9 month old European dummy were used. The 3 year old dummy shows a 57% lower HIC-value and a 20% lower chest acceleration when sitting facing the rear. The 9 month old dummy shows a 36% lower chest acceleration when sitting facing the rear.

child will hit some structural components in the seat back, the head restraints, or, if it is an angled collision, the B-pillar. Furthermore, in a multiple collision the infant seat can move around in the compartment in an uncontrolled way.

These disadvantages were taken into account in the design of the new combined infant and child seat.

Test Results

A comparison of the injury criteria has been carried out between the rearward and forward facing installation of the combined child seat. The tests were carried out in a Volvo 480 car body on a Hyge sled, with a frontal impact at 30 mph. The crash pulse is shown in Figure 12.

Design Concept

Volvo's Combined Infant and Child Seat

The child seat is intended for children weighing up to 18 kg, i.e. in the age-group from newborns to approx. 4 years old. It is designed to give the child a high crash protection in all kinds of accidents.

The seat is approved according to ECE 44, FMVSS 213 and CMVSS 213. It is possible to place the child seat in the rear seat or in the front seat, facing rearwards and forwards. We believe, however, that all children in this age group, as far as possible, should be transported in rearward facing child seats.

Figure 14. Rearward facing front seat. Infant and toddler position. The child seat is placed on the passenger front seat with its back leaning against the dashboard. It can be adjusted to the desired angle by sliding the front seat forwards or backwards. The child seat is secured by an extra strap together with the passenger safety belt

Figure 15. Rearward facing, rear seat
The child seat is placed on the rear seat with its back leaning against the back of one of the front seats. It can be adjusted to the desired angle by sliding the front seat forwards or backwards. The child seat is secured by an extra strap around the head restraint of the front seat together with the passenger safety belt

Figure 16. Forward facing
The child seat is secured with the ordinary seat belts of the car, i.e, the lap/shoulder belt or the lap belt

The installation of the child seat is very easy in both the rearward and forward facing positions.

A Half-integrated Child Seat

A new type of child seat which is the first step towards integrated child safety has been developed for children with a weight between 9 and 18 kg, i.e. in the age group of about 9 months to 4 years of age.

The child seat is installed on the rear of the front passenger seat and is intended for use in the Volvo 740/760 from model year 1985. This means that it is a car-specific child seat which can only be approved according to ECE 44.

The big advantage of the seat is that it can simply be folded out of the way when it is not in use. This

Figure 17. The half-integrated child seat. The child is secured by the front passenger safety belt

Figure 18. The booster cushion can be supported with a backrest. The backrest works as a head restraint when the child's head reaches above the edge of the seat back

means that it is possible to transport 5 adults without having the problem of removing the child seat.

Booster Cushion

When the child has grown out of the child seat, a booster cushion together with a lap/shoulder belt can be used. The booster cushion affords greater comfort for younger children when the seat belt is used and it eliminates the risk of submarining.

Conclusions

- Restraint use amongst children travelling in Volvo cars in Sweden has increased from 22% in 1976 to 72% in 1986. This is probably a result of intensive campaigns, the infant seat loan programme for newborn babies, the seat belt law for adults, increased public safety awareness.
- The effectiveness of all kinds of restraints for children (child seat, booster cushion, seat belt) is 16% for AIS 1 injuries, 47% for AIS 2-3 injuries and 60% for AIS 4-6 injuries.
- The effectiveness in reducing AIS 2-6 injuries to children in the age group 3-10 years is 58% for seat belts and 63% for booster cushions.
- The most common type of injury for both restrained and unrestrained children is the head injury.
- Severe neck and abdominal injuries caused by booster cushion use or seat belt use are almost non-existent among children in Volvo's accident material.
- Misuse frequency of rearward facing child seats is low.
- Although misuse of booster cushions is relatively high (approx. 40%), there is a clear effectiveness in reducing more severe injuries.
- We find from accident studies that frontal collisions are the more frequent types of collisions and are usually more severe than rear end impacts.

This means that a rearward facing child seat offers better protection to the child, since the crash forces are spread over the back, neck and head of the child. A further advantage with the rearward facing seat is that the risk of submarining is almost negligible.

Volvo's accident research shows that the rearward facing child seat has a very high injury-reducing effectiveness—90% for AIS 2-6 injuries.

Volvo's laboratory crash tests also show that although results and behaviour are good with the new forward facing seat, they are even further improved with the rearward facing one.

- It exists many different regulations on child safety. A harmonisation of the requirements is necessary to encourage car manufacturers and others to increase the development of "international" child safety.
- If a crotch strap is used it shall be fixed (the North American way) and not released at a certain force (the European way). No crotch injuries has been found either in North America or in England.

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Appendix

Case 1 (child seat)	The car was hit obliquely from behind by a truck. There was extensive deformation of the right rear end of the car. The belted female driver (injured AIS 1) and two occupants were travelling in the car. One of them was a 7 year old boy (injured AIS 2) who travelled unbelted in the left rear seat and the other was a 2 year old girl who travelled in a rearward facing child seat located in the right front seat. The girl in the child seat was unproperly restrained and was thrown backwards in the car. She sustained a facial laceration (AIS 1) and concussion (AIS 2).
Case 2 (cushion)	The car was hit in the right side by another car. The deformation was 2 according to the VDI-scale. A belted female driver was travelling in the old car (not injured) together with a 31 year old male (injured AIS 2) in the left rear seat with a 2 year old girl on his lap (injured AIS 1) and a 5 year old girl in the rear right seat using a booster cushion. She sustained an abrasion on the left eyebrow and on the chin (AIS 1) and concussion (AIS 2), probably caused by interaction with the adult rear occupant.
Case 3 (cushion)	The car skidded on a snowy road with the left side first into a big tree. The deformation was concentrated to the left side behind the B-pillar (VDI 3). A belted male driver was travelling in the car (uninjured), together with a 35 year old belted female right front seat passenger (uninjured), a 8 year old girl in the right rear seat using a booster cushion (injured AIS 1) and a 5 year old girl in the left rear seat also using a booster cushion. This girl sustained minor lacerations on the upper and lower extremities and a more severe concussion (AIS 3), probably caused by direct head impact to the tree.
Case 4 (cushion)	Another car was overtaking a lorry and caused a severe front end offset impact to the case vehicle. The deformation to the Volvo car was extensive and concentrated to the left side of the front (VDI 6). The occupants of the car were a belted 27 year old female who was killed, a belted 38 year old male in the right front seat (injured AIS 3), an unbelted 32 year old female in the right rear seat (injured AIS 4) and a 6 year old boy using a booster cushion in the left rear seat. The boy sustained a fractured right forearm (AIS 2) and a fractured right lower leg (AIS 2).
Case 5 (cushion)	The car skidded sideways and was hit in the right side by another car. The deformation was concentrated to the area of the right rear seat passenger (VDI 2). The occupants of the car were a belted 50 year old female driver (injured AIS 1), a belted 17 year old male front seat passenger (injured AIS 2) and a 6 year old girl using a booster cushion in the right rear seat. The girl's injuries were facial lacerations (AIS 1), a severe concussion (AIS 4) and right side rib fractures including lung contusion (AIS 3). The injuries were probably due to head impact with the right rear window frame and chest impact with the car side interior.

Modern Testing Techniques in Motor Vehicle Safety Research With Regard to Rear End Crash Properties

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Summary

Modern, lightweight three-door vehicle concepts, like the new Volvo 480, demand a special approach in the development of their rear end crash properties. With this type of car designers encounter special issues, mainly caused by the limited permissible deformation area in respect of the interior compartment, the fuel tank position, the spare wheel and other components.

In order to develop the Volvo 480 efficiently from the viewpoint of its rear end crash behaviour, different techniques were used. Some of these techniques are common practice in the automotive industry, others are less usual. The latter techniques are the subject of this paper.

Introduction

When addressing issues of design in the development phase of a new car, the design engineer will make full use of his or her automotive experience; data from computer simulation techniques will also be used, like the well-known finite element method, or modern test methods as described in this paper.

The various computer simulation techniques (linear-elastic or non-linear) have been extensively described in several outstanding papers and will therefore not be discussed further within the scope of this paper. An additional reason for limiting this paper to a description of test methods is that the F.E.M. calculations were restricted to the linear-elastic part of the deformation (see Fig. 1). Because of the large plastic deformations in crash tests, these calculations only cover the initial part of the crash.

At the start of our development testing phase, efforts concentrated mainly on optimizing the structural performance. It was decided that the best way of achieving this goal was to set up a sequential test programme based on the following principles:

- Perform full-scale crash testing to obtain reference data.
- Detail studies by component testing, both static and dynamic.
- Verify the results by full-scale crash testing.

Test Methods: The Approach Used by Volvo

The complete test programme

Very early in the development of the Volvo 480 an extensive programme of body tests was scheduled with the objective of gaining more information in less time and at lower cost by using component and semi-full-scale tests and a limited number of full-scale reference tests. The complete test programme comprised:

Dynamic tests (*crash* tests)	Full-scale. Semi-full-scale. Component.
Quasi-static tests (slow *crush* tests)	Semi-full-scale. Component.
Other tests not mentioned in this paper. .	

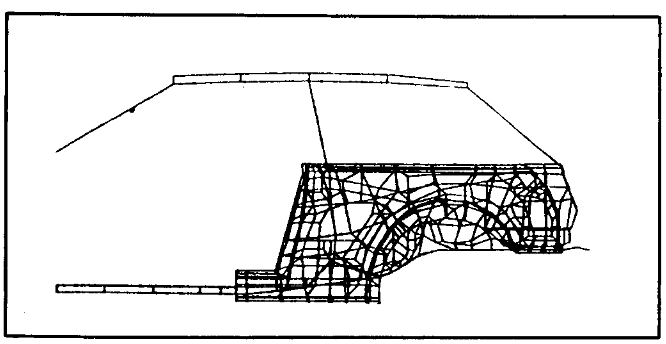

Figure 1. An example of a finite element model of the Volvo 480 body.

The first step consisted in obtaining reference data by a full-scale dynamic test. All other test methods were referenced to this full-scale test, in particular in order to establish a good simulation.

Fig. 2 shows the difference between straightforward development testing and the multipath approach adopted by Volvo in the development of the 480.

The main difference between the two methods lies in the amount of usable information collected compared with the number of tests. With the multipath approach, using a reduced number of full-scale crash tests, the same amount of information can be collected compared with the customary system; then again, on a detail level even more data can be collected from this reduced testing using the multipath approach.

Examples of Multipath Testing

Standard reference: full-scale crash test

The full-scale tests were conducted in accordance with the test method described in TP 219-02, although different test speeds were used. The information derived from these tests concerned:

- general body deformation
- fuel system integrity
- interior compartment performance
- interior trim performance
- seat belt performance
- fixation of accessories such as audio equipment and child safety seats
- injury criteria for all occupants

The advantages of the mobile barrier test according to TP 219.02 are:

- The high level of reproducibility due to the very simple geometric shape of the mobile barrier front. This means that improvements made in the design of the vehicle can be confirmed conclusively, whereas tests with

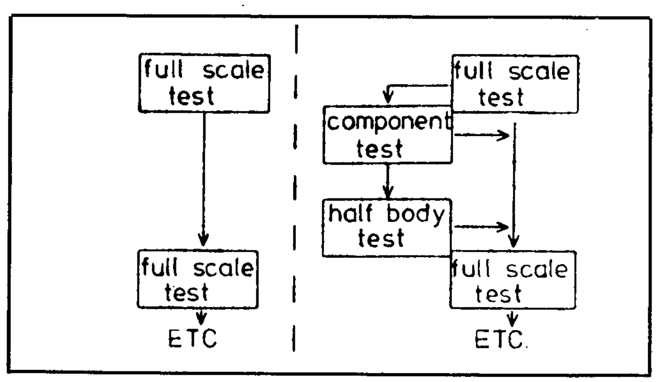

Figure 2. Illustration of conventional rear end crash development testing (l.h. side) and the multipath approach adopted by Volvo (r.h. side)

deformable barriers, or even car-to-car tests, may show some spread of results, especially when relatively small detail solutions are being investigated.

- The severity of the impacts is quite realistic for crash safety testing.
- The simple overall crash kinematics of both the car and the mobile barrier make high-speed filming possible in most areas.

The disadvantages of the mobile barrier test according to TP 219.02 are also obvious:

- High cost per test, since every test requires a complete prototype.
- These tests are not very realistic compared with the actual shape of normal passenger cars or trucks.

The first full-scale crash test was carried out with a complete prototype in order to establish the basic behaviour of the chosen principles. This test was used as reference for the subsequent tests; all other test methods used later were checked against this full-scale test in order to establish their validity.

Component testing

The component testing embraced crash tests both on simple components, like individual side members, and complex assemblies which together make up a complete body. Since bending and compression tests of simple side members are common practice in today's automotive industry, these tests will not be discussed here. Of more interest are the tests performed on complex parts and assemblies of the body.

The rear floor

Since the rear floor of the car is one of the most important structural parts for energy absorption in a rear end crash situation, many tests were performed on this section. The object of these tests was to determine the possibilities of influencing the energy absorption and deformation kinematics of this assembly. Accordingly, it was decided to isolate the rear floor section from its structural environment in order to establish its principal deformation mode. Both static and dynamic crash tests were then carried out on several rear floor sections.

The tests were carried out on a component mounted against a fixed barrier and then either crushed (quasi-static) or crashed (dynamic). The method used to mount the component against the fixed barrier was found to be very important since the mountings should not introduce any unrealistic stress concentrations in the floor section and consequently provoke failure at unintended points.

Even if the mountings are chosen with care, one should still be aware of their influence. The component parts of the rear floor sections used in the tests can be regarded as two box-sections (the side members) interconnected by a transverse diaphragm (the floor). Theory(1) shows that the influence of the mountings is of no specific importance at a distance of approximately 1 to 2 times the height (h) of the side members.

This means that, providing the mounting are chosen with care, any deformation occurring from h to 2h away from the mountings will not be due to the mountings but to the actual test.

The most important requirement to be met by the mountings is that they only restrict the movement of the component comparable to the degree of freedom of the component in the full-scale test situation. The easiest way to comply with this is to select the floor sections for the test so that the mountings coincide with the places where no relative movements occur in the plane of the cross-section during the full-scale test.

By carefully selecting the mounting points in this way, it becomes possible to weld the component to a 5 mm thick plate that can then be fastened to the fixed barrier. The mounting orientation of the component should be in accordance with the orientation of the component on the car during the full-scale test. If the orientation of the cross-section plane changes during impact, it should also be made to change during the component test. Fortunately this can be avoided in most cases by careful choice of the cross-section plane. Many tests have shown that this way of mounting the component is suitable for this type of testing.

Quasi-static testing

Quasi-static testing (crush testing) is done by crushing a test sample at a very low deformation speed (approximately 10 mm/s overall deformation) with prescribed displacement approach.

In crush tests the dynamic properties of the test materials are left out of the test measurements. In most of our tests the material was sheet steel, a material that has the property of developing higher tensile strength and yield strength at higher strain rates.

In the actual crush process this means that, because of the strain rate effect, the forces imposed on the complete structure will be lower in the crush test than in the actual crash test. Other dynamic effects—like stress waves and inertia effects—are also left out of the crush test protocol. However, when evaluating the test results these phenomena should always be borne in mind.

If the test component shows a wanted and expected behaviour in the crush test (i.e. quasi-static), it cannot simply be assumed that it will perform similarly in the crash test (i.e. dynamic). Nevertheless, after many tests it was established that the actual difference in deformation mode between crash tests and crush tests is very small. Crush testing was found to be a very useful and relatively economical tool in the development of the Volvo 480.

The greatest advantage of component crush testing, however, is the ability to isolate the component from its structural environment. This offers the tester diverse opportunities of obtaining more information out of the component test than the information resulting from a full-scale test. The advantages may be summarized as follows:

- Isolating the component from its structural environment shows the component's natural behaviour, and this is essential for its crash performance.
- Isolation of the component and the low deformation speed make numerous measurement techniques possible, such as still photography, accurate stress measurement (e.g. using strain gauges), inside filming, crush force measurements, etc.
- The low deformation speed allows close and protracted observation by engineers and designers. In dynamic testing this is not possible; visual examination in such tests is limited to image recording by high-speed cameras.
- As the deformation speed is almost zero, the test can be interrupted at any time without influencing the test result. This offers the possibility, for example, of repairing or reinforcing the component during the test. The need for this may arise if parts of the structure start to fail prematurely.
- These tests are relatively inexpensive on account of the simple test set-up and the fact that there is no need for high-speed measuring techniques.
- Flexibility of execution is a favourable characteristics of crush testing. It is possible to test four to five test samples each day, whereas a single dynamic test may take several days to prepare and complete.
- Less prototype building capacity is used for components compared with complete prototypes. The tests are therefore less time-consuming, both before and during the test.
- The measuring techniques are less troublesome and more accurate than those customarily used in high-speed dynamic testing.

Dynamic component testing

Dynamic component testing, or component crash testing, is a method that is used in an attempt to investigate the influences of the strain rate-dependent mechanical properties of steel. With this type of dynamic testing the advantages of component crush testing, as described in the previous paragraph, are retained and the dynamic tests are done in a similar way to the crush tests.

However, some differences should be noted. The greatest difference is found in the input used in the test protocol. In the crush tests the input is the prescribed deformation. Forces and tensions as well as detail deformations are also measured.

In the case of dynamic tests, however, the input is energy. Unlike the crush tests, which can be interrupted at will, the overall deformation in dynamic testing cannot be influenced once the test has started. Forces, tensions and detail deformations can, however, still be measured during the crash, but the accuracy is much lower.

The great difficulty with component crash testing lies in the modelling of the boundary conditions. Although energy is the main input, the objective is to make deformation speeds as realistic as possible. This can cause problems, as indicated below. If the test component belongs entirely to the deforming part of the car, this means that this component may never experience a strain rate equalling zero.

In the component crash test, however, the impacting barrier must at some time have a zero relative velocity, thus making the strain rate equal zero as well. As a consequence, it is not possible to test the component over its full length since space is needed in which to arrest the mobile barrier and space to accommodate the compressed material. Accordingly, only a limited period of initial deformation can be investigated and compared with the crush test.

In order to make deformation speeds as realistic as possible during the component crash test, a set of boundary conditions has to be created for the test. Before starting the test, a careful estimate should be made of the crash pulse of the component and its environment (now missing) and with these two estimates a correct barrier weight must be chosen. An estimate of the barrier velocity upon impact with the component during a full-scale crash test should likewise be made.

If, for example, the test component consists of the rearward section of the rear floor complete with bumper, this will be relatively simple because the barrier velocity will then be the actual impact velocity. With this information in respect to barrier velocity, barrier weight and both crash pulses, a proper set of boundary conditions can be created for the test. This is done by adding extra braking devices alongside the test component. These braking devices, such as crumpling tubes or honeycomb blocks, adjust the deformation speeds to the correct values during the test period of the crash, simulating the parts of the car (the structural environment) that have been left off.

A second braking device comes into action during the braking period of the test. As mentioned earlier, the component in test will be too short to stop the mobile barrier completely during the test period. This second braking devices (a second set of crumpling tubes or honeycomb blocks) is therefore needed to dissipate the residual energy.

Although some disadvantages are inherent to this type of test—we have already mentioned the limited crush area of the component, the problems of high-speed testing in general and the difficult modelling of the boundary conditions—these tests also offer considerable advantages:

- The cost of these tests is still much lower than the cost of a full-scale test (although higher than crush tests). This makes it possible to test more detail solutions (like trigger shapes, influences of the spare wheel and brackets) to a relatively high degree of accuracy.
- The previously mentioned advantage, i.e. that of knowing and understanding the component's natural behaviour, still applies.
- Prototype building and test execution take less time than full-scale testing.

Semi-full-scale testing

After determining the behaviour of the components, the next step is to study their interaction with their environment. This was done by semi-full-scale testing, also known as half-body testing. As the name of this method suggests, these tests involve the crushing or crashing of half-bodies of cars, in this case the rear ends of bodies cut just in front of the B-pillar. Half-body testing is in fact a high level of component testing with some typical features of full-scale testing, as will be shown later.

Half-body crush testing (quasi-static semi-full-scale testing)

For the half-body crush tests the rear end of a body was mounted against a fixed load cell barrier and the loads were measured at the points where the body was supported: the sills, roof cant rails and at waist-line level. This made it possible to determine the load paths inside the body during the crushing process. The body supports were made in such a way that they did not introduce undesired loads in the car.

The first half-body to be tested was a reference body. This body, which had the same structure as the body used in the initial full-scale crash test (the reference test ex paragraph 2), was crushed to determine the validity of the test method.

In these tests it was found that pin-jointing the half-body to the fixed-barrier was an acceptable alternative.

The results of the half-body reference test proved to be quite comparable to the full-scale reference test. While this applies especially to the failure mode of the body, it is also true of the force deflection characteristics, as will be shown later.

All the advantages and disadvantages mentioned for the component crush tests were found to be valid for the half-body crush test.

Half-body crash testing (semi-full-scale dynamic testing)

The final stage before full-scale dynamic testing is semi-full-scale testing, also known as half-body crash testing. This type of testing could be regarded as the most complete component testing possible.

As in the half-body crush tests, this type of testing uses a rear end of the car which is mounted against a fixed load cell barrier.

Having already completed the half-body crush testing, it was now known that the body could be clamped against the load cells so long as the body section was cut in front of the B-pillar. Pin-jointing the body to the load cell barrier was not tried since it was not certain whether the pin joints would be capable of resisting the dynamic effects of the tests. Also, since the bodies clamped to the load cell barrier gave valid results, it was decided to stay with this practice.

The purpose of half-body crash testing is to obtain—at relatively low cost—a good impression of the car's rear end crash performance together with additional information on load paths.

By choosing the weight of the mobile barrier in the half-body crash test so that(1)

$$M_b^* = \frac{M_b \quad M_c}{M_b + M_c} \qquad \text{, in which}$$

M_b^* = mobile barrier mass in half-body test; M_b = mobile barrier mass in full-scale test; M_c = mass of the car in full-scale test, it can be proven that the half-body test will be equivalent to the full-scale test within an accuracy of 95% for all test parameters. The initial test speed of the mobile barrier in the half-body test must be the same as in the full-scale test.

Comparison of Test Results

When comparing the half-body test results with those of the full-scale crash tests, the following trends are found.

Half-body crush test vs. half-body crash test

Fig. 3 shows the force deflection curves of a half-body crush test and an identical half-body crash test to be comparable. It can further be seen that upon initial impact the strain rate is high in the

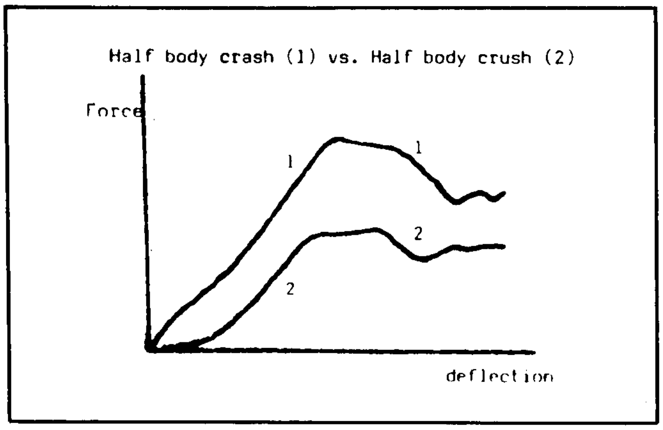

Figure 3. Comparison of test results of a half-body crush test and a half-body crash test showing the correlation between the two tests.

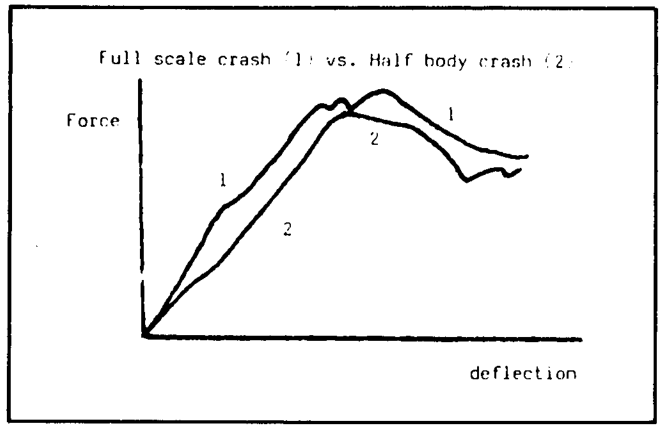

Figure 5. Comparison of test results of a half-body crash test and a full-scale crash test.

dynamic test, thus generating higher forces compared with the quasi-static tests. As the deformation speed reduces after initial impact, the differences between crash and crush test diminish. This, of course, was to be expected.

An interesting result of these tests are the load distribution through the body. The loads are plotted as a percentage of the total load against deformation of the rear panel. This is shown in Fig. 4, where it can be seen that the half-body crush test gives the same information as the half-body crash test. That seems to indicate that the crush test is a good enough simulation of the crash test for determining load paths inside the body.

Half-body crash test vs. full-scale crash test

Fig. 5 shows atypical curves for a half-body crash test and a full-scale crash test. As can be seen, the graphs show a good correlation. The results of the half-body crash tests were found to be very useful in the development phase of the Volvo 480, because they

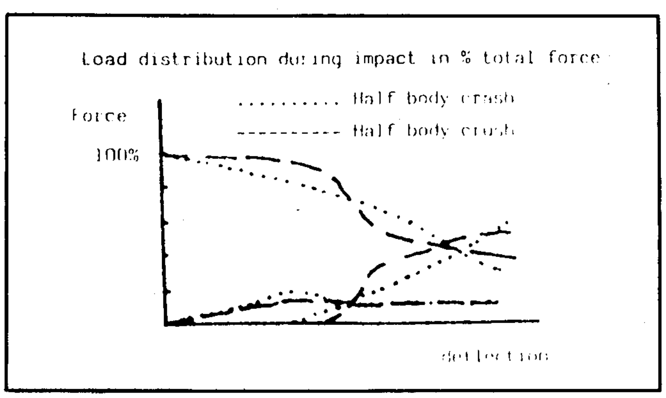

Figure 4. Comparison of test results of a half-body crush test(1) and a half-body crash test(2) showing the load distributions at roof, waist and floor levels.

gave a useful indication of the car's crash behaviour and energy absorption properties.

Future Developments

As the costs involved in a half-body crash test are much lower than those of a full-scale rear impact test, and since the results obtained with the former method are shown to be valid, it has become possible to carry out other component tests on half-bodies. This means that items like the influence of the type of mobile barrier used can easily be checked in half-body tests.

This is done by replacing the flat rigid barrier surface by a deformable barrier. For this purpose a CCMC side impact barrier front is used(5). This barrier front is chosen as being more reproducible than frontal sections of rear cars since the latter may vary in frontal crash stiffness.

The test set-up is identical to the set-up for half-body crash tests and the choice of boundary conditions is also the same. The only change made is to raise the test speed. This is done in view of the energy-absorbing property of the deformable barrier.

It is not possible to determine the exact distribution of the energy dissipation between the target and the impactor, but a good estimate can be made by evaluating the static deformation of the car after the test. However, this is only possible when there is sufficient data available in respect of dynamic body deformation, as is the case with the Volvo 480.

As this test method is still in a development phase, this subject will not be addressed in detail in this paper. We do however intend to pursue this line of investigation because we believe that it can be developed into a useful method for determining dynamic crash performance.

Conclusions

While there are certain difficulties in making detailed comparisons between crash results and crush results, even so the following conclusions can be drawn from the experience gained in the development of the Volvo 480:

1. The *deformation modes* of components and complex structures (half-bodies), both in a crash test and a crush test, are very comparable. Crush testing therefore provides a good prediction of the failure mode when special attention is given to modelling the boundary conditions.
2. The *overall force distribution* in the body during the total deformation distance is independent of crash/crush speed.
3. Isolation of the component in component tests and low-speed deformation offer the possibility of making detailed studies and using numerous measuring techniques, including—inter alia—strain gauges, force measurement and still photography.
4. The cost of these tests is relatively low in comparison with extensive full-scale testing, and comparable results are generated.
5. When crush testing is considered as a means of obtaining a major *indication* of failure modes in the structure, then it is a very useful tool in the development of rear end crash properties.
6. The multipath method offers far greater flexibility for comparing different alternatives.

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Technical Session Six

Heavy Duty Vehicle Safety

Chairman: Dr. Lennart Strandberg, Sweden

Large Truck Accident Exposure in the U.S.

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Abstract

In order to assess large truck safety, information is needed on vehicle *exposure* to accidents (travel) as well as on *involvement* in accidents. Since 1979, the University of Michigan Transportation Research Institute has collected detailed data on every fatal large truck accident in the U.S., about 5,000 cases a year. With the completion of the first comprehensive large truck exposure data survey, by UMTRI, we will be able to go beyond past analyses of large truck accidents. The National Truck Trip Information Survey (NTTIS) was initiated in late 1985. Data collection was completed in February 1987. For this survey the owners of over 4,000 large trucks were contacted four times over a twelve-month period (16,000 survey days) to obtain detailed information on the use of the truck on a randomly-selected survey date. The information collected includes the configuration, cargo, actual weight, and the route the truck followed. The combination of accident data with miles traveled from NTTIS will enable the calculation of fatal accident involvement *rates* by vehicle type, road class, etc.

Background

As the U.S. economy came out of the recession of 1974 and '75, truck mileage increased. With the increase of mileage came an increase in fatal truck accidents. The subject of large truck accidents became a matter of increasing public concern. This concern has continued and intensified as the U.S. moved to economically deregulate motor carriers, with many claiming that lower freight rates and increased competition led to decreased concern with safety. Although table 1 shows that the increase in combination truck fatalities and fatality rate abated after 1979, it also shows that the combination truck fatality rate continues to run about twice that of all vehicles in the U.S.

Although manufacturers, carriers and government saw the number of fatalities increase between 1976 and 1979 they did not know the specific reasons for the increase. Information on large truck accidents was fragmented, incomplete, and, in many cases, simply unavailable or so inaccurate as to be little value in determining accident causes. Yet accurate accident data are the key to finding effective solutions to truck accidents. Without good data, there is an excellent chance that solutions based on intuition alone may be suggested. Such solutions may impose high costs without providing actual improvements.

Truck manufacturers believed that a comprehensive, coordinated research effort—involving government, industry, the academic and scientific communities—was required to collect and analyze in-depth data on large truck accidents. Such an effort would include collection of miles traveled (exposure) by different kinds of trucks, and could be used to identify the role

Table 1. Fatalities & fatality rates (U.S.).

Year	Combination Truck Fatalities	Combination Truck Mileage ($\times$ 10^6)	Combination Truck Fatalities/10^8 Miles	All Vehicles Fatalities/10^8 Miles
1976	3,909	57,937	6.75	3.25
1977	4,198	61,179	6.86	3.26
1978	4,643	65,636	7.07	3.26
1979	4,950	66,313	7.46	3.34
1980	4,238	67,386	6.29	3.34
1981	4,388	69,388	6.32	3.17
1982	3,911	71,129	5.50	2.76
1983	4,079	73,562	5.54	2.57
1984	4,257	76,986	5.53	2.58
1985	4,650*	79,402	5.86	2.47

*preliminary

of vehicle designs, the effects of speed differentials, and other factors that had not been investigated in a systematic manner.

The Motor Vehicle Manufacturers Association and the Western Highway Institute took the initiative by sponsoring a comprehensive study of large truck accidents in 1979. The American Trucking Associations joined later. They sponsored the University of Michigan Transportation Research Institute (UMTRI) in conducting this pioneering study and set the following goals:

- Determine the accident, injury and fatality rates (in terms of events per vehicle-mile, ton-mile and/or cube-mile) for a broad range of heavy trucks operating on U.S. highways. These should include at least comparisons among straight trucks, tractor-trailers, doubles, and triples; cabover vs. conventional designs; and, combinations of various lengths.
- Determine the causes of accidents involving heavy trucks.
- Achieve an understanding of the possible countermeasures which are likely to prevent or reduce the frequency of such accidents.

The study is called "Acquisition/Analysis of Truck Accident and Exposure Information." Phase I (completed in 1979) assessed the status of available data and determined what was needed to conduct a comprehensive analysis of medium and heavy truck accidents on a national scale. It recognized the need for both accurate accident counts *and* mileage at comparable levels of detail if accident *rates* were to be determined. It pointed out the fallacy of comparisons which used simply "vehicle miles" but failed to recognize that, for example, a far greater percentage of truck miles than passenger car miles were accumulated on rural roads and at night. UMTRI concluded that "The availability and the nature of present . . . accident and exposure information . . . in much detail is rather limited." and outlined a program which would provide information to fill these gaps.

Accident Data

Phase II of the UMTRI study evaluates all large (over 10,000 pounds gross vehicle weight rating) truck fatal accidents. Initial information is taken from the National Highway Traffic Safety Administration's Fatal Accident Reporting System (FARS), and augmented with police reports and information from the Federal Highway Administration's detailed files on accidents in interstate commerce. When Federal Highway Administration accident reports are not available researchers go directly to vehicle owners, motor carriers, and drivers. This permits researchers to determine factors such as kind of road, weather, cargo, trip, weight, length, ownership, time of day and vehicle configuration. UMTRI publishes these analyses on an annual basis in the form of a factbook entitled "Trucks Involved in Fatal Accidents."

These yearly factbooks, providing data on fatal accidents beginning in 1980, and special studies have already yielded insights, for example:

- Rollover is involved in almost 60% of accidents fatal to combination vehicle drivers, ejection in 34%. Extrication is involved in 22% (these are accidents in which the victim must be physically removed from the damaged vehicle; in some of these cases, the victim was crushed within the truck cab) and fire in 16% (these figures do not add to 100% since more than one may be involved in a single fatal accident).
- The greatest number of fatal accidents take place on non-Interstate rural roads (54% of the total). Five-sixths of these are on two-lane rural roads. Only 25% of the fatal accidents take place on Interstate Highways, both urban and rural, yet combination trucks operate 43% of their miles on the Interstates. A tractor-semitrailer is about three times as likely to have a fatal accident on a rural 2-lane road as on an Interstate. Because vehicles are traveling at high speeds in opposite directions on two-lane rural roads, accidents can be more frequent and more serious than on Interstates or other divided multi-lane highways.
- Tractor-semitrailer combinations are involved in 72% of heavy truck fatal accidents, single unit vehicles in 21%, doubles or triples in 3% and bobtailed tractors in 2%.
- Tractor-semitrailers have as many fatal accidents at dawn, dusk and night combined as they do in daylight.
- Tank trailer combinations are about twice as prone to rollover as are van trailer combinations.
- Rollovers are almost 50% more likely in fatal accidents on dry than on wet pavements while jackknives are almost twice as likely on wet pavements.

Data on all medium and heavy truck fatal accidents in the United States from 1980 through 1984 are now on the University of Michigan's computer system undergoing analysis. Published data code books provide tabulations such as those illustrated in table 2 below from "Trucks Involved in Fatal Accidents, 1983":

Data in the form shown provide basic information which can be analyzed to answer specific safety questions, for example the seriousness of the drunken

Table 2. Sample code book data tabulations: "Trucks Involved in Fatal Accidents, 1983".

Variable 207	DRIVER DRINKING
FREQ Prcnt	DRIVER DRINKING
4716 95.4	0. No drinking reported
228 4.6	1. Drinking reported
0 0.0	9. Unknown
Variable 210	LICENSE STATUS
FREQ Prcnt	LICENSE STATUS
0 0.0	0. None required
132 2.7	1. None
4477 90.6	2. Valid
92 1.9	3. Suspended
17 0.3	4. Revoked
19 0.4	5. Expired
1 0.0	6. Cancelled or denied
2 0.0	7. Learner's permit
1 0.0	8. Temporary
203 4.1	9. Unknown
Variable 1028	CAB STYLE
FREQ Prcnt	CAB STYLE
2628 53.2	1. Conventional
2195 44.4	2. Cabover or cab-forward
121 2.4	9. Unknown

driving problem among truck drivers, compared to the general driving population.

In a study of truck crashworthiness now underway, UMTRI is examining the role of cab deformation in truck occupant fatalities and identifying specific ways in which occupants are injured in the cab. The discovery that 16% of truck occupant fatality accidents involved fire in some way has led to in-depth investigations of accidents involving fire or fuel spillage to determine the role the truck fuel system plays in these accidents.

It will be necessary to continue the accident data collection which has now been underway since 1980, so that both long and short term safety trends can be identified. This will also allow us to see the long term results of actions taken to improve the truck safety picture.

Exposure

Only a limited amount can be learned by studying fatal accidents alone. One needs to know something about accident frequency; how often do accidents of a certain type happen in relation to vehicle miles traveled; on what type of highways; and at what time of day or year. If accidents of a certain type are frequent there is a greater benefit to investigating them and finding a solution. And if a certain type of accident is found far more frequently on one type of road or during a particular time of day, a clue toward its cause may already exist.

Phase III of UMTRI's overall study, called the National Truck Trip Information Survey (NTTIS) is providing information which allows accident frequency to be calculated. The truck trip survey provides information on truck population and exposure (miles traveled with detail comparable to that collected on fatal accidents).

In the past students of truck accident exposure in the United States have gone to the Truck Inventory and Use Survey (TIUS), conducted every five years by the Bureau of the Census. Data for this survey are obtained by requiring a sampling of truck owners to fill out a form providing annual mileage and typical use of the vehicle. When the owner notes on the form that the truck is "typically" used in over-the-road service, other uses, such as mileage accumulated in city delivery, are not included.

The NTTIS focuses on all mileage during a single day's use in order to get more accurate and detailed mileage data. The day's mileage is categorized according to road class (interstate, major artery, or other), rural versus urban, and day versus night. Urban areas are identified according to the FHWA definition. Two sizes of urban area are distinguished: areas with 50,000 population or more, and areas with 5,000 to 49,999 population. The NTTIS also categorizes milege by carrier operating authority, vehicle configuration (cab style, number of axles, trailer type and body), length, cargo, cargo weight, and gross combination weight. This will provide information on truck use at a level of detail previously unavailable in the U.S.

To provide a proper sample of trucks for the survey, the University of Michigan worked with the R.L. Polk company, the only organization in the U.S. which collects information on vehicle registrations directly from the individual states. A stratified random sample of trucks registered as of July 1, 1983, was chosen. The Bureau of the Census also uses R.L. Polk in setting up its sample for the Truck Inventory and Use Survey. Based on both funds available and a statistical evaluation of the level of data accuracy which might be expected a target sample size was chosen. Owners of 2,000 tractors and 2,000 straight trucks were telephoned four times within the last year, providing 16,000 survey-days of truck exposure information. Since the operation of double trailer combinations is of concern, but still relatively rare in the U.S. the states of California and Michigan, where the majority of current doubles operation can be found, were oversampled.

It is recognized that the National Truck Trip Information Survey will leave many questions unanswered. There is the obvious statistical error in a sample of the size being used. UMTRI has calculated that we can expect almost a 9% error in average annual mileage for a category of vehicles comprising one-fourth of our total, and a 20% error for a category comprising 5% of our total. For example since sales of low-bed trailers run about 5% of total trailer sales, we could expect to find a 20% error in our calculation of their annual mileage.

Beyond statistical error, the study is restricted to vehicles registered in 1983 while the survey of owners took place in 1985 and 1986. This was necessary because good vehicle registration data runs that far behind the current year. So most 1984, 1985, and 1986 model year vehicles are excluded from the survey. Since later model vehicles are more likely to be used for longer mileage, over-the-road operation, and move to shorter mileage and local operation as they age, the survey should show lower annual mileages and a higher percentage of local and shorthaul operation than is actually the case in the total U.S. fleet.

Finally, and perhaps most importantly, as soon as the collected data is analyzed, many questions are sure to be found which these data do not answer. Already the questionnaire was modified to provide more information on rural versus urban operation of vehicles and the type of engines and fuel-saving equipment in use to answer questions raised by new U.S. emission regulations. One must be reconciled to this sort of problem, knowing that good research often asks more questions than could be expected when the project was begun.

Although UMTRI has completed the data collection phase of the National Truck Trip Information Survey, substantial time will be needed to complete the analysis of the data before it can begin to provide answers to many questions which will help us understand the nature of the U.S. truck fleet and pinpoint areas where safety improvements might be made.

One of the earliest answers which *has* come from NTTIS is an accurate estimate of the actual population of large trucks (over 10,000 pounds gross vehicle weight rating) in the United States in the year 1982. Earlier population data, based on the 1977 TIUS and the Federal Highway Administration had been at variance—no one actually knew how many large trucks there were in the U.S.! Changes were made in the TIUS calculation procedure between 1977 and 1982 so there should be reasonable agreement between the NTTIS and TIUS numbers on nationwide truck population, since they came from surveys drawn from similar R.L. Polk samples. Table 3 shows the results.

The NTTIS telephone survey has shown a relatively low proportion of trucks actually in use on any given day. This information must be tempered by remembering that the three latest model year trucks (1984-86), those most likely to be in service more days and more miles, are not included in NTTIS as noted above. Survey data are shown in Tables 4 and 5.

Table 3. Estimates of the U.S. large truck (over 10,000 lbs. GVWR) population*.

Truck Type	Source	
	TIUS	NTTIS
Straight Truck	2,608,300	2,068,495
Tractors	876,700	886,643
Total	3,505,000	2,955,138

*Excluding Alaska, Hawaii and Oklahoma

Table 4. Truck-tractors in use.

	Weekday	Weekend
in use	49.1%	10.8%
not in use	50.9%	89.2%

Table 5. Straight trucks in use.

	Weekday	Weekend
in use	37.9%	8.3%
not in use	62.1%	91.7%

The Future

The combination of accident data with miles traveled from the NTTIS will enable the calculation of accident involvement rates by vehicle type, road class, etc. New and important insights into the causes, and hopefully, therefore the solutions to many large truck safety problems, will be found. Yet it is easy to see there is much left to be completed. The various segments of the trucking industry, labor, insurance and government can all capitalize on the investment already made by providing sustained financial support for a comprehensive plan of action. Such a plan would include:

First	The University of Michigan's fatal accident data collection program would be continued on an annual basis. If it can be more closely allied with the Federal Highway Administration's accident reports and data collection than it is now, it can provide both the government and industry with information of better quality and completeness than it now has for analysis purposes.
Second	The National Truck Trip Information Survey exposure study is collecting data for only one year. To meet future accident data needs, the survey should be continued on an annual basis. Again, such a program would supplement both government and industry knowledge and avoid duplication of effort.

Third	While the truck accident files created at the University of Michigan are the best available, they are limited to fatal accidents only. But fatal accidents represent just over one percent of all large truck accidents. The overwhelming majority of truck accident don't kill people, but they do injure people, damage property, snarl traffic and sometimes, release hazardous materials. These accidents are probably as much responsible for the industry's poor safety reputation as are those in which people are killed. So it makes sense to collect and analyze data on the 99 percent of truck accidents that are non-fatal. Although NHTSA's National Accident Sampling System was originally conceived to provide this kind of data (for autos and trucks), it is being cut back substantially and will not provide the information needed on heavy truck safety.
Fourth	Analysis of the data already collected and of that which will be acquired in the future must be performed if the information is to be of use to decision-makers, safety officials, designers and operators. To date, very limited analyses have been made of these data. A great deal more can be done, because the information is now available, particularly for fatal truck accidents, in sufficient detail to isolate patterns of high frequency occurrences.
Fifth	The final element of the plan is the performance of special, in-depth, studies. One example of a special study which should be done concerns the accident experience of longer combination vehicles. The need for this study was pointed out in two Federal Highway Administration Studies of the benefits and problems of such vehicles. In-depth studies, such as those mentioned above on truck crashworthiness and fire and fuel spillage, can be used to pinpoint problems that are already identified, or those which show up in accident studies.

Having come this far, U.S. industry and government should now be prepared to take the next step. The trucking industry is on the threshold of having a great resource at its disposal to help find solutions to truck accidents. But it will be wasted if we fail to continue to underwrite the costs necessary to continue collecting and analyzing accident data and making truck exposure surveys.

It will take more than just one or two organizations to do the job. Government, insurance and academic interests should be involved in forming a coalition with manufacturers, suppliers, motor carriers, and other interested groups to provide sustained financing for this program.

As I see it, the benefits are many. We shall have facts to counter growing criticism of the industry's safety record; the industry will have a solid basis for arguing for greater productivity in equipment and operations; and, it makes good sense to prepare now for truck safety issues that may become the subject of regulations in the future.

Perhaps the greatest benefit—the one the public is most concerned about—is that we will finally have the information needed to learn how to decrease the number of collisions between large trucks and passenger cars. That alone will make the whole effort worthwhile.

Vehicle Factors in Accidents Involving Medium and Heavy Trucks

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Abstract

Among the many interrelated causes of truck accidents, vehicle-related topics play a critical, if somewhat unrecognized and underreported role. In many cases, these factors, if they do not directly cause an accident to occur, make it more difficult—or in some cases, impossible—for a driver to recover from an error or avoid an unforeseen conflict. Once a crash occurs, the way trucks are designed can affect the severity of the trauma sustained by the occupants of all the vehicles involved.

This paper highlights the fact that efforts to prevent truck accidents could be substantially aided by working to upgrade the performance of the truck brake systems as well as truck handling and stability properties—especially as it relates to their tendency to rollover. Truck occupant crash protection could be enhanced by improving truck occupant restraint systems, providing a reasonable amount of protection from post-crash fires, making cab interiors free of sharp, hard objects that can cause injury during impact—especially steering wheel rims and hubs, and by improving cab designs to provide occupant survival space in a crash. Finally, an opportunity also exists—by working on the designs of the front ends of trucks—to reduce the number of fatalities among occupants of other vehicles killed in collisions with medium and heavy trucks.

Introduction

This paper summarizes two recently completed Congressional reports (Sections 216 and 217 of the Motor Carrier Safety Act of 1984, P.L. 98-554, October 30, 1984)[1][2] on the general topic of medium/heavy truck safety. The two reports focus primarily on vehicle-related issues that influence the safety performance of medium and heavy trucks. These include: braking, handling and stability, crashworthiness, and truck occupant crash protection.

Each report identifies the key issues related to each of these topics, summarizes what is known about each, describes what might be done in the near term to make improvements, and lays out research agendas for the remaining longer term issues.

The reports were developed with the assistance and participation of the complete range of interests associated with trucking, including: truck and trailer manufacturers; truck operators; driver groups; state, federal and foreign government research and regulatory organizations; representatives from the vehicle inspection and traffic law enforcement community; and, representatives of safety advocacy organizations. The Society of Automotive Engineers sponsored, in cooperation with the National Highway Traffic Safety Administration, a public symposium at which draft versions of the reports and the research plans were discussed and critiqued by all these interests. Consensus was reached that the reports had identified the key vehicle-related safety issues facing the trucking industry and that the research proposals contained in the reports reflect the best way of addressing those issues.

Priorities for addressing these subject areas should be dictated by the size of the accident problem affected by each and the availability of achievable solutions. For this reason, efforts to improve truck brake systems should receive the highest priority since improvements achieved there are likely to be significant. Regardless of efforts to prioritize these topics, each plan represents a technically sound, consensus approach as to how each of the topics could be pursued, given that priorities and resources are allocated to that subject.

The plans that are presented are not an exclusive agenda for government. Rather, it is hoped that they will serve as a blueprint that government and industry can use to address the topics discussed.

U.S. Medium and Heavy Truck Accident Patterns*

Heavy truck accidents are complex, often lethal events that have many interrelated underlying causes. They include factors related to driver performance/behavior, vehicle performance capability and condition, operating environment, and the amount and quality of safety management exercised by the motor carrier responsible for the driver and vehicle. Accidents occur when the "margin of safety" is reduced because the performance of one or more of these factors is low and compensation by the driver and/or the vehicle cannot be or is not made. A balanced heavy truck safety improvement program, if it is to be effective, needs to be cognizant of the relationships among all these factors and must incorporate elements that simultaneously address all of them in some reasonable fashion.

*Bracketed numbers indicate references given at the end of the paper.

*Unless otherwise noted, the accident statistics cited throughout this paper were derived from: NHTSA's Fatal Accident Reporting System (FARS) and the National Accident Sampling System (NASS) for 1984, and from the States of Washington and Texas for 1981-1983.

Viewed comparatively, medium and heavy trucks are involved in a relatively small proportion of the overall number of motor vehicle accidents which occur each year (382,736 trucks were involved in accidents in 1984—3.8 percent of the total). On the other hand, because of their size and a number of other factors, when they do become involved in accidents, they are often severe.

As a result, their proportional involvement in fatal accidents is higher (5,188 trucks were involved in fatal accidents in 1984—8.9 percent of the total). Normalizing for exposure, heavy trucks experience less non-fatal accidents per 100 million miles of travel than do passenger cars (288 versus 614, in 1984), but experience more fatal accident involvements per 100 million miles of travel (4.0 versus 2.8, in 1984) than passenger cars.

One way of gauging the relative importance of medium and heavy truck safety is to assess the consequences of these vehicles' accidents in terms of the total number of fatalities and injuries that result. Viewed in this way, medium and heavy truck accidents result in 12.8 percent (5,657) of all highway related fatalities and 4.8 percent (171,232) of the injuries that occur in highway related accidents each year. The majority of these (118,835 of the injuries and 4,019 of the fatalities) were sustained by occupants of other vehicles involved in collisions with medium and heavy trucks (see Table 1).

Truck drivers are involved in one of the nation's most hazardous occupations. They sustain 9.3 percent of all work-related fatalities, yet comprise only 1.8 percent of the employed work force. Truck driving ranks second only to mining and quarrying in terms of occupational fatalities that are sustained per 100,000 workers per year (see Table 2).

If a medium or heavy truck is involved in an accident it is most likely to be a collision with another motor vehicle. This pattern is typical for most other vehicles as well. Trucks are, however, proportionally more involved in single-vehicle accidents (rollover, loss-of-control/jackknifes, and collisions with road-

Table 1. Consequences of medium and heavy truck accidents in 1984.

	Killed	Injured
Medium and Heavy Truck Occupants	1,087	42,999
Occupants of Other Vehicles Involved in Collisions with Medium and Heavy Trucks	4,019	118,835
Pedestrians/Cyclists Involved in Accidents with Medium and Heavy Trucks	551	9,398
Total	5,657	171,232
Total (all highway related accidents)	44,241	3,573,210
	12.8% of all Fatalities	
	4.8% of all Injuries	

SOURCES: FARS 1984 and NASS 1984

Table 2. Occupational fatalities—1984.

Industry Group	Workers (x1000)	Deaths*	Deaths Per 100,000 Workers
All Industries	104,300	11,500	11
Trade	24,000	1,200	5
Manufacturing	19,000	1,100	6
Service	28,900	1,200	7
Government	15,900	1,400	9
Transportation & Public Utilities	5,500	1,500	27
Construction	5,700	2,200	39
Agriculture	3,400	1,600	46
TRUCK DRIVERS	1,876**	1,087***	58
Mining, Quarrying	1,000	600	60

SOURCES: *Accident Facts 1985, National Safety Council
**Employment and Earnings January 1985, U.S. Department of Labor
***FARS 1984

side fixed objects) than are passenger cars and light trucks/vans.

Medium and heavy trucks, like most other vehicles, experience most of their accidents on the roadway itself (79 percent)—as opposed to off-road, in daylight (76 percent), on straight (79 percent), dry (66 percent), and level (69 percent) roads. All vehicle types have similar patterns. Observed variations from state to state are more indicative of geographic or weather pattern differences than they are of differences in truck accident involvement propensity.

Combination-unit trucks experience a large proportion (59 percent, Washington 1981-83) of their accidents on roadway types likely to be used in over-the-road operations (i.e., Interstates, U.S. and State routes). This contrasts with the accident experiences of single-unit heavy trucks which reflect travel patterns in urban/suburban settings (68 percent of single-unit truck accidents occur on city streets or county roads). The speed involved with travel on the former types of roads, has a direct effect on accident severity outcomes.

A large portion of combination-unit truck accidents (55.8 percent in Washington) occur on undivided highways. Travel on undivided highways provides an increased opportunity for the truck to be in conflict with other vehicles. Also, the occurrence of an accident on this type of road increases the likelihood of it being serious, since head-on collisions are possible.

As previously discussed, there are typically numerous overlapping factors which combine to ultimately "cause" an accident to occur. Some of these are documented in accident data collection systems. Driver-related errors, infractions, or misjudgments are among the frequently cited factors contributing to the cause of truck, as well as other types of vehicles', accidents. In Washington, for example, in 54 percent of all accidents in which combination-unit trucks were involved, the truck driver was cited for some type of error or infraction.

While significant reductions have been made in recent years, alcohol is still involved in 43.3 percent of all fatal accidents. Alcohol is not involved proportionally in as many medium and heavy truck accidents, however, either in terms of the truck drivers or the other vehicle drivers involved. Based on a 15 state sample of fatal accidents where blood alcohol concentration (BAC) levels of fatal accident involved drivers are routinely gathered, it was found that 2.9 percent of all truck drivers, and 16.6 percent of the drivers of other vehicles involved in accidents with heavy trucks, had BAC's greater than 0.1.

Factors related to the mechanical condition of the truck are sometimes noted as having contributed to the cause of an accident. Problems of this type are typically coded in most accident reporting systems only when equipment is obviously broken or worn out, as determined by visual inspection. Equipment that is degraded, but still intact, such as brakes that are out of adjustment is usually not reported. For example, in Washington in 1981-1983, only 8.9 percent of all the combination-unit truck accidents were cited as being attributable to vehicle component part deficiencies. Brake system deficiencies are the most prevalent. This contrasts with roadside vehicle inspection findings[3] where, routinely, 20 percent or more of the vehicles inspected are placed out-of-service for vehicle component part deficiencies, most of these being related to brake system deficiencies.

In summary, medium and heavy truck accidents are not particularly numerous nor are they overrepresented among all motor vehicle accidents. They are, however, unusually lethal and more often than not, it is other highway users, with whom trucks share the highways, that are the victims in these accidents. Among the most significant reasons why this pattern of fatal accidents occurs are: the large disparity in size and weight between trucks and other vehicles, the typically high travel speeds at which trucks are operated and, travel patterns that, in many cases, place them on undivided highways where the likelihood of collisions with other vehicles increases.

Medium and Heavy Truck Dynamic Performance

As with all motor vehicles, driver control of medium/heavy trucks is limited to braking, acceleration and steering inputs. Any or all of these control applications are utilized to operate the vehicle under routine conditions or in the attempt of non-routine, often severe, avoidance maneuvers when the driver is confronted with a potential crash threat. In the case of most four-wheel vehicles, comparatively severe levels of either steering or braking must be made to induce dynamic instabilities in the vehicle. This is not the case with medium/heavy trucks. These vehicles are susceptible to rollover, spin-out and jackknife in much less severe steering maneuvers.

The ultimate criteria for judging the stability and control performance of a motor vehicle is whether or not the vehicle's driver can maintain stable control under all intended and foreseeable conditions of operation. In this regard, one can consider that the expectation of good dynamic behavior is fulfilled when the vehicle:

- Attains a desired deceleration level during braking,
- Follows a desired path in response to steering,
- Remains upright (i.e., does not roll over),
- Maintains a limited swept path, and
- Does not oscillate from side to side in an uncontrollable manner.

In practice, medium/heavy vehicles often fail to meet these desired criteria for a variety of reasons. For example, they have the following performance limitations:

- *Poor wheels-unlocked stopping performance.* This results primarily from the general mismatch between the brake torques developed at each wheel and the prevailing wheel loads. This mismatch occurs due to the tremendous changes in wheel loading (both static and dynamic) that take place as a result of payload weight and placement. In addition, truck brakes often fail to deliver their designed torque output because they are not properly adjusted.
- *Poor retention of braking capacity during descent of long and/or steep grades.* The braking horsepower necessary for a fully-loaded vehicle to safely descend a substantial grade at highway speed places a large demand on the capacity of most truck brake systems. Parasitic losses which would normally aid in slowing the vehicle are low relative to the total vehicle weight. The search for improved fuel economy continues to reduce these parasitic losses even further.
- *Loss of directional control.* Exceeding the vehicle's yaw stability limit results in vehicle spin-out (single-unit trucks), and jackknifing or trailer swing (combination-unit trucks) conditions. The primary cause of these phenomena is the rearward bias of braking forces typical in the brake system designs of U.S. medium/heavy trucks. This increases the probability of rear wheel lockup. When lockup occurs, tires lose their ability to generate side force and the vehicle becomes unstable in yaw. Unstable yaw response in a

medium/heavy truck is likely to generate turning responses which exceed the vehicle's roll stability limit, thus precipitating a rollover.

- *"Crack-the whip" response of multiply-articulated vehicles (doubles, triples and certain truck-full-trailer trailer combinations).* Multiple-articulated vehicles, have a tendency for the rearmost unit of the vehicle to show exaggerated or amplified response relative to the towing unit in certain types of severe obstacle avoidance maneuvers. "Rearward amplification" has important safety consequences when, during such maneuvers, the rearmost trailing unit exceeds its own roll stability threshold and rolls over.
- *Straightforward vehicle rollover.* Attempting turning maneuvers at too high a speed results in the vehicle's roll stability limits being exceeded.

The stability and control characteristics of medium/heavy trucks are direct indicators of their safety performance. A driver's ability to control his vehicle is ultimately limited by the response of the vehicle to steering and braking inputs. Limitations on the dynamic control capabilities of the vehicle reduce the viable options which are open to a driver in maneuvering to avoid traffic conflicts produced by other vehicles and also reduce the tolerance which is available to compensate for any inappropriate control inputs made by the driver. In effect, the vehicle becomes less forgiving of control errors.

The Performance Characteristics of Medium and Heavy Trucks in Maneuvers Involving Braking

The Size Of The Brake System Related Safety Problem

There are basically four different types of truck accidents that could be related to braking system performance: accidents due to failed or inoperative brakes; runaways on down grades; accidents where the vehicle was unable to stop in time (brakes did not fail nor were they ineffective due to heat but they simply did not provide the stopping force necessary to avoid the accident), and; skidding or loss-of-control accidents where wheels locked during braking.

Collectively, the performance of truck brake systems could be a contributing factor in as many as one-third of all truck accidents[1].

Truck Brake System Limitations

Truck brake systems have a number of critical limitations, namely:

- Inadequate Capacity in Continuous or Repeated Braking Situations—The adequate sizing of truck brake systems in terms of braking torque and thermal capacity is dictated by more than just the mass of the vehicle. For example, a tractor-trailer typically weighs approximately 30 times as much as a passenger car but needs 167 times as much braking power to maintain a steady speed on a 6 percent grade[4]. In addition, the capacity of truck brake systems has not increased to match the increasing demand placed on them as a result of fuel economy enhancement efforts to decrease parasitic drag. As a result truck drivers must compensate even more than in the past, especially when descending grades. Lower descent speeds (and lower transmission gear ranges) must be used to prevent runaways.
- Poor Brake Distribution—U.S. trucks and combination-units typically have a strong rearward bias in the application of braking force. Front wheel/steering axle braking is usually low. This results in stopping distances which are longer than those of other vehicles, especially under emergency conditions. Additionally, combination-unit trucks can easily become unstable due to locked wheels under many brake application conditions.
- Incompatibility of Tractor and Trailer Brake Systems—Many tractor and trailer brake systems are not compatible—i.e., they do not function well together to provide desirable overall combination-unit vehicle braking performance. Often, the amount of braking force being applied by the tractor's axles greatly exceeds that of the trailer's, or vice versa. Similarly, the brakes may apply or "come on" quicker on the tractor than on the trailer. Incompatibility compromises both vehicle stability and brake effectiveness which can result in uneven brake wear problems and brake fade on downhill descents. In addition, brakes on trailers often apply and release slowly compared to those on the tractor. This is due to the distance between the brake control valve (treadle) and the trailer brake valve(s). Slow brake application times increase stopping distance and slow release times make it difficult to recover quickly from trailer wheel lock-up should this occur. This problem is more pronounced with longer combinations.
- Sensitivity to Brake Maintenance—Because truck brake systems are more complex and

experience comparatively more severe service conditions than passenger car brake systems, they require a great deal more maintenance. Frequent inspections and repairs must be made to assure that systems are operating and are properly adjusted (since, unlike passenger car brake systems, most truck systems do not self-adjust with wear). Roadside inspections have, for many years, indicated that many operators do not adequately maintain their vehicles. This is compounded by the absence of consensus measurement standards, performance criteria and marking/labeling schemes for components within the brake system. This makes it difficult, if not impossible, for truck operators to obtain replacement parts such as valves and linings which exhibit comparable performance to the parts that were originally installed on the vehicle. Because of this, compatibility problems are often created or worsened when repairs are made.

In order to address these problems, a three phase program of research is suggested. It would deal: first, with compatibility and brake maintenance problems; secondly, with controllability problems associated with braking maneuvers, especially while operating lightly loaded or empty on slippery road surfaces, and; thirdly, with efforts to optimize the brake system to improve stopping performance.

Research Program To Address Brake Compatibility, Brake Adjustment, And Component Performance Problems

There exists a need to ensure that today's brake systems function as well as possible. In any discussion of heavy truck braking systems—especially among truck users—the subjects of compatibility, component level performance, and brake adjustment always surface as priority concerns. The reason for this is that poor compatibility becomes obvious very quickly in terms of excessive brake lining and drum wear, brake drum cracking, the need to adjust brakes constantly, etc., on the "over-braked" unit of the combination.

The safety implications of operating a truck with incompatible brakes are subtle and difficult to isolate in accident statistics. Marginal stopping performance would not, of itself, precipitate an accident. It only becomes a problem if a crash avoidance braking maneuver is attempted, and these are not everyday occurrences. In addition, if a crash does occur under these circumstances, other more apparent factors are likely to draw attention away from the fact that poor braking performance was a contributing factor.

The overall research and implementation program needed to address brake system incompatibility and brake maintenance issues is shown in Figure 1. Industry, government and professional standards setting organizations are currently actively involved in programs to address the issues of pneumatic timing, brake force balance, performance and labeling of valves and linings and improved means for ensuring proper brake adjustment.

These activities need to be actively supported and continued, and, where possible, accelerated. Compatibility and maintenance issues are the everyday concerns of conscientious motor carriers. They must be satisfactorily addressed before motor carriers will become more receptive to the new technology that must be incorporated in trucks in order to make significant improvements in truck controllability and stopping capability when braking.

Research Program To Address Braking-Induced Instability Problems

Even if substantial improvement can be achieved with regard to tractor-trailer compatibility and maintenance issues, truck brake system performance would still be deficient at limit conditions, i.e., when making an emergency stop or when making a brake application that is too "hard" for conditions. An example of the latter case would be when the driver has misjudged the amount of brake pressure he can safely apply when operating an empty or lightly loaded vehicle on a slippery roadway. Without load proportioning systems and/or antilock braking systems, compatibility can only be achieved for a single design loading condition, typically the fully loaded condition. Many trucks operate lightly loaded or empty a significant portion of the time.

The most promising technology that is currently available for significantly improving braking performance at these limit conditions is the antilock brake system (ABS). These systems are the only solution to the wheel lock and resultant loss-of-control tendency typical with currently designed U.S. vehicles. Almost everyone in the trucking industry agrees that antilock has the potential to significantly improve the braking performance of heavy trucks by eliminating the directional instabilities which occur when wheels lock. Many, however, question the reliability and maintainability of the systems in actual use as well as the ability of the systems to fail safe (i.e., in the event of a malfunction the system reverts back to a normal brake system without antilock). Lack of reliability was the major reason for the Court's decision in 1978 to set aside the wheels unlocked stopping distances in FMVSS 121.

The question to be addressed is, can the current generation systems function reliably on trucks operating in fleet service in the U.S.? To answer this question a government-sponsored, comprehensive,

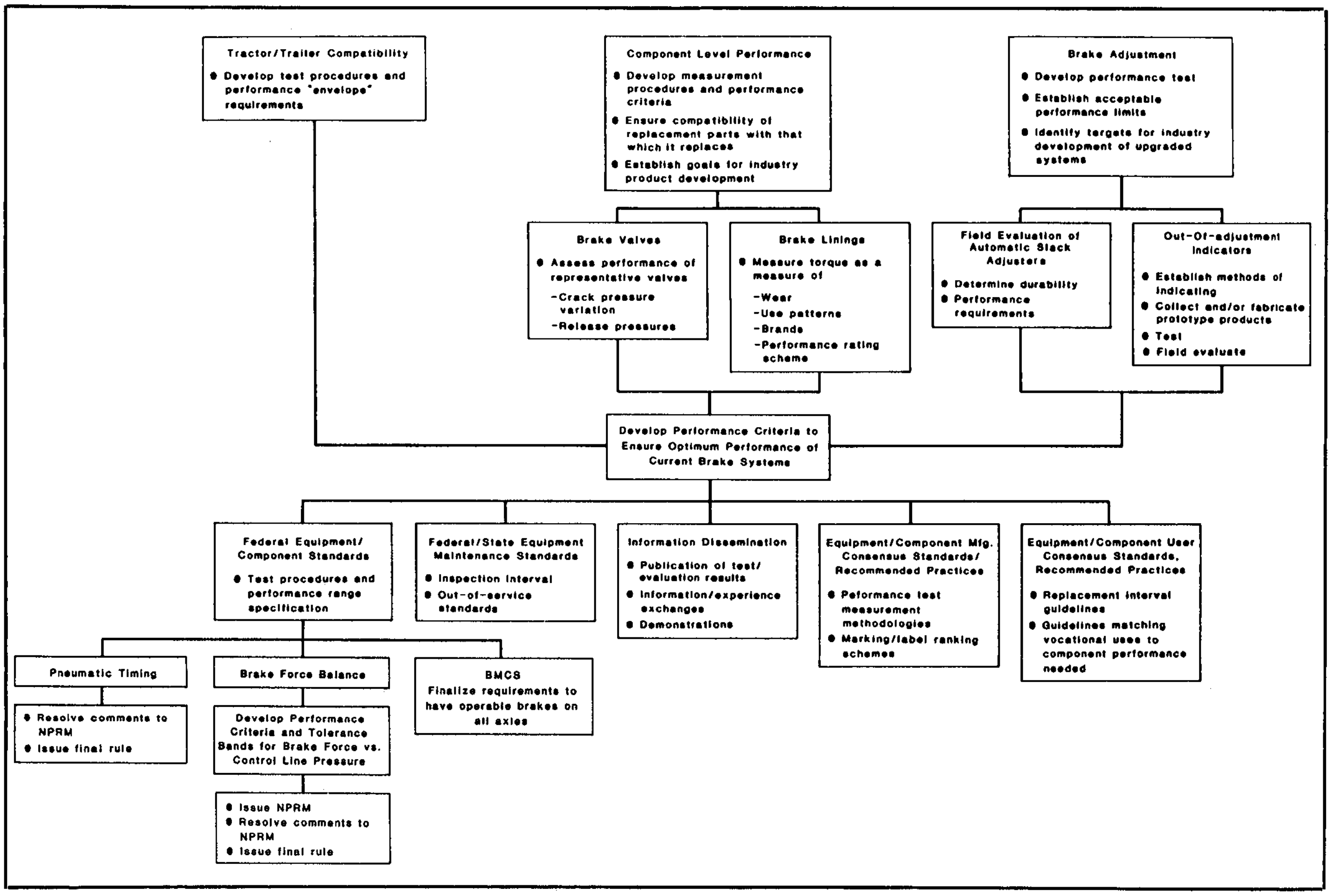

Figure 1. Truck brake performance improvement program—brake compatibility and maintenance

closely monitored fleet study, in cooperation with antilock suppliers, truck manufacturers and motor carriers is suggested. This study would yield sufficient performance, reliability, maintainability and cost data to support intelligent decision-making on the part of motor carriers and government relative to the suitability of this technology for widespread application in trucking. Initiating such a study now would result in the data being available in the early 1990's. Thus, it is imperative that this portion of the brake research program be conducted in parallel with the compatibility research project discussed previously. The project is outlined in Figure 2.

Research Program To Improve Truck Stopping Performance

The objective of this part of the program would be to achieve the maximum practical limit braking performance possible. It would build on the improvements expected to result from the first two portions of the program.

Ultimately, a vehicle's stopping performance is limited by the overall amount of brake force capacity that foundation brakes can generate and by the traction properties of the vehicle's tires. Truck brake force capacity on domestic vehicles is already at its limit with the exception of front wheel brakes. The power of this part of the system could be increased as could tire longitudinal traction. Research is necessary to understand the trade-offs involved in achieving these objectives and to establish reasonable goals for product development by the industry.

This part of the overall program would, in general, follow the previously discussed research. It is shown diagrammatically in Figure 3.

The Performance Characteristics of Medium and Heavy Trucks in Maneuvers Involving Steering

The steering response characteristics of a vehicle are one of the principal descriptors of its safety performance capabilities. In addition to braking capabilities, these properties define the inherent limits of safe vehicle operation.

As a result of extensive research conducted since the early 1970's, considerable progress has been made in identifying the factors which affect the directional control and stability of medium/heavy trucks. The current state-of-knowledge indicates that some trucks

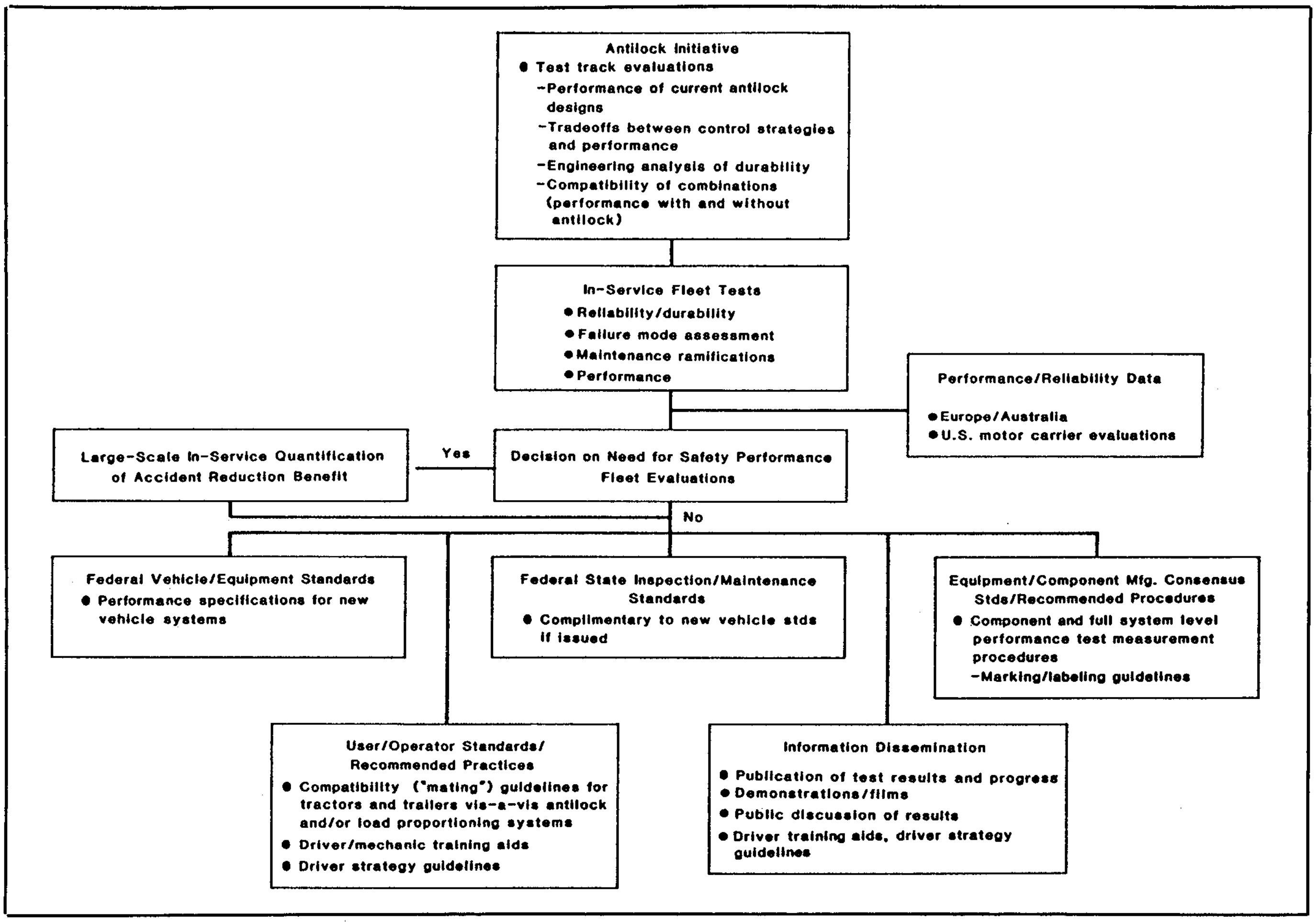

Figure 2. Truck brake performance improvement program—braking-induced instability

have safety-related response tendencies that limit the range of performance over which they can be operated.

Heavy trucks cannot be steered around corners, change lanes, or avoid unexpected obstacles as quickly as a car can, nor are they able to make right-angle turns the same way cars can without experiencing difficulties. They are more prone than cars to rolling over in turns or, in the case of some multiply-articulated combination-unit trucks, when attempting quick lane change accident avoidance type maneuvers. Multiply-articulated vehicles also may exhibit oscillatory behavior when simply travelling in a straight line.

Of the topics previously discussed, rollover has direct and significant safety consequences and is in need of additional work to translate previous research findings into implementable solutions. Accordingly, a program for further research in this area is proposed.

Rearward amplification is a problem unique to a special class of vehicles (multiply-articulated, larger combination unit vehicles (LCV's)). Some vehicle design-related changes could be made that would help reduce the likelihood of this occurring. These are close to being implementable. Other factors also affect this tendency, and, in many cases, to a greater degree than do vehicle-related factors. These include operational use practices and legislative choices relating to vehicle size, weight, and configuration allowances.

Problems associated with low speed off-tracking are certainly a concern from a traffic engineering and operations viewpoint, but are not significant from an highway safety viewpoint. Few traffic accidents are likely to be associated with this characteristic of trucks. Those that do occur are likely to be low severity, property-damage-only events. Accordingly, no additional work on this subject is proposed.

High-speed yaw instability, while demonstrable from an engineering viewpoint, is not evident as an accident causal factor. It is not likely to ever be evident in mass accident data files, since when it does occur, it is likely to result in the vehicle rolling over. This topic is best addressed in conjunction with efforts to improve truck roll stability.

The oscillatory behavior of multiply-articulated vehicles is also not likely to be a significant highway safety problem. It is of concern, however, in that trailing units may encroach on other travel lanes or

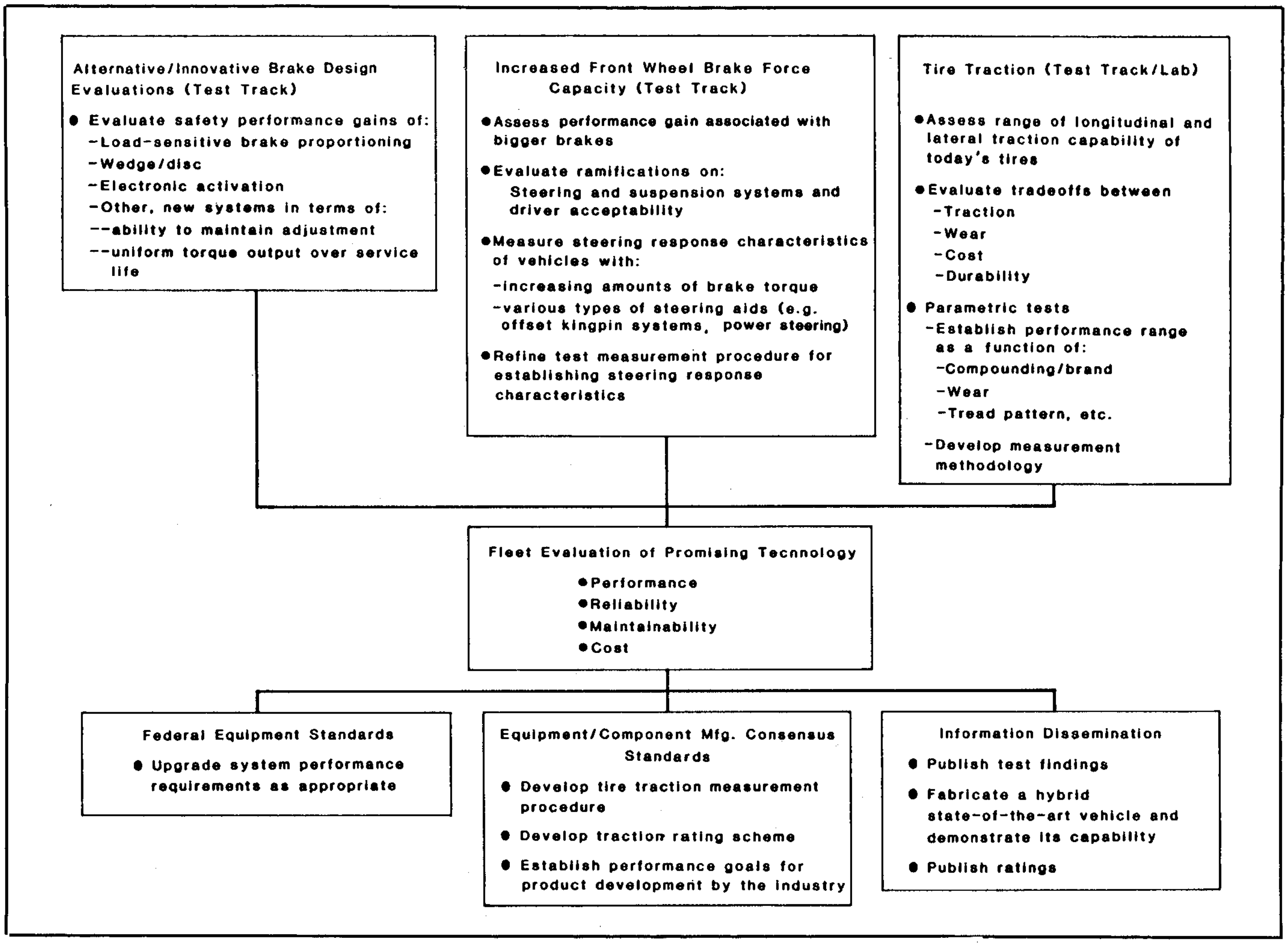

Figure 3. Truck brake performance improvement program—stopping performance

intimidate other motorists and thereby cause erratic maneuvering actions.

The Prevalence And Characteristics Of Rollover Accidents

Rollovers constitute a very visible and serious type of commercial vehicle crash. Although vehicle rollover is involved in from 4 to 9 percent of all medium/heavy truck crashes, it accounts for approximately one third of the single-vehicle accidents. Rollover occurs in approximately 15 percent of the fatal crashes and is a contributory factor in nearly 60 percent of the medium/heavy truck occupant fatalities.

Research Plan For Improving Truck Handling And Stability Performance

Rollover is given the highest priority among handling and stability related issues because it is well understood and has an obvious direct link to safety. The program to improve the roll stability properties of trucks follows three parallel paths.

One of the paths would be directed towards developing the best methods of gauging the relative roll stability performance of trucks. It would take into account static and dynamic considerations.

A second path would attempt to establish what motion and visual cues drivers sense (or possibly fail to sense) prior to a rollover. This information could help driver training efforts and could possibly result in more of the "good" cues being built into future trucks.

Another path would study in-service trucks to assess how many of them are typically being operated close to their stability limits. This would include studies of truck tires to determine the degree to which their performance properties affect vehicle stability and control. A determination would also be made as to which properties of in-service trucks are most responsible for stability limits being approached. Finally, an assessment would be made of the impacts that would result from design-related changes that might be contemplated to enhance roll stability of future trucks.

The overall roll stability enhancement research program is shown in Figure 4.

Medium and Heavy Truck Crash Performance—Truck Aggressivity

When any two vehicles of dissimilar size collide with each other, the larger of the two vehicles typically inflicts much more damage and injury trauma than it sustains. In this case, the larger vehicle is said to be more "aggressive" relative to the smaller vehicle.

In medium and heavy truck collisions with other vehicle types, the truck's "aggressivity" occurs for two principal reasons: geometric mismatch (the fact that the physical shapes of the two vehicles, particularly the front end of trucks, do not match each other), and mass mismatch (the difference in weight between the two vehicles).

Many believe that little can be done about truck aggressivity short of segregating trucks from other vehicles. This may not be totally true since a significant portion of the problem arises from geometric rather than mass differentials. Practical improvements may be possible for at least this part of the problem.

Extent Of The Aggressivity Issue

Two-vehicle collisions are the largest single category of fatality producing motor vehicle/highway related accidents. In 1984, two-vehicle collisions accounted for 37.7 percent (16,668) of all highway related fatalities. Collisions between medium/heavy trucks and other vehicles resulted in 21 percent (3,423) of all the fatalities sustained by occupants of other smaller vehicles involved in two-vehicle collisions. The majority of these victims (71.9 percent, 2,461) were passenger car occupants. In all, these 3,423 fatalities represented 7.7 percent of all the highway related fatalities occurring in 1984.

Research Plan For Reducing Heavy Truck Aggressivity—Frontal Impact Attenuation/Override Prevention

The extent to which the effects of collisions between medium/heavy trucks and other smaller vehicles can be ameliorated is not clear. Readily achievable solutions are not apparent, however, given the appreciable number of occupants of other smaller vehicles (3,423) who are killed in collisions of this type, it appears worthwhile to study the possibility that even small incremental improvements can be achieved.

Sixty-eight percent of the fatal car/combination-unit truck collisions involve the fronts of trucks. Logically, then, work would begin on this portion of the vehicle. The program to explore the feasibility of practically modifying truck front end designs is shown in Figure 5.

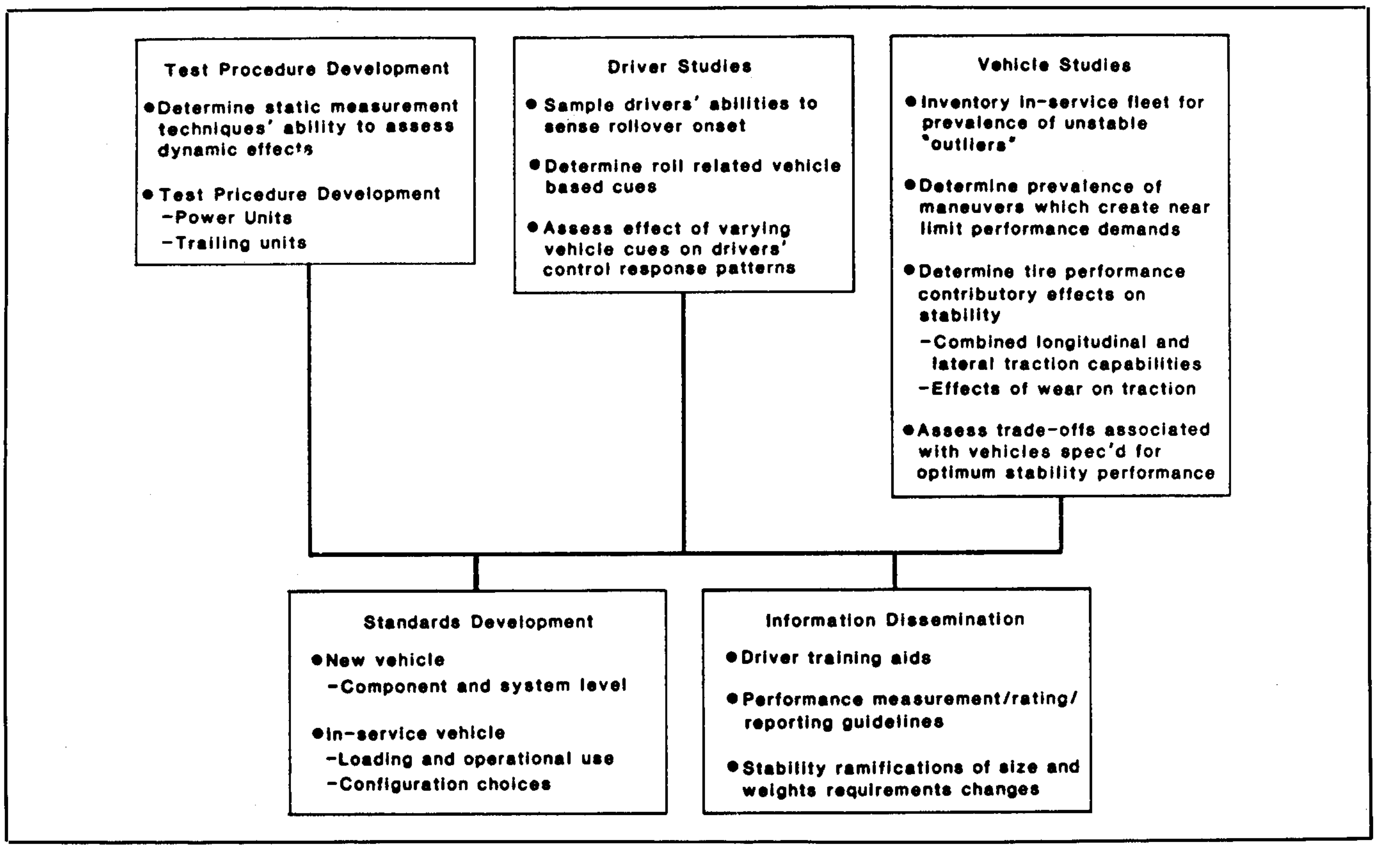

Figure 4. Truck roll stability enhancement research program

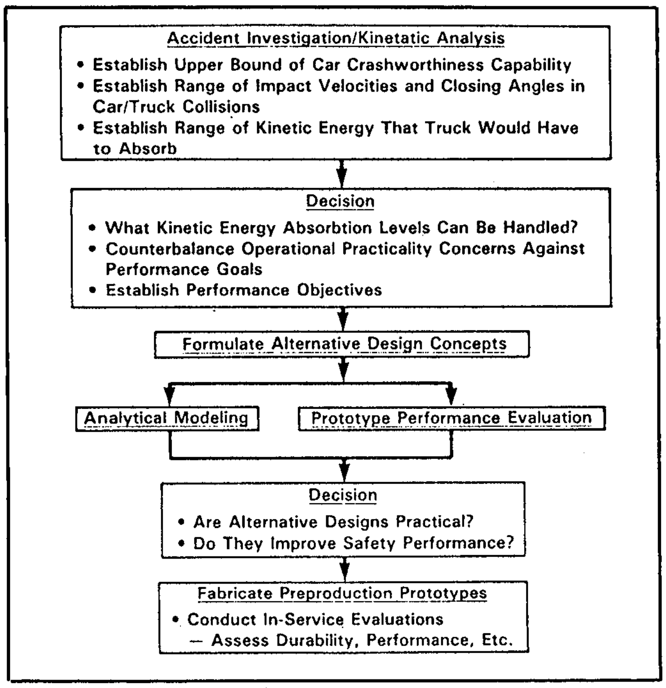

Figure 5. Truck frontal attenuation/aggressivity reduction research program

Truck Occupant Crash Protection

Each year about 1,000 occupants of medium and heavy trucks are killed and 400,000 injured in crashes. Most are drivers of combination-unit trucks. Highway crashes are the principal occupational hazard faced by truck drivers.

Accident data analyses have indicated that not all heavy truck occupant fatalities occur in catastrophic accidents. Rollovers, ejections, entrapment in crushed cabs, contact with interior surfaces, and fires are the primary mechanisms responsible for the majority of truck occupant fatalities. Most of these fatalities occur in single-vehicle accidents which involve running off the road and hitting fixed objects, simple on-road overturning, jackknifing (with or without overturning), and collisions with low roadside structures such as guard-rails, sign posts, and embankments.

In many crashes which are now fatal to truck occupants, use of a safety belt more than likely would have been all that was necessary to avoid the fatal outcome of the accident. No other countermeasure is likely to reduce the number of truck occupant deaths and serious injuries as greatly as would increased restraint system use. Beyond this, further improvements could be realized through truck designs that provided:

- Seats and restraint systems that keep the driver firmly in place and prevent him from being thrown around inside the cab or being ejected,
- A reasonable amount of protection from post-crash fire,
- Cab interiors free of sharp, hard objects that can cause injury during impact—especially the steering wheel rim and hub,
- Strong, rigid cab designs that provide occupant survival space in a crash (within reasonable expectations), and after a crash, means of escape.

Before practical solutions can be sought in any of these areas, both the types of crash environments for which protection would be sought, and an upper range of severity for which practical solutions are deemed feasible, would have to be established. Using this approach, multiple-event collisions/impacts represent the most reasonable group of accidents to address, while those involving high-speed impacts into rigid non-yielding objects (such as bridge piers) clearly are not. Initially, 9 g crashes would be assumed to be the maximum level for which improvements could practically be sought[5]. Further attempts to refine this estimate of a reasonable upper bound of deceleration would have to await the completion of detailed reconstruction and analysis of accidents that have been investigated in-depth.

Research would proceed along both analytical and experimental lines. The analytical work would consist of in-depth accident investigations and mathematical modeling.

The accident investigations would be targeted to better define typical crash-induced forces, direction of force application, fuel and ignition sources in accidents involving fires, occupant trajectories, and causes and amount of occupant injury trauma sustained.

The modeling activities would focus on defining vehicle dynamics both before and during the crash event, describing the response of the vehicle structure to crash-induced loads, and predicting occupant trauma outcomes (given assumed crash forces).

Existing approaches and/or standards would be used wherever possible. Component-level testing (as opposed to full systems-level tests) would be the preferred method for experimental work. Existing standards (for example, those now used for rollover protection systems (ROPS) for construction equipment and the Swedish and ECE cab structural integrity tests) would serve as initial reference points for experimental testing.

The emphasis on all this work would be to achieve practical incremental improvements rather than strive for total, yet unattainable, solutions.

Research Program To Improve Heavy Truck Occupant Restraint Systems

Observational surveys of seat belt use among combination-unit truck drivers indicate that few (6.25

percent) use them[6]. Surveys of truck drivers' opinions relative to safety belts indicate that many erroneously believe that they are not effective in reducing injuries in crashes. In addition, many drivers feel safety belt designs are deficient. Drivers said safety belts were often dirty (because of lack of retractors) and uncomfortable because they did not "give" in the truck's relatively rough riding environment.

Efforts to improve this situation have taken several tacks. First, an industry/government ad hoc group developed a packaged safety belt use promotional and information kit aimed exclusively at motor carriers and truck drivers. The package has been promoted through the American Trucking Associations' 50 state trucking organizations.

Secondly, NHTSA issued an Notice of Proposed Rulemaking (NPRM) to amend FMVSS 208 to require emergency locking retractors (ELR'S) and push-button releases on truck and bus safety belt assemblies. The purpose of the proposed amendment was to promote the use of safety belts by ensuring that they are comfortable and easy to use and that they remain clean.

Truck safety belts without any retractors often become entangled in the suspension mechanism of truck seats where they become dirty and difficult to extract and use. It was reasoned that by requiring ELR's, safety belts would be kept clean while ensuring that they not cinch up around drivers' waists in the comparatively rough riding environment of a truck.

Finally, efforts have proceeded to further upgrade the performance and comfort of truck safety belt designs. There is growing interest among some truck operators in having workable, comfortable 3-point restraint systems in heavy trucks. Research is needed, however, to determine how best to modify and adapt 3-point systems to the ride environment typical in heavy trucks since passenger car systems cannot be used in unmodified form. Manufacturers also report that their efforts to design 3-point systems for heavy trucks are often constrained by restraint system strength requirements they feel are unnecessarily high.

Accordingly, the primary thrust of engineering research to improve restraint systems should be directed towards ascertaining strength requirements for truck safety belts that are appropriate for truck occupant protection goals. These levels should be based on the objectives of preventing occupants from being thrown about inside the cab while at the same time preventing them from being ejected. The objective would be to achieve a balance which optimizes occupant protection performance while also enabling maximum design flexibility.

The program is shown diagrammatically in Figure 6.

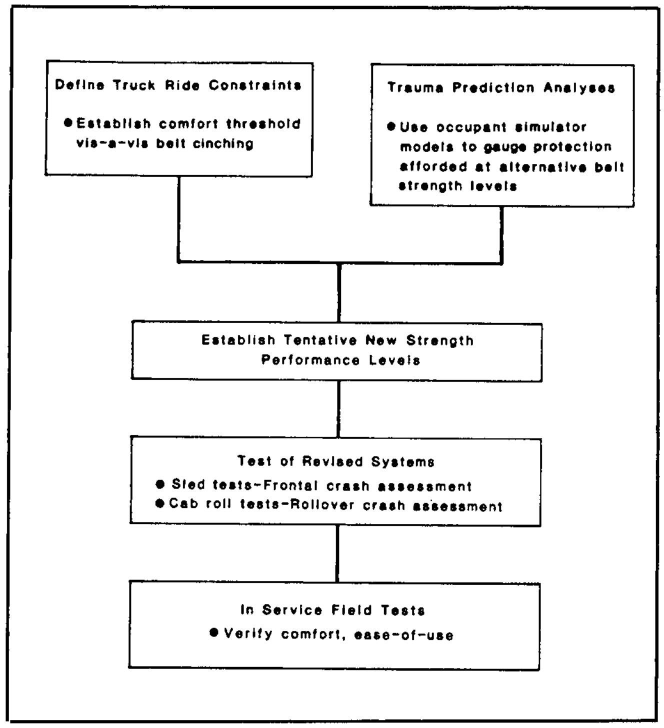

Figure 6. Truck occupant restraint systems research program

Research Program To Prevent Post-Crash Fires

Post-crash fires are involved in nearly 16 percent of all fatalities to the occupants of medium and heavy trucks. The comparable figure for passenger cars is 4 percent. Studies that have been conducted on this issue indicate that between 10 and 50 percent of these fires result from fuel loss from the truck's fuel containment and delivery system [7].

Fire related accidents are obviously very severe. In many cases, it is impossible to determine whether the crash events precipitating the fuel release were so severe—in and of themselves—as to preclude vehicle-based upgrades. Nevertheless, reasonable incremental improvements may be possible.

Accordingly, a research program seeking ways of preventing post-crash truck fires would be directed towards enhancing the ability of truck fuel systems to withstand crashes and remain intact. It would include investigations of both the feasibility and desirability of using: "kill switches" in the truck's electrical system to eliminate this a source of ignition; bladders in fuel tanks to contain fuel in damaged tanks; non-toxic, flame-retardant materials in truck cabs; on-board fire suppression systems, and; fuel delivery and return lines rerouted away from engine and/or exhaust heat sources. The program is shown diagrammatically in Figure 7.

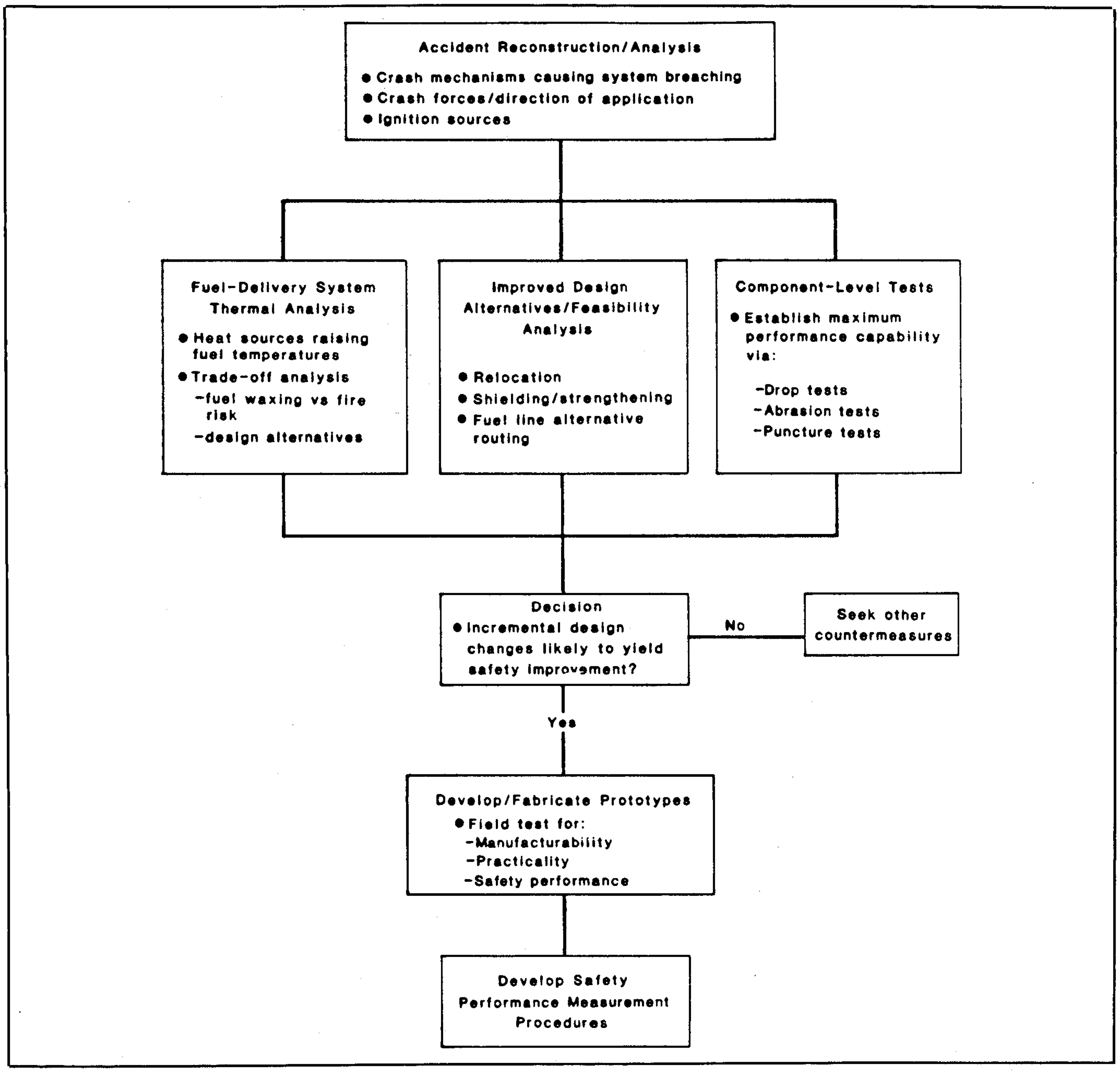

Figure 7. Truck fire prevention research program

Research Program To Improve Steering Assemblies And Other Cab Interior Components

Serious injuries sustained by truck occupants who remain inside the vehicle in accidents (where intrusion of the occupant compartment is not sufficient to cause entrapment) result from contact with interior components. Where specific injury information is available, the steering wheel has consistently been identified as the primary source of serious injury to heavy truck occupants, followed by sun visors/roof top moldings, roof and side rails, and instrument panels.

Analyses of occupant trajectories in accidents reveal that the majority of crash-involved drivers move in more than one direction during the crash sequence. This is possible because many truck crash events occur over long time periods (seconds) compared to those of cars (milliseconds). Given this much movement by occupants, it is not suprising that besides impacts into steering wheels, contacts with the windshield, instrument panel, and surfaces of doors and door headers have also been identified as common sources of injury.

The goal of this research program, therefore, would be to develop means of reducing occupant injuries that result when truck drivers, both restrained and unrestrained, impact cab interior surfaces and components, especially truck steering wheel/column assemblies.

The program is shown in Figure 8.

Research Program To Enhance The Structural Integrity Of Truck Cabs

A crash performance objective that heavy trucks share with passenger cars is maintenance of sufficient space within the occupant compartment to "ride-out" the crash event. This capability is of equal importance with that of restraining/containing occupants within the vehicle since it logically follows that if ejection is prevented it is of no avail if the occupant is subsequently killed or injured due to the cab crushing.

Therefore, the objective of research on heavy truck cab structures would be to determine if presently available cabs could be strengthened practically to enhance their ability to protect occupants in non-catastrophic, yet potentially lethal crash environments. Excluding those ejected, 20 percent of all fatally injured combination-unit truck occupants are extricated from their cabs—a surrogate indication that cab structural integrity is, at least partially, involved.

European (Swedish and ECE) standards exist which, if applied to U.S. trucks, would result in incremental strengthening of most U.S. designed truck cabs. Research is needed, however, to determine if practical modifications can be made to U.S. cabs that

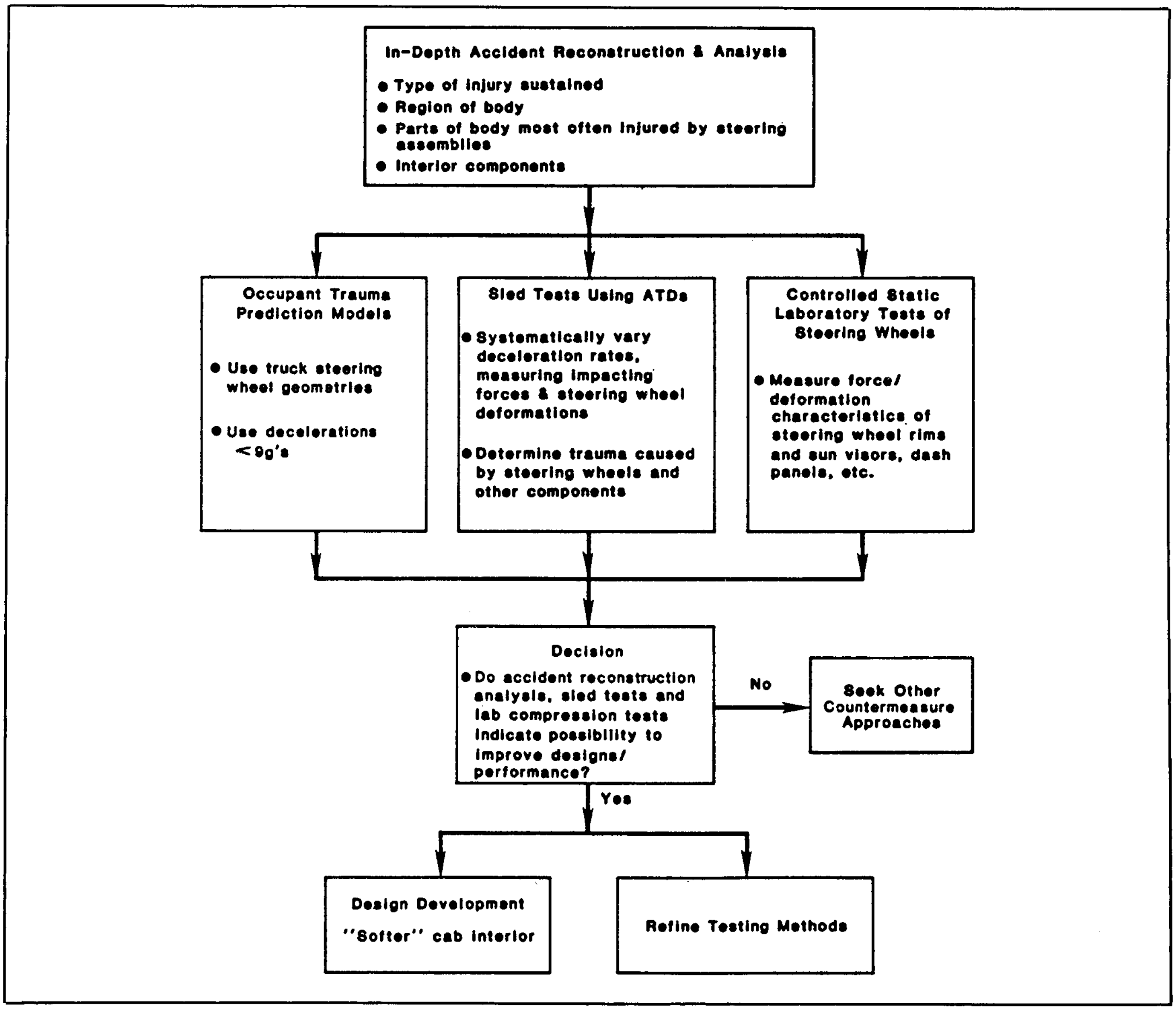

Figure 8. Truck cab interior components research program

would materially improve the likelihood of saving an appreciable number of victims in typical U.S. truck crashes, especially rollovers. Limitations and trade-offs are inherent in such an endeavor and must be recognized and acknowledged at the outset. The program is shown in Figure 9.

Summary

In the U.S., medium and heavy trucks are annually involved in crashes which result in over 5500 people being killed and 170,000 injured. There are many complex and interrelated reasons why these crashes occur, among them being factors which relate to the way the vehicles are designed, maintained, and perform. This paper identifies the key vehicle-related medium and heavy truck safety issues and briefly summarizes programs of research to achieve improvements. It is hoped that these research agendas will serve as blueprints for both industry and government efforts to achieve those improvements.

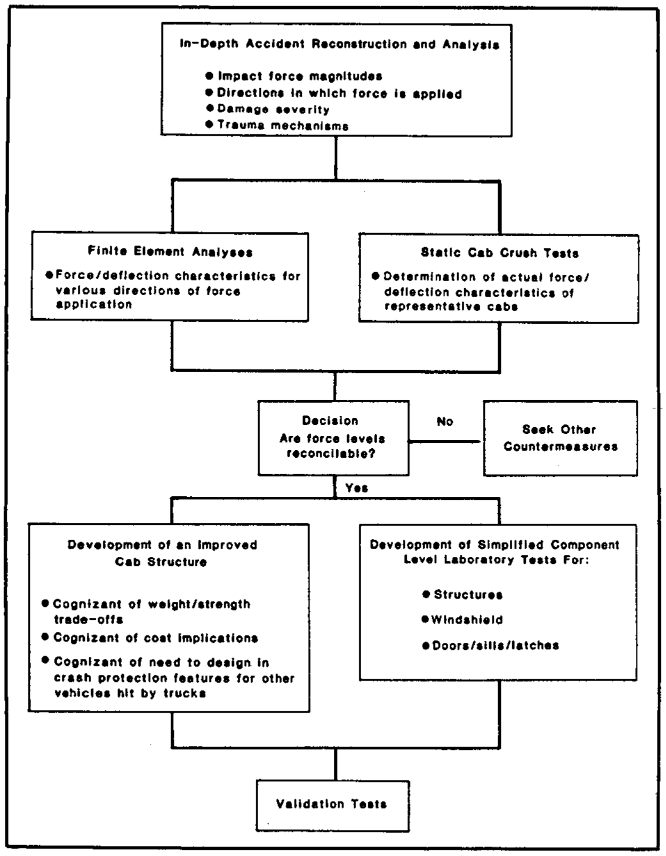

Figure 9. Truck cab structural integrity research program

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European Review of Heavy Goods Vehicle Safety

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Preface

The Committee of EEVC (European Experimental Vehicles Committee) decided at their 1984 policy meeting that an informal Working Group on Heavy Goods Vehicles be set up to consider the road accident situation in Europe for these vehicles. It should also report upon the progress being made to improve the accident avoidance capability of such vehicles, the protection that would be provided for the crew and the protection that might be possible for other road users involved in accidents with Heavy Goods Vehicles. This informal Working Group did not start work until 1986. It was requested by the main EEVC committee in 1986 that it report in time to present papers to the OECD Symposium on this subject in April 1987 in Montreal, Canada and to the 11th ESV Conference in May 1987 in Washington, D.C., U.S.A. The former objective has not been met, but the present report in preliminary form and without the final approval of the main EEVC committee is now presented.

The informal group consisted of:

Mr Pullwitt,	BASt, FR Germany
Mr Cesari,	INRETS, France
Mr Fline,	INRETS, France
Prof. Strandberg,	VTI, Sweden
Mr Tromp,	SWOV, Netherlands
Mr Riley,	TRRL, UK
Mr Neilson,	TRRL, UK (Chairman)

Introduction

In several parts of the world there has been increasing concern about the contribution that Heavy Goods Vehicles, and similar vehicles designed for special purposes, are making to the overall road accident situation. The present study shows that generally speaking the situation in Europe is somewhat improving for a complicated set of interrelated reasons. This report is intended to review the situation and to discuss engineering means by which the design of these vehicles can be further improved. This is a continuing and important objective because in some countries there has been a widespread fear of these large road vehicles and the accidents and injuries that they can cause. This has been hindering the development of road transport and has been leading to restrictions on the operation of such vehicles in some places.

Any discussion of heavy commercial vehicles should start with some definitions about which vehicles are included and how the sizes are divided up. The present study is concerned with Heavy Goods Vehicles and this excludes light goods vehicles, bus and coaches and miscellaneous large vehicles which do not carry goods. The point of division between light and heavy is not the same in different countries in Europe and varies between about 3.5 and 7.5 tonnes Gross Vehicle Weight. Taking the lower limit, HGVs are about 3% of the vehicle population, but they may cover 8% of the distance. Other large and specialist vehicles which do not carry goods are an addition to the number of vehicles, but they appear to have a much lower involvement rate in accidents presumably because these vehicles cover much shorter distances each year. The following section gives an indication of the part that Heavy Goods Vehicles play in the overall road accident pattern in one country. It is mostly based on the accident statistics for Great Britain.

The Accident Situation in General Terms

The Heavy Goods Vehicle fleet in Europe may be made up of about 60% two axle rigid trucks, 10% rigid trucks with more than two axles and 30% articulated or full trailer vehicles. The traffic and distance travelled by these vehicles has been increasing slowly during the past ten years with the rather variable economic situation and the increase may be only about 15% over that period. However in some countries their involvement rate in accidents per distance covered has dropped by almost a third and the fatal injury rate for their drivers by a larger amount (possibly this has almost halved). These reductions are additional to the rather similar large reductions which occurred in the previous ten year period up to about 1975.

Almost all casualties in accidents involving HGVs are to other road users rather than to the drivers and passengers in them. These casualties are mostly divided between car occupants, riders of two wheelers and pedestrians with the HGV occupants accounting for only about a tenth of the total. The implications of this situation are discussed later on in some detail. For example, it has been noted that a proportionally large number of fatal pedestrian casualties occur in accidents involving articulated vehicles. Presumably it is the sides of these vehicles which cause additional fatal injuries.

Most HGV accidents leading to fatal injuries to their occupants occur away from built-up areas but for those injuring other road users perhaps a third occur in built-up areas. It is noteworthy that most fatal accidents outside built-up areas occur on the main roads, whereas in built-up areas there are a surprisingly large number on minor roads. It is the larger vehicles within the HGV category which have many of their accidents outside built-up areas and the two axled vehicles which have relatively more accidents in towns. Another way of considering risks to HGVs on the different roads is to compare their accident rates per distance travelled. By this measure, if the rate for roads outside towns is taken as standard, then the rate on Motorways is about a half of that, but the rate in towns is only slightly higher than outside them.

About half of the accidents to HGVs occur at junctions, but with rather fewer at junctions for the largest vehicles. It is these vehicles which have rather more accidents away from junctions and this largely accounts for the difference. Skidding of HGVs tends to be reported when control is lost of vehicles and they depart from their intended path. It is also reported when tyre marks are deposited on the road surface. A half of reported skidding accidents occur on wet roads, but there are almost as many in dry conditions. In most countries the totals in snowy and icy conditions are relatively small.

Road accidents occur in many different circumstances. A comparison with the manoeuvres of cars before accidents shows that HGVs have relatively few accidents, presumably because their drivers are professional. They have particularly few when waiting but held up, when turning to the offside and when just going ahead. However for accidents when going ahead at bends when overtaking, when changing lanes, when reversing and when parked, their accident rates approach those for cars. These findings rather suggest that HGV drivers have real problems in seeing obstacles and other vehicles but not when turning across traffic at junctions because they have a good view to the offside.

The main sections of this report now follow. There is a detailed comparison between the accident statistics for several European countries. Then there is consideration of the remedial measures for HGVs which are becoming available or which appear to be required. Firstly there are accident avoidance features such as improved braking. Then there is a section on protective measures. This is divided between protection for the occupants of HGVs and protection for all other road users likely to be involved in accidents with HGVs.

Accident Situation

For a comprehensive assessment of the European accident situation involving heavy goods vehicles (HGV), it is necessary to come to common definitions of HGV weights, sizes and other aspects.

It is reasonable to use already existing common definitions as far as available, e.g. within the Directives of EEC, where uniform definitions for HGV sizes, weights and axle loads will become effective gradually by 1991 (85/3/EEC).

This report discusses only vehicles for the transport of goods with a gross weight of more than 3.5 tonnes. As far as possible the corresponding figures for different countries are split up in this way. Figures for buses and in special cases for cars too, should underline the comparisons for specific situations for HGVs.

In Appendix 1 a detailed comparison is given between the national regulations of Great Britain and Germany. The use of the international provided rules shows that in general nearly the same differences exist between the classes of speed limits and driving licences. In Great Britain there is a more varied classification for HGV licences and there are higher maximum speed limits for HGVs. In order to compare the accident situation in the European states contributing to this report, it would be necessary to consider details of the relevant figures and of accident rates and special accident risks. In Table 1 a comparison is made over eight years of European figures for registration and mileage of HGVs (gross weight over 3.5 tonnes) to give some information about the increasing presence of HGVs on European roads.

The trend of slight increasing figures of usage is combined in nearly all countries with decreasing accident figures. This reflects in principle the effectiveness of rules and regulations in the area of road and road equipment, training and monitoring of the drivers, the regulations for vehicle design and its supervision.

In Appendix 2 a review is made for Great Britain and Germany about the development of national roads with reference of increasing length and width of different road categories. These figures show a further possible reason for the decreasing number of accidents, namely the increasing length of Highways and Motorways and the widening of other roads.

There is also the possibility of a great influence of technical improvements made on HGVs and of obligatory technical inspections upon the accident figures. Technical improvements for safety are discussed later in this report. In Reference 3 and Appendix 3 some European figures are available for an estimation of the technical inspections made on HGVs which suggest that in Germany the number of defects found on trucks has declined, although this has not happened for cars.

All those important factors of road traffic in the countries considered cannot give a complete reason for the common trend of decreasing figures of fatalities and injured persons, as shown in Table 2.

Other road users are endangered to varying extents by trucks in the event of accidents. This circumstance can be studied in Table 3, where the fatalities are shown by category of road users in HGV accidents in a European comparison. The largest numbers of persons killed in the countries listed are in accidents of HGVs versus cars; this is certainly caused by the great number of cars in traffic. But there are obvious differences between France and Sweden on one side and Great Britain and Germany on the other. In France and Sweden with a greater amount of traffic outside built-up areas it seems to be that accidents of HGVs versus cars produce injuries of greater severity; the lack of figures for comparison does not allow a more concrete statement. The same circumstances probably give a reason for the low percentage of injured HGV occupants in Germany in comparison with the others.

For Germany the low figures of injured or killed HGV occupants might be also founded on the high mileage on highways and the high safety level of such roads.

Table 1. HGV population and distances travelled

Country	Gross weight ≥ 3,5 t Road User	Indication	1976	1978	1980	1982	1984
France	Trucks (Rigid)	Number of vehicles			345 o14		287 o61[1]
		Distance travelled (Mill. km)					
	Tractors (Articulated Vehicles)	Number of vehicles			123 893		133 426[1]
		Distance travelled (Mill. km)					
Germany	Trucks (Rigid)	Number of vehicles	501 746	617 221	659 169	640 221	621 531
		Distance travelled (Mill. km)	11 189	13 825	14 765	13 509	13 114
	Tractors (Articulated Vehicles)	Number of vehicles	130 105	147 628	171 143	188 400	209 179
		Distance travelled (Mill. km)	3 667	3 689	3 934	4 657	4 891
Great Britain	Trucks (Rigid)	Number of vehicles	433 3oo	384 8oo	374 9oo	3474oo	346 5oo
		Distance travelled (Mill. km)	11 396	11 164	1o 455	1o 425	1o 382
	Tractors (Articulated Vehicles)	Number of vehicles	1o2 8oo	1o4 8oo	99 7oo	88 1oo	9o 7oo
		Distance travelled (Mill. km)	5 477	5 953	5 474	5 549	6 o22
Sweden	HGVs[2]	Number of vehicles	79 o79	82 o16	83 321	82 464	85 6o9
		Distance travelled (Mill. km)			2 8o5		

[1] with or without trailers (semi-trailers), estimated on 1st January, 1985.

[2] Motor HGVs (not trailers) with maximum permissible gross weight above 3500 kg, registered on 1st January.

The proportion of killed pedestrians is nearly the same in Great Britain, Sweden and Germany but in France it is clearly lower. Compared with Germany and Sweden and to a lesser extent in Great Britain the percentage killed of the unprotected road users is very low in France—but here we have the highest absolute number for motorized two-wheeler riders killed.

In the Nordic countries the accident risk increases more for HGVs than for cars when the road surface becomes more slippery. Snow or ice is a major environmental factor also in absolute figures, since it was present in about every second HGV accident during a whole year period according to data from the Swedish National Road Administration. It can be assumed for HGVs that the low coefficient of friction and the inferior yaw stability in association with poor brake force distribution causes an overrepresentation in accidents on slippery roads.

The problem of inferior yaw stability is also a problem in France, where a lot of traffic is on roads with insufficient width for driving manoeuvres. In the United Kingdom there is more concern about the braking power available on wet roads at high speeds but accidents on urban roads are also important.

In Germany serious problems are given by the possibility of high speed driving on highways (Autobahn). Investigations of BASt in 1985 give some figures for the speed of HGVs. The background for these measurements was equally in relation to roads and road equipment, traffic volume and so on. The Investigation[1] resulted in following figures: 16% of all HGVs drove within the allowable speed of 80 km/h, about 60% had a speed between 80 and 90 km/h, 20% were measured with a speed between 90 and 100 km/h and 2% ran over 100 km/h. Almost every fourth HGV was faster than 90 km/h.

In connection with inclemency of the weather this high speed level can cause several accidents with a large number of vehicles involved, as occurred in Germany in 1985.

Safety Measures

As pointed out above, official statistics from many countries indicate that Heavy Goods Vehicles are overrepresented in fatal accidents. In fact, the absolute numbers of fatalities involving HGVs are alarming enough to motivate special research on this category of vehicle and road user.

Table 2. Comparison of accident figures in Europe

Country	Indication	1976	1978	1980	1982	1984
France	Accidents with personal injuries				223 162	202 016
	HGVs involved					11 609
	Fatalities				12 102	11 685
	HGVs involved					1 773
	Injured occupants				312 822	284 905
	HGVs involved					15 272
Germany	Accidents with personal injuries	359 694	380 352	379 235	358 693	359 485
	HGVs[1)] involved[2)]	27 938	28 191	26 297	22 264	22 841
	Accidents with fatalities	13 550	13 368	11 911	10 581	9 304
	HGVs[1)] involved[2)]	1 715	1 736	1 464	1 223	1 113
	Accident involving serious and slight injury	346 144	366 984	367 324	348 112	350 181
	HGVs[1)] involved[2)]	25 223	26 455	24 833	21 041	21 728
Great Britain[5)]	Accidents with personal injuries	258 639	264 769	252 300	255 980	253 183
	HGVs[1)] involved[2)]	13 800[6)]	13 858	11 417	10 589	10 171
	Accidents with fatalities	6 006	6 304	5 560	5 447	5 138
	HGVs[1)] involved[2)]	770[6)]	752	624	620	591
	Accident involving serious and slight injury	252 633	258 465	246 740	250 533	248 045
	HGVs[1)] involved[2)]	13 030[6)]	13 106	10 793	9 969	9 580
Sweden	Number of accidents with injuries	17 043	16 028	15 231	15 288	16 531
	HGVs[3)] involved[4)]	1 288	1 145	1 132	1 019	1 072
	Number of accidents with fatalities	1 035	911	755	681	717
	HGVs[3)] involved	181	162	121	116	119

1) All HGVs without weight limit (in GB, goods vehicles greater than 1,5 tons unladen weight)
2) Only accidents with one and two parties involved
3) HGV with maximum permisseble Gross Vehicle Weight above 3.500 kg
4) Number of involved (not only primarily) vehicles
5) England, Scotland and Wales only (does not include Nothern Ireland)
6) These are numbers of accidents involving HGVs

Though a general decline may be found in the total number of traffic accident fatalities, the relative aggressiveness of HGVs (when compared to cars or other motor vehicles) seems to be comparatively constant. In fact, data from a U.S. review 1985, "Big Trucks", from the Insurance Institute for Highway Safety, IIHS (Watergate 600, Washington D.C.) exhibit a slight increase of HGV aggressiveness in the last decade: "In 1977, a car occupant was 26 times more likely than a truck occupant to be killed when the two vehicles crashed. Now the ratio is 35 times more likely."

During the last decades, some international seminars and multilateral reviews have been devoted particularly to HGV safety, for instance by HSRI (1975, later University of Michigan Transportation Research Institute UMTRI, Ann Arbor), OECD (1977), and OECD (1983). More recent studies on HGV safety have been (NHTSA, 1986) and will be published by the National Highway Traffic Safety Administration (NHTSA), since 1985 arranging a special session on HGVs in its bi-annual Technical Conferences on Experimental Safety Vehicles (ESV).

The operating conditions of HGVs are quite different from light road vehicles. In addition, the great dimensions and great (variations in) weight of HGVs have lead to design principles deviating substantially from cars. Many of these HGV peculiarities deteriorate their safety performance to an extent that probably would not be tolerated in more commonly known road vehicles.

Safety-related quantities and possible technical countermeasures have been studied experimentally, and sometimes in real traffic, by a number of

Table 3. Casualties in accidents involving HGVs

Country	Severity of Injury	Number of Casualities: Pedestrians No.	Pedestrians %	Pedal Cyclists No.	Pedal Cyclists %	Motorized two-wheelers No.	Motorized two-wheelers %	Car Occupants No.	Car Occupants %	HGV Occupants No.	HGV Occupants %	Others No.	Others %	Total (100%)
France	fatal serious slight	144	8,1	79	4,5	176	9,9	1154	65,1	159	9,0	61	3,4	1773
Germany	fatal serious slight	176 704 855	14,5 9,8 5,0	167 803 1532	13,8 11,2 8,9	139 1068 1724	11,5 14,9 10,1	618 3867 11324	51,0 53,9 66,1	70 481 1029	5,8 6,7 6,0	41 255 655	3,4 3,6 3,8	1211 1) 7178 1) 17119 1)
Great Britain	fatal serious slight	104 267 463	15,1 8,0 5,5	52 184 366	7,5 5,5 4,3	84 393 526	12,2 11,7 6,2	352 1689 4487	51,1 50,4 52,9	56 517 1589	8,1 15,4 18,7	41 299 1050	6,0 8,9 12,4	689 4) 3349 4) 8481 4)
Sweden 2)3)	fatal serious slight	21	14,4	17	11,6	5	3,4	81	55,5	21 2)	14,4	1	0,7	146

1) HGV with each gross weight, excluding accidents with more than two parties involved.

2) Number of killed persons in accidents where goods vehicles (irrespective of weight)

3) were primarily involved during 1985.

4) In accidents involving HGVs over 7,5 tons gross weight.

investigators in Europe. Below, a few of them will be referred to directly or indirectly. However, in the EEVC HGV Working Group we hope that our unintended ignorance of other studies will stimulate the investigators in question to contact EEVC-representatives and to submit papers to the HGV session in future ESV conferences.

Accident Avoidance

Involved Institutions. In the HGV session of the tenth ESV Conference 1985, original research results on HGV and bus accident avoidance properties were presented from European institutions such as: TUV Rheinland (Institute for Traffic Safety) in FRG; Renault Vehicules Industriels in France; Cranfield Impact Centre in U.K. From other publications it is known that important experimental research on the active safety of HGVs also has been conducted recently: in FRG at TUV Essen and at the Technical Universities of Aachen, Berlin, Braunschweig, and Hannover as well as at HUK-Verband (insurance company cooperation) in Munchen and at Daimler-Benz; in the Netherlands at the Technical University of Delft; in Sweden at Road and Traffic Research Institute (VTI, S-58101 Linkoeping).

Analyses of HGV active safety problems in real accidents have been reported by TRRL (Transport and Road Research Laboratory) and by MIRA (The Motor Industry Research Association) in U.K. as well as by the VTT (Technical Research Centre) in Finland through their Road Accident Investigation Teams.

Accident avoidance contributions to the HGV session of this 11th ESV Conference have been submitted also by Union Technique de l'Automobile, du Motorcycle et du Cycle in France and by the Institute of Technology in Darmstadt, FRG.

Literature references and additional hints on European HGV research into the active safety area may be found in a state-of-the-art survey by Vlk (1985, Int. J. of Vehicle Design, vol. 6, no. 3, pp. 323-361) from the Technical University of Brno in Czechoslovakia.

Experimental studies and theoretical analyses have revealed numerous accident avoidance problems related to the HGVs themselves and their load. Hence, it is essential to identify more precisely the vehicle design and operation variables decisive of HGVs' active safety. Such variables have been listed in numerous studies, and some of them will be elaborated on below. (See OECD paper[9] by Strandberg, 1987)

Yaw Stability of Articulated Vehicles in Non-Braking Situations. The deteriorating influence on yaw stability from articulation was investigated in full scale driving experiments, computer simulations and theoretical analyses in the early '70s at the Swedish Road and Traffic Research Institute, VTI. Though articulation has been introduced in HGV design to improve the manoeuvrability and decrease the inwards off-tracking at low speeds, it was found to impair the handling properties and to increase the outwards off-tracking at highway speeds, when the sideslip angles no longer may be neglected. See fig. 1. Safety

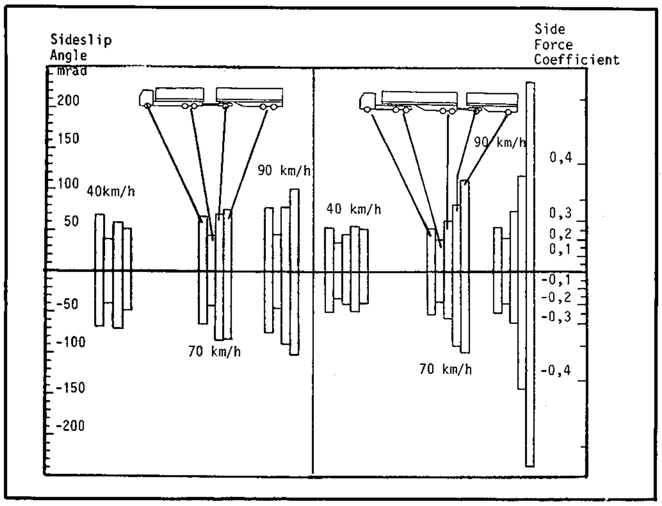

Figure 1. Sideslip angle peaks for different axles of a single trailer (two articulations) and a double trailer (three artics) HGV with the same overall length (24m) when making a double lane change manoeuver. The lateral acceleration at the mass centre of the truck or tractor was 1.75m/s^2 and independent of speed. The scale for Side Force Coefficient (side force divided by normal force) is an approximate average for tyres and loads typical for these HGVs as measured on wet asphalt. Computer simulation data from Nordström, Magnusson, Strandberg (1972, VTI report no. 9)

measures in this context have been experimented with by Woodrooffe, Billing, Nisonger (1983, SAE paper 831162).

Though the lateral friction utilization (or Side Force Coefficient, SFC = Side Force divided by normal force) in fig. 1 is at a moderate level for the triple artic tractor, the rear wheels are skidding at 90km/h with a SFC close to the coefficient of friction for the actual road surface. During this kind of manoeuvre, very light braking may cause sudden and severe skidding at the rear end of the vehicle combination.

Similar results were arrived at with experimental research also at the University of Michigan Transportation Research Institute (UMTRI, previously HSRI) and are supported by others according to the above mentioned survey by Vlk (1985), who reviewed 70 articles on the handling performance of truck-trailer vehicles. These experimental evidences are now supported by the novel results from a case-control study of HGV accidents by Stein and Jones (1987) in fig. 2.

Yaw Stability During Braking. According to schematic descriptions of tyre force characteristics, eq. 1 expresses the approximate relationship between Coefficient of Friction (CoF) and the maximum available (due to road friction) Braking Force Coefficient (BFC = braking force divided by normal force) and

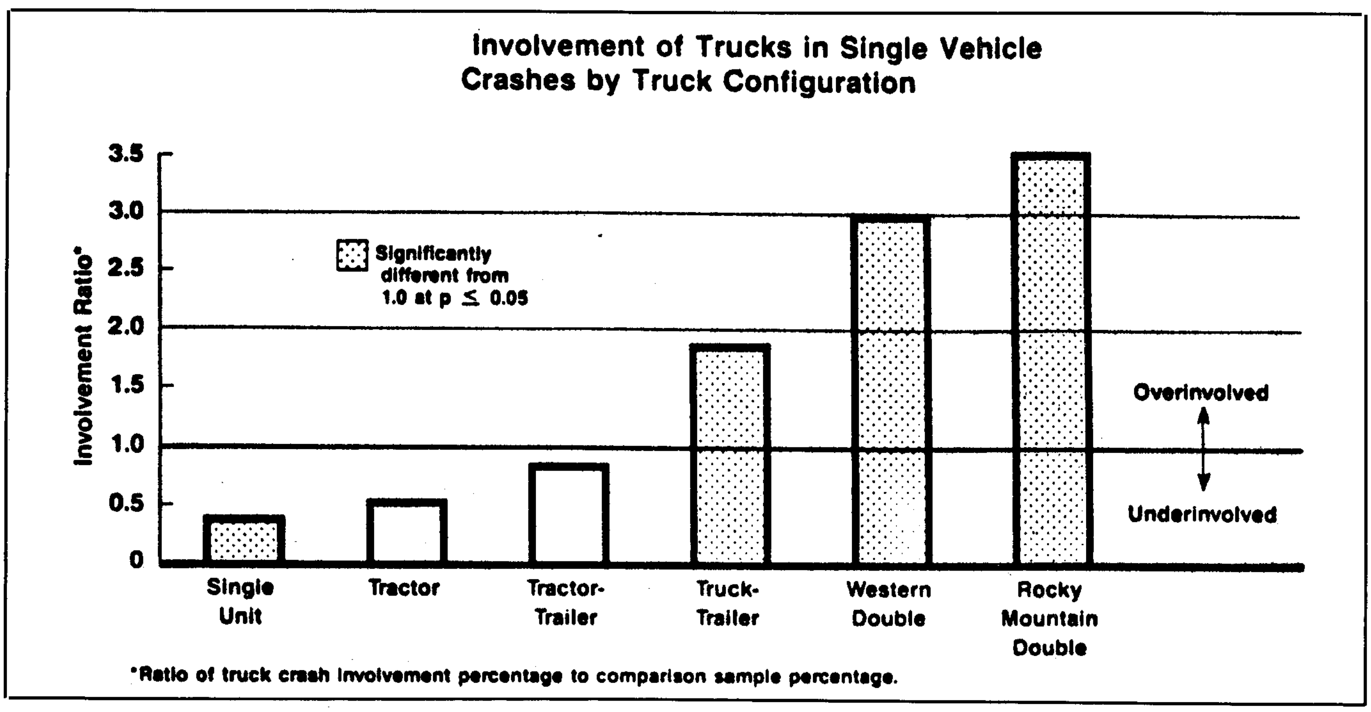

Figure 2. Involvement of HGVs in single vehicle crashes by HGV configuration. Data from 222 HGVs involved in single accidents and from 666 comparison HGVs. From "Crash Involvement of Large Trucks by Configuration: A Case-Control Study" by Howard S. Stein and Ian S. Jones (January 1987) with kind permission from Insurance Institute for Highway Safety, Watergate 600, Washington, DC 20037

Side Force Coefficient (SFC). It is often visualized as the so called friction circle.

$$BFC^2 + SFC^2 < CoF^2 \tag{1}$$

Due to the great rearward amplification of the SFC in a manoeuvering artic (see fig. 1), very small braking forces may then lock up the rear wheels and result in severe skidding, since the tyre force direction becomes indefinite when the unequality in eq. 1 approaches equality. The necessary side force for yaw stability is then no longer available. On icy and very slippery roads, skidding and jackknife accidents may be initiated also by traction or retarder forces.

Skidding will occur even during straight driving, if the wheels at some vehicle end are braking too hard in relation to their load. This will bring the BFC too close to the CoF. Therefore, the driver has to restrict the brake pedal force below the level where the least (relatively) loaded wheels lock up. If no load-compensating device is installed, Table 1 demonstrates that the non-locking braking distance will increase more than twice when only the trailer has been unloaded.

In Table 4 it is assumed that the brake force distribution is constant and adapted to the maximum permissible gross weight in Sweden, where no devices are required for load compensation of the braking torque. Initial speed: 20m/s or about 70km/h. Winter road with CoF = 0.2. Dynamic load transfer neglected. These examples of unloading will lead to trailer wheel-lock if the braking force exceeds 25% of the value, possible to apply at full load.

Braking Performance of HGV Combinations in Traffic. The handling and stability of HGVs is particularly poor during braking. Therefore, a case-control study (similar to the above mentioned by Stein & Jones, 1987) in the four Nordic countries was opened during 1986 with measurements of decisive quantities in the air brake systems and of the braking characteristics of HGVs. In each country, 100 HGV-combinations were randomly selected from the normal traffic flow on suitable roads.

Data for Sweden, now being analyzed, indicate that quite a large proportion of the HGV-combinations in use are unable to reach the minimum deceleration performance required in legislation, i.e. $5m/s^2$ with available air pressure when fully loaded. Wheel-lock and corresponding skidding tendencies (or excessive braking distances) seem to be another major problem in Sweden, where no devices are required for load-compensation of the braking torque. Out of 100 combinations investigated, 75 were completely without such equipment.

In fig. 3 the deceleration values measured on these HGVs have been transformed to the corresponding braking distance from 70km/h at maximum control pressure (6bar). So have the Swedish deceleration requirements for HGVs and for cars. Cars are required to decelerate $5.8m/s^2$ without any wheel-lock, and the rear wheels must not lock before the front ones between 5.8 and $8.0m/s^2$. HGVs must have brakes capable of decelerating the fully laden vehicle at $5.0m/s^2$ (from 60km/h). However, the Swedish rules corresponding to the ECE corridors (restricting the deceleration-pressure ratio in both directions) are rarely checked since the announcement of the general exemption from load sensing valves in the early '70s.

More distinct and general conclusions may be drawn later in 1987, when data from all participating countries are available for analysis. Deceleration performance and skidding tendency will be quantified and compared between the four countries. Substantial differences in these qualities are expected, since very few Swedish HGVs have load compensation, while many Finnish ones have manually controlled pressure-reduction valves, and while Danish and Norwegian legislation requires automatic load sensing valves. Practical valve problems have been reported, and the outcome of the comparisons between countries seems difficult to predict.

Characteristics of HGV Air Brakes Impairing Safety. Apparently, HGV braking properties are even more inferior to that of the cars than what the (quite moderate) demands in legislation permit. This inferi-

Table 4. Non-skidding braking distances with different loading on a typical HGV (such as the truck-trailer in fig. 1a) without load sensing valves

Loading condition	Weights at the different axles (tonnes) Truck Front	Truck Drive	Truck Trailing	Trailer Front	Trailer Bogie	Gross Weight (tonnes)	Braking Force part	Braking Distance (m)
Both loaded	6	8	8	10	16	48	100%	100
Both empty	5	6	Up=0	2.5	4	17.5	25%	175
Trailer empty	6	8	8	2.5	4	28.5	25%	238
Tr.bogie unload	6	8	8	10	4	36	25%	300

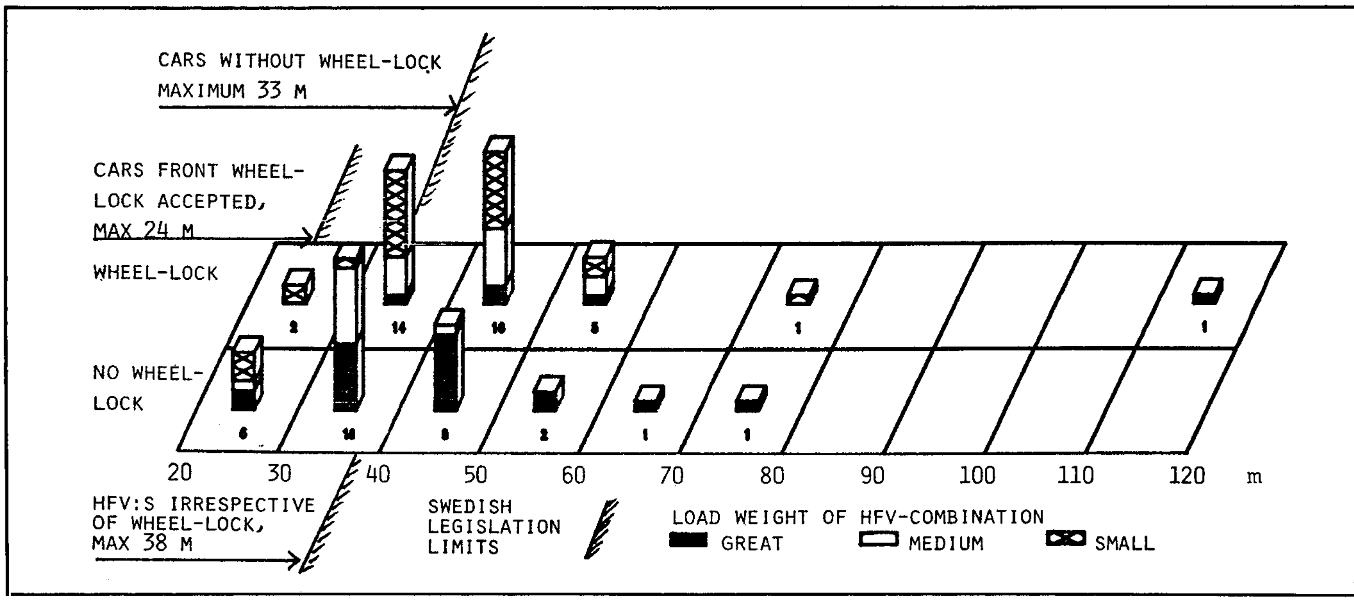

Figure 3. Distribution of estimated non-locking braking sistances from 70km/h in a sample of 75 HGV trailer combinations (without load sensing devices). Random selection from the traffic on suitable roads in Sweden 1986. Preliminary results. Estimate based: (back row) on the smallest recorded deceleration at wheel-lock or (front row) on extrapolation to 6bar control pressure from recorded decelerations in driving tests at 3bar and at 4.5bar, if no wheel-lock was detected

ority in deceleration performance and in wheel-lock resistance (i.e. yaw stability) may be considered a natural consequence of contemporary air brake design, if not equipped with wheel speed sensors of the anti-lock type. These unwanted characteristics deteriorate the control accuracy and delay the response of the brake system, thereby reinforcing its open-loop nature.

Most of the transient and steady-state deviations (from the ideal and theoretical relationship between the control air pressure and the Braking Force Coefficient at every wheel) attenuate the brake torque. Therefore, insufficient deceleration performance may be seen as a secondary consequence of the great brake force variations within and between wheels. Consequently, it seems more appropriate to reduce the brakes' sensitivity to varying operating conditions, and to improve the control system properties than to increase their peak force and to introduce more valves or open-loop components.

A list of safety impairing characteristics particular for HGVs and air brakes is given below without any rank order. Apart from by national governmental bodies, many of these problems are considered by the Economic Commission for Europe, Group of Rapporteurs on Brakes and Running Gear (ECE-GRRF), by the Society of Automotive Engineers (SAE), and by the International Organisation for Standardization (ISO/TC22/SC9). An overview of the technical and committee work issues in this context was made by Nordström (1983, VTI report no. 257).

a) Air brakes have substantially longer response times than hydraulic brakes, due to the compressibility and comparatively low wave propagation velocity of air. In addition, HGVs have long expandable air hoses and a long distance from the control valve at the brake pedal to the farthest wheel brake chamber. The maximum permissible response time for the worst brake chamber in the trailer is 0.8s according to the Swedish regulations (corresponding to ECE no. 13).

b) A number of pressure modifying valves, air lines, and connectors may be installed by one (e.g. a trailer) designer in a way that was not predicted by another (e.g. the axle-designer).

c) To reduce the brake wear of their own vehicles, it is said that some owners install optional valves at the air coupling: the truck/tractor owners enhance the pressure to alien trailers; and the trailer owners attenuate it.

d) Many sequential brakings may consume air to such an extent that the spring brakes are automatically applied, which may cause surprising wheel-lock and dangerous skidding.

e) Load sensing valves of the mechanical type (transforming axle suspension motions to pressure reductions), if mounted, are often partially or completely out of order due to their tough operating conditions. Many types are not designed to change their pressure input/output ratio quickly enough upon dynamic load transfer, which may be substantial with high and short vehicles.

f) The brake chambers are not linear (pressure to force) transducers. When the diaphragm stroke and push-rod travel becomes long at some wheel, the force declines without any easily visible indication to the driver.

g) Since a brake lining may be "glazed" and lose much of its friction when its brake power dissipation is small in a number of brake applications, anti-lock brakes seem more favourable than load sensing valves even in this respect.

h) The trailing shoe is particularly susceptible to glazing, since the leading shoe makes more of the braking work from the beginning. Differences between the leading and trailing shoes may therefore be exaggerated, making the whole brake less efficient and more likely to fade or to lock up.

i) Great variations in temperature and friction coefficients for different linings on the market add further balancing problems when some linings have been worn out.

j) Overheating and eccentricity problems are more pronounced with drum brakes than with disc brakes. However, the peak force of air actuated disc brakes are often less than what is demanded in a HGV.

These and other peculiarities of HGV brakes make it seem very unlikely that any conventionally braked HGV (truck, tractor or trailer) would keep the originally intended brake force balance during its whole lifetime. Considering that many other vehicles also may be coupled to it, closed-loop (anti-lock) brake systems appear to be essential for the braking safety of articulated HGVs.

Roll Stability. Apart from increasing the overturning risk, the high position of the mass centre in relation to the track width results in great lateral load transfer to the outer wheels. Since the SFC decreases with increasing tyre load, this transfer will also aggravate the skidding tendency. In addition, the great lateral forces on the outer tyres will reduce the effective track to considerably below the nominal value.

The roll stability becomes particularly poor if the load is unrestrained laterally, such as in many partially loaded road tankers. The lateral acceleration at overturning may be raised more than twice at certain steering frequencies, if longitudinal cross-walls are mounted. See Strandberg (1978, VTI report no. 138).

In highway speed manoeuvres the trailer is often moving with a greater lateral acceleration than the towing vehicle. Therefore, it is particularly unfortunate that the mass of the trailer mostly is higher above the road than that of the towing truck.

Space Demand. The great dimensions of HGVs increase the probability of collisions, particularly on narrow roads and in urban areas. The great length of each vehicle unit means that comparatively moderate sideslip angles will result in considerable lateral deviations for the rear axles. Such deviations may be aggravated at highway speeds, if articulation or axle steering is introduced (to decrease the inwards off-tracking at low speeds).

The large front and side areas may induce dynamic air forces hazardous to both the unloaded HGV itself and to other road users.

Indirectly Contributing Risk Factors. A number of design-, maintenance- or load-related factors affect the accident risk indirectly. For instance, the splash and spray from a HGV may contribute to an accident without any HGV involved or present. However, this paper will not deal further with this matter, due to the difficulties to determine the effect on safety from these factors and their relative importance.

Methods to Identify Relevant Accident Avoidance Parameters in HGVs. In general, deviations in reporting routines and tendency make it difficult to find reliable numbers on travelled distance and on non-fatal accidents for valid risk comparisons between countries or between different vehicle types. Though statistics show that HGVs have considerably less accident risks than cars, the fatality risk (fatal crashes per 100 million miles) was similar to that of cars for single unit trucks and substantially greater for articulated HGVs in a U.S. study by Eicher, Robertson, Toth (1982, NHTSA no. HS-806-300).

Several studies indicate that single artics (tractor-semitrailer combinations) have better highway handling properties than double artics (truck and full trailer) and triple artics (double bottoms). Nevertheless, their greater low speed off-tracking may impair the safety for unprotected road users in urban areas. The smaller dimensions and weight of tractor-semitrailers may also lead to a greater mileage in urban areas compared to truck full trailer combinations. In addition, the yaw stability during braking may be poor with semitrailers as compared to full trailers. In the U.K. for example, many years ago jackknifing occurred in about 15% of articulated vehicle accidents. Load sensing valves were then fitted to tractor rear axles and the incidence of jackknifing reduced to no more than about 5 per cent.

Effects like this may have contributed to the higher accident risks for tractor-semitrailers (as compared with truck-trailers of similar length) found in a Swedish study of long vehicles by Trafiksäkerhetsutredningen (1977). At that time Sweden considered a reduction of the maximum permissible length of vehicle combinations from 24 to 18 metres. However, the 24-metre limit appeared safer from several viewpoints and the 18 metre idea was abandoned.

Though Trafiksäkerhetsutredningen compared the single and double artics in many ways, the poor matching of exposure and accident data in official statistics made it virtually impossible to isolate the relevant parameters in the vehicles themselves. A more suitable method for this purpose is the case-control study technique, often used in epidemiology. An accident group of vehicles is then compared with a control group passing the accident site at about the same time as the accident occurred. The basic idea is that significant group differences found in design-, load-, maintenance-, driver-, and employer parameters reflect safety relevant factors associated with the vehicles themselves, since both groups have been exposed to the same environmental risk factors.

Such a case-control study was conducted by Stein and Jones (1987, see Figure 2 above) on interstate highway crashes during two years in Washington State. Their results indicate clearly that certain vehicle parameters, such as the number of articulations, may be even more decisive of safety than driver parameters. This adds further doubts against the common conclusion that driver education is more important than vehicle design improvements.

Accident Avoidance Determined by Vehicle Design and Compatibility. Similar conclusions are often drawn on the basis of ambiguous results from accident investigations, stating that vehicle factors play a negligent causal role compared to human factors. No causal factor can be identified, if one does not know about its existence in general, and if one does not search for it. Many of the factors mentioned above are of that kind, particularly those hazards that are not considered in legislation. In addition, accidents are multicausal phenomena by definition.

Therefore, it is impossible to find really objective figures on the distribution of accident "causes" between drivers, vehicles and traffic environment. If such global cause categories are used, the presented figures tell more about the investigators and their methods than about the actual accidents. The important thing is to improve safety by the best measures accessible to ourselves, and to withstand the temptation to blame the accidents on factors that we cannot affect.

This problem is particularly pronounced for HGV combinations, even if one accepts that vehicles are easier to change than basic human behaviour. The towing vehicle and the trailer are often designed by different manufacturers and sometimes they also have different owners. Therefore, unusual efforts on the compatibility aspects are required by the involved parties to improve the accident avoidance properties of the whole HGV combination.

Injury Protection

The tables earlier in this paper show that the total numbers killed per year in accidents involving heavy goods vehicles varied considerably within the European countries, the proportions of most of the different categories of road user were similar. The exceptions were that France had a smaller proportion than average of pedestrian and pedal cyclist fatalities whereas Germany had a higher proportion of pedal cyclists.

The laboratories in Europe which have been working on injury protection measures for accidents involving Heavy Goods Vehicles are INRETS (LCB) in France and TRRL in the UK. A detailed investigation of 25 collisions between trucks and cars by INRETS[4] have confirmed the leading features of such incidents. TRRL reviewed all fatal accidents in one year in Great Britain involving these vehicles[5]. These studies form much of the background to the following comments.

In these accidents a heavy goods vehicle occupant is much less likely to be injured than other road users. Accidents that injure occupants involve either an HGV as a result of rollover or an HGV when it impacts a solid object, or when an HGV collides with another large vehicle.

The largest single category of road user at risk in all countries in accidents involving HGVs is the car occupant, followed by the various unprotected road users (pedestrians and two-wheeler users).

HGV Occupant Protection. The main causes of fatal injury to these occupants is either by ejection from the cab or by crushing of the cab structure.

Considering firstly ejection, this frequently occurs in single vehicle accidents, particularly as a result of rollover or impact with a solid object. Ejection through the windscreen is common, an occurrence made easier by the fact that only small amounts of deformation of the cab structure causes the windscreen to break or fall out. An obvious remedy is for the occupants to wear seat belts, but the belts would either have to cater for the larger vertical movement of HGV seats that is usually present compared with that of car seats, or belts integral with the HGV seat would be needed. A British Study[6] of fatal accidents involving HGVs suggested that lap and diagonal belts could prevent over 30 per cent of fatalities to all HGV occupants, and a simpler lap belt only, just under 30 per cent.

It has been suggested that HGV drivers might object to wearing belts because they are concerned about loads coming forwards through the rear of cabs in frontal impact accidents. An alternative solution might therefore be to use air bags, although these would possibly prevent ejection only in frontal impacts and during the initial impact. In relatively slow accidents such as rollovers where subsequent ejection might take place they may be less effective than seat belts.

Stronger cab structures would help prevent some of the injuries due to crushing. They would be more beneficial in many of the single vehicle accidents where the crush forces are likely to be lower than when other large vehicles are struck. As an example, if an unladen platform type HGV rolls over it is often the cab roof that collapses and crushes the occupant. More substantial pillars connecting the roof to the rest of the cab together with more secure glazing might also prevent the windscreen coming out and therefore reduce the incidence of ejection. However strong the cab structure, it is difficult to envisage much protection being offered in the far more violent HGV to HGV type of accident or when the HGV impacts a very solid object such as a bridge parapet or other roadside furniture. However the first design of the Leyland National bus did have the structure around the driver locally strengthened so that it displaced backwards without crushing the driver in frontal impacts.

At present only Germany and Sweden have requirements for cab strength.

Car Occupant Protection. Heavy goods vehicles are very aggressive towards cars in collisions. The large mass ratio ensures that the velocity change of the car is much greater than that of the truck. The height of structure around the perimeter of most trucks is such that when it strikes or is struck by a car, the car can under-run the truck, often to the extent that the truck structure comes into direct contact with the car occupant. By the same mechanism the important energy absorbing zones of the car, which tend to be below the truck structure, are not used to their best advantage. Finally, the rigidity of the truck structure ensures that most of the energy of the impact is dissipated in the car structure rather than in that of the truck.

Typically, in European countries, the distribution of impact of cars around the truck perimeter is that approximately 60 per cent or more impact into the front of the truck (usually front of car to front of truck), around 25 per cent or more into the sides and up to 15 per cent into the rear.

In all these types of impact, the important primary objective is to provide a strong structure or guard, fitted to the heavy goods vehicle, which is low enough to impact the car structure. Two advantages are then gained by the car occupant—the truck structure is kept away from the car passenger compartment and the energy absorbing properties of the car are utilised. The latter is only a real advantage if the car occupant is wearing a seat belt. A secondary but important desirable objective is to make the guard or structure which strikes the car energy absorbing. For this to be effective the forces needed to deform the guard should be compatible with those required to deform the car structure. Energy absorption introduced in this way has the potential to increase the maximum survivable speed of impact for the car occupants.

The relative importance of special low structures or guards at front, sides and rear depends on several factors. It is difficult to justify sideguards fitted to trucks to protect car occupants. They would of necessity have to be fitted to both sides of trucks and, because of their length and strength requirements, would be very heavy. Because of the weight and payload penalties incurred and the relatively small benefit in terms of lives and injuries prevented, they would probably not be cost effective.

Guards and special low structures fitted to the front and rear of the heavy goods vehicle are more likely to be cost effective because of their lighter weight and their potential, at least as far as the front guard is concerned, to save more lives. The mechanism of impact of cars into the fronts and rears of trucks is similar except that there is usually a greater degree of under-run at the rear of the goods vehicle because of its high structure and space under the rear. Also the speeds of impact of cars into the fronts of trucks are usually higher. Because of the much larger number of car occupants killed in impacts into the fronts of trucks compared with those killed in impacts into the rear, there is a stronger case for the fitment of protection for cars at the front.

A British study in 1985[7] suggested that an estimated 60 car occupants might be saved out of about 2,000 killed each year in Great Britain. This was based on fitting energy absorbing front under-run guards and an experimental impact test programme suggested that such guards could offer protection to seat belted car occupants at closing speeds up to 65 km/h.

The concept of including energy absorption in guard design is important, particularly in front of truck to front of car impacts where closing speeds are higher. There is however probably a limit in the amount of energy absorption that may be provided. The linear crush of the car plus the crush of the guard must not exceed the original length of the bonnet of the car otherwise the upper structure of the truck may impinge on the car occupant compartment. Also the forces necessary to deform the truck guard must lie within the range that will also deform the car structure. These two factors may well imply that the design of the low front or truck guard is closely determined by the dimensions and crush forces of small cars. The previously mentioned British study showed that for small car impacts it was possible to utilise at least 25 kJ of energy absorption built into a guard.

The ground clearance of the guards is also important and would need to be about 300 mm for good performance, and certainly no more than 400 mm, otherwise the structural parts of smaller cars would be overridden.

As yet there are no European countries that require the fitment of front protection to heavy goods vehicles. Many countries do however have a requirement to fit rear protection, mainly to meet EEC Directive 70/221. These countries include Austria, Belgium, Great Britain, West Germany, Italy, Luxembourg and the Netherlands. Sweden has also had a mandatory requirement to fit rear guards, to their own specification, since 1973. France also has a national requirement. It has not yet been possible to estimate what its effectiveness has been, but a large improvement is not expected because the incidence of cars striking the rears of trucks seems to have been declining over the years.

With the increased use, throughout Europe, of seat belts in cars, the case for fitting trucks with both front and rear protection is now much stronger. Considerable research has been done and is being done in countries such as Germany, Sweden, France and Great Britain to develop and promote effective structures and guards.

In France in 1985, the "Laboratoire des Chocs et de Biomecanique" of INRETS started a new experimental research programme on compatibility between cars and HGVs in frontal impact. The first phase consists in determining the geometry of an "average" HGV. This work based on the sizes of 20 trucks is completed and suggests the dimensions of a rigid barrier to simulate a truck front end. In the second phase which is in progress several cars will be tested against this obstacle at two impact speeds: 50 and 57 km/h. Already one car model has been tested at the two speeds. The first results seem to show that the injury parameters currently used to evaluate the protection of car occupants are not sufficient, and are not really valid for such a crash configuration. The third phase will lead to the proposal of a methodology for the evaluation of compatibility between trucks and cars.

In the Federal Republic of Germany the Technical University of Berlin has investigated the performance of energy absorbing low front bumpers to protect cars against under-run.

In Great Britain the TRRL is developing a test procedure for evaluating frontal impact protection on trucks and heavy vehicles. It consists of a set of dynamic impact tests using an impactor of 250 kg which strikes the truck front at several different points across the width at low bumper height.

Pedestrian, Pedal Cyclist and Motorcyclist Protection. The unprotected road user, as this group is collectively known, is mainly at risk from being run over at the sides of the trucks. The cutting-in of articulated vehicles as they turn sharply can trap pedestrians and others. In the Netherlands a down looking nearside mirror is required which may help drivers avoid this. Pedal cyclists are also at risk in normal overtaking manoeuvers when they may topple towards the vehicle as it overtakes them and fall under the side and be run over by the wheels.

Lightweight sideguards would help prevent running over but their design should be considered carefully in order to make them effective.

A large proportion of accidents where pedestrians or two-wheeler users fall into the path of the truck wheels occur, not surprisingly, in urban areas and often at quite low speeds. Guards need to be strong enough to withstand normal everyday use but from an accident point of view, need not be very strong. More important requirements are that, if the guard is a horizontal rail type of structure, the rails should be close enough to prevent the road users falling through them and the supporting structure should be recessed to prevent injury. The whole guard should also be close to the outside edge of the vehicle or trailer to reduce head injury caused by the aggressive protrusions such as loading hooks often to be found in that area. Ground clearance is possibly the most important factor and it should be as low as operating conditions will allow. A maximum height of the lowest edge of the guard from the ground of about 400 mm would minimise the tendency of a road user to roll under the guard.

All European countries have similar problems in protecting their pedestrians and two-wheeler users and should benefit from the fitting of sideguards to HGVs. At present only France, the Netherlands and Great Britain have a requirement for them to be fitted. The subject of sideguards is however being considered by ECE Geneva with the intention of producing an ECE Regulation to specify their design and performance. Their potential for saving lives will of course vary slightly from country to country but as an example it has been estimated that in Great Britain about 50 lives per year could be saved by fitting effective sideguards, but less stringent requirements could lead to this being more than halved.

Priority of Measures to Provide Injury Protection. The following is suggested for the order in which measures to provide injury prevention should be introduced. It is based only on the liklihood of saving lives rather than on cost benefit considerations and it assumes that nearly all car occupants wear their seat belts.

1. *Front under-run measures (guards or low structure)*

   Energy absorption should be incorporated and ground clearance should be around 300 mm.

2. *Sideguards*

   Lightweight structures are needed with a ground clearance of no more than 400 mm and several other detailed but minor requirements if their effectiveness is not to be needlessly reduced.

3. *HGV occupant protection*

   Seat belts suitable for trucks would be a very cost effective means of preventing ejection if they were to be worn by drivers. Air bags might be an alternative but might be less effective in some ejection situations. Stronger cab structures are also desirable to reduce the injuries due to crushing and would also make seat belts more worthwhile.

4. *Rear under-run guards*

   The priority here is less than for front guards but similar requirements are needed. The primary need is to prevent under-run but energy absorption can also be beneficial.

Methods of Introducing Injury Protection. With the exception of HGV occupant protection, the measures to protect other road users are of little benefit to the truck operator. He has to pay for the devices. The only economic advantages lie in the possible reduction of aerodynamic drag by suitable design of the guards fitted to the front and sides of the truck. For example the front guard could incorporate an air dam[8] and the sideguards could be of skinned design. Both these ideas have been shown to reduce drag and also to reduce spray around the vehicle. The only other advantage is that front and rear guards may reduce the extent of damage to the truck itself.

These factors may not be sufficient to persuade many operators to fit protection, in which case compulsory fitment through legislation is the only certain way to improve the situation. This method is never popular with operators or manufacturers even if there is a strong case for the proposed legislation. It may be possible occasionally to trade off benefits to the operators in exchange for enhanced safety. For example, in Great Britain in 1984, sideguards and rear under-run guards were introduced at the same time as an increase in the permissible maximum gross weight. Although this eased the introduction of safety features there was opposition from many sources to the increased weight.

Conclusions

The last ten years has seen a continuation of the improvements in the accident situation for Heavy Goods Vehicles in most European countries which had been apparent during the previous ten years. Progress in reductions in casualties has been made in different countries at different times in this period and not always for reasons that are readily apparent. To some degree these results are a consequence of changes in the transport of goods. In some countries the numbers of goods vehicles has increased while in others it has decreased. Total distances travelled have been fairly stable but there has been a tendency to increase. It is clear that in many countries there has been a tendency towards higher capacity vehicles. As a result the total of goods transported has probably increased. Generally increases in goods traffic have been greater in European countries outside the EEVC membership.

Precise comparisons in accident trends between countries are difficult to determine because of differences in the minimum size of vehicle regarded as being a Heavy Good Vehicle. However even between the four countries studied in detail (France, Federal Republic of Germany, Sweden and Great Britain) there is a factor of at least two to one between the highest and the lowest fatal casualty rate measured against the various base statistics. The reasons are clearly a combination of geographic and demographic factors combined with goods vehicle factors such as the amounts of goods transported by road, the road user behaviour prevalent in the different countries. The lowest national fatal accident rate per million km travelled by HGVs (1 and 2 vehicle accidents only) is about 0.04.

In the paper the fatal casualties in HGV accidents are sub-divided according to their road user category. About 10% are occupants of the HGVs, while car occupants make up between 50% and 65% and pedestrians about 15% (although only 8% in France). The remainder are mostly riders of two wheelers. These figures suggest that there are five main factors determining the accident rates for these heavy vehicles—

- Road (design and conditions).
- HGV (design and operation and divided between accident avoidance and protective features).
- Other vehicles involved (design and operation and divided between accident avoidance and protective features).
- HGV drivers (skill, behaviour and operational conditions).
- Other road users involved (speed and behaviour).

This report deals only with HGV vehicle factors from among these five.

The review of accident avoidance possibilities for HGVs starts with a list of institutions interested in their study. It generally concludes that there is a large diversity of design practice and requirements in Europe, never-the-less progress in this aspect of safety is readily possible.

The stability in yaw of the multi axle HGV is shown to depend on the load balance between its many wheels in relation to the side force demands made on them when cornering and in conditions of low grip. The problems increase as the numbers of articulations and axles are increased.

These problems are compounded when braking is involved. The brake distribution has to be set and the maximum braking available without locking wheels is dependant on many factors which are listed. Various systems of valves to alleviate the situation are mentioned as are the limitations of current air brake systems. It may be concluded that anti locking brakes and electronic control of the braking of each wheel are essential if full braking with stability is to be achieved.

Limited stability can lead to an HGV not always following in the path taken by its driver in difficult conditions and this leads to accidents, especially on narrow and on crowded roads.

Roll stability is limiting when the load is placed high and also for fully and partially loaded tankers. Lowering the height of the load is perhaps the only viable design improvement.

The review of injury protection possibilities for road users involved in accidents with HGVs showed that for their occupants the most desirable measures are those to prevent ejection and crushing within the cab. Seat belts might prevent 30% of all fatalities if worn. Several countries require strengthening of cab structures in overturning but the great need is to better resist longitudinal crushing on to the driver in frontal impacts.

Car occupants whose cars strike HGVs can best be protected by front and to a lesser extent rear underrun bumpers or low structures. Studies are well in hand for front underrun protection, while rear bumpers are already required.

Unprotected road users can best be helped by the provision of side guards to prevent them falling under the sides of HGVs and then being run over. Several European countries require them to be fitted and the ECE Geneva organisation is working towards a Regulation. As for the underrun protective features the overall effectiveness is much increased by careful detailed design of the guards.

It is concluded that the design of Heavy Goods Vehicles can further be improved in a number of different ways to reduce its contribution to the road casualty situation. Most of these features are being studied at either the research or the design stages.

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Twelve additional references are given directly in the text on Safety Measures and on Accident Avoidance.

Appendix 1 (a)

GB National regulations for operation of vehicles

Motor Vehicle Category	Driving Licence Class	Speed limits (km/h)			Vehicle Weight
		Motorway	Dual Carriageway	Other Roads	
Passenger cars	Car	112	112	96	Less than 7.5 tonnes gross.
Public service vehicles (buses)	PSV[1]				Unladen weight exceeding 3.05 tonnes and:-
		112	96	80	Length not exceeding 12 metres.
		96	96	80	Length exceeding 12 metres.
Heavy goods vehicles					
Rigid, 2 axles	Car	112	96	80	Up to 7.5 tonnes gross.
Rigid, 2 axles	HGV[2] Class 3	96	80	64	Greater than 7.5 tonnes gross and up to 16.26 tonnes gross.
Rigid, 3 and 4 axles	HGV Class 2	96	80	64	Up to 30.29 tonnes gross.
Articulated, 3 and 4 axles	HGV Class 1	96	80	64	Up to 32.52 tonnes gross.
Articulated, 5 or more axles	HGV Class 1	96	80	64	Up to 38 tonnes gross.

1. Driving licence needed for any bus carrying fare paying passengers.

2. HGV driving licences classes 1A, 2A and 3A are also available restricting driving to vehicles with automatic transmission.

Appendix 1 (b)

COMPARISION OF NATIONAL AND INTERNATIONAL REGULATIONS FOR THE OPERATION OF VEHICLES
FEDERAL REPUBLIC OF GERMANY, StVZO[1] / EEC[2] AND ECE[3]

National regulation for the operation of vehicles StVZO				International regulation corresponding to EEC and ECE				
Motor vehicle category	Driving-licence class	Speed limit[4] [km/h]	Gross vehicle weight rating[5] t	Motor vehicle category	Driving licence class[6]	Maximum weight [t]	Seats	Explanations
Passenger cars	3	100 ( )*	7.5	M1	B	3.5	≤ 8	*On highways (autobahn) no speed limit, but advised speed of 130 km/h
Motor busses[7]								*100 km/h if -the max. speed 100 km/h (it depends on the car licence) -the engine power 11 kW/t of the gross vehicle weight -"100" badge with seal
2 axles	2	80 (80)*	16					
2 axles	2	80 (80)*	22	M2	D	≤ 5	> 8	
2 tandem axles	2	80 (80)*	30	M3	D	> 5	> 8	
Articulated busses	2	80 (80)*	28					
Heavy goods vehicles								
2 axles	3	80 (80)	2.8-7.5*					*Trucks up to 2.8t are treated like passenger cars
2 axles	2	60 (80)	17	N1	B	≤ 3.5	-	
3 axles	2	60 (80)	24					
4 axles	2	60 (80)	32					
Articulated vehicles								**Driving licence "E" is necessary for operation with a trailer
Numbers of the axles: Semi-trailer trucks / Semi-trailers				N2	C**	>3.5≤12	-	
2 / 1	2	60 (80)	27					
2 / 2	2	60 (80)	37					***Gross vehicle weight rating by: first registration after 19.01.'87 :35t from dec.'91 all :35t
2 / 3	2	60 (80)	40					
3 / 2	2	60 (80)	40					
Articulated vehicles with ISO-Container				N3	C**	>12	-	
3 / 2	2	60 (80)	44					
3 / 3	2	60 (80)	44					
Trailers[8]								
1 axle	3	80 (80)	*	01	-	≤0.75	-	*Must not surpass the gross vehicle weight of the pulling motor vehicle
2 axles	2	60 (80)	10	02	-	>0.75≤3.5	-	
2 axles	2	60 (80)	18	03	-	>3.5≤10	-	
3 axles	2	60 (80)	24	04	-	>10	-	

[1] Road Traffic Registration Act
[2] Directions of the European Economic Community for road vehicles (EEC-directions)
[3] Regulations of the Economic Commission for Europe for motor vehicles and their trailers
[4] Out of towns, ( ) on highways (Autobahn)
[5] Separate settlement for frontier traffic in the Saarland (§ 34, StVZO)

single axle	13t
tandem axles, axle base ≥1.35m	21t
motor vehicle or trailer with 2 axles	19t
motor vehicle or trailer with more than 2 axles	26t
motor vehicles with 2 axles and trailer with 2 axles	38t

[6] All data refer to the international driving licence
[7] With passengers without seats, maximum speed: 60 (60)
[8] Driving licence class 2 is necessary for train with more than 3 axles, without regard to the pulling vehicle

Appendix 2 (a)

Improvement of road situation in Europe

Great Britain

Road Lengths in 1000 km.

Year	Total	Motorway	Non-built-up	Built-up[1]
1973	327.1	1.731	198.5	126.9
1974	329.0	1.869	199.0	128.2
1975	330.0	1.975	199.2	128.9
1976	333.4	2.155	200.3	131.0
1977	334.7	2.236	200.1	132.4
1978	336.3	2.394	200.5	133.3
1979	338.0	2.455	201.0	134.5
1980	339.6	2.556	200.9	136.2
1981	341.9	2.628	201.2	138.1
1982	343.6	2.666	201.9	139.0
1983	345.4	2.720	202.1	140.5
1984	347.2	2.802	202.8	141.6
1985	348.3	2.838	202.6	142.9

1. Roads with speed limits of 64 km/h or less.

Appendix 2 (b)

Improvement of Road Situation in Europe

Germany

Year	Road Length in 1000 km			
	Road Category			
	Total	outside build up areas		Urban
		Highway	Rural	
1970	162,3	4,110	158,3	270
1971	164,5	4,461	160,0	276
1972	165,3	4,828	160,5	282
1973	166,7	5,258	161,4	286
1974	167,5	5,481	162,0	290
1975	168,2	5,748	162,4	294
1976	169,1	6,213	163,0	292
1977	169,6	6,435	163,1	299
1978	170,1	6,711	163,3	302
1979	170,7	7,029	163,7	305
1980	171,5	7,292	164,2	308
1981	172,4	7,538	164,9	310
1982	172,5	7,784	164,7	312
1983	173,0	7,919	165,0	314
1984	172,6	8,080	164,6	316
1985	173,0	8,198	164,9	317

Lengths of public roads according to road categories /2/

Appendix 2 (b) continued

Width of the roads	Road category		
	Highway (Autobahn)	Rural	Urban
1.1. 1966			
smaller 4 m	-	11527	97188
4 - 5 m	-	39578	75603
5 - 6 m	-	52449	42678
6 - 7 m	-	30238	19356
7 - 9 m	76	15980	
9 - 12 m	-	2615	15385
12 and more	3296	1798	
total	3372	154160	250219
1.1. 1971			
smaller 4 m	-	7848	102092
4 - 5 m	-	31567	77627
5 - 6 m	-	52072	52888
6 - 7 m	-	38952	25016
7 - 9 m	69	21908	
9 - 12 m	-	3913	18752
12 and more	4392	2991	
total	4461	160008	276375
1.1. 1976			
smaller 4 m	-	5361	102317
4 - 5 m	-	25270	79500
5 - 6 m	-	53197	62585
6 - 7 m	-	44414	30637
7 - 9 m	80	27495	
9 - 12 m	-	5375	21699
12 and more	6127	3823	
total	6213	162933	296732
1.1. 1981			
smaller 4 m	-	4206	
4 - 5 m	-	20222	
5 - 6 m	-	49158	
6 - 7 m	-	48038	
7 - 9 m	136	32191	
9 - 12 m	-	6382	
12 and more	7402	4657	
total	7538	164854	310000

Lengths of public roads according to road widths and categories /2/

Appendix 3 (a)

Heavy Goods Vehicles and Trailers*

Percentage failure by category in Great Britain**:

	1979/80	1983/84	1984/85	1985/86
Body & Chassis	4.51 %	4.89 %	4.63 %	4.49 %
Suspension	8.01 %	8.15 %	7.83 %	8.07 %
Exhaust	2.77 %	2.35 %	2.18 %	2.07 %
Tyres & Wheels	5.4 %	6.39 %	5.92 %	6.75 %
Steering	3.1 %	2.89 %	2.77 %	2.83 %
Brakes	45.97 %	48.15 %	49.14 %	51.26 %
Lights	9.17 %	11.59 %	11.9 %	12.55 %
Fuel & Tank	1.5 %	1.33 %	1.33 %	1.38 %
Tachograph	-	3.5 %	2.71 %	2.49 %
Oil & Waste	1.87 %	1.72 %	1.51 %	1.44 %
Electrical	0.96 %	0.91 %	0.89 %	0.92 %
Others	8.13 %	9.03 %	9.52 %	10.12 %
Total numbers of defects	819221	901642	899960	964380

* Goods vehicles over 1525 kg unladen weight and trailers over 1020 kg unladen weight.

** The table shows the percentage of vehicles failing as a result of the defect indicated. A vehicle may have more than one fault and consequently the totals of the percentages of separate faults exceed the percentage of vehicles failed.

Appendix 3 (b)

Results of general technical inspections for motor vehicles according to type of defect for Germany(3)

Year	Rolling stock ~~Motor vehicles~~ and Trailers in [1000]	Identified Defects [in 1000]						
		Total	Lights	Steering assembly	Brakes	Tires	Chassis and Propulsion	Emission and Sound level
1970	17416	7307	1360	861	1665	503	1009	383
1975	21942	10956	2131	782	2460	536	2178	851
1980	28267	10041	2080	711	1996	423	2286	833
1981	29077	10533	2164	681	2242	422	2568	811
1982	29664	10498	2118	684	2195	448	2599	798
1983	30342	10150	2029	668	2109	452	2649	770
1984	31162	10238	1991	694	2075	463	2774	786
	Cars and Wagons							
1970	13941	5739	1064	716	1358	395	734	328
1975	17898	9256	1805	665	2158	430	1797	798
1980	23192	8490	1770	603	1754	319	1932	783
1981	23730	8989	1852	578	1985	320	2209	762
1982	24105	8923	1795	578	1938	340	2240	745
1983	24580	8644	1720	569	1861	346	2290	721
1984	25218	8682	1670	588	1820	354	2395	732
	Trucks							
1970	1119	750	126	73	158	54	139	32
1975	1245	664	114	42	143	38	153	31
1980	1448	547	101	36	99	26	131	26
1981	1468	540	99	32	106	24	134	25
1982	1478	507	93	30	98	23	125	24
1983	1475	462	86	27	89	22	118	22
1984	1487	445	83	25	86	21	115	22

Priorities in the Active and Passive Safety of Trucks

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Abstract

In a representative study of some 1,200 accidents the accident characteristics of the truck were ascertained and priorities for partner-protection measures were derived from them.

A truck front protection in collisions with cars and side guards for pedestrians and motorcyclists are of paramount importance. The protective effect deduced by the rear underride protection hitherto and the necessity for further improvements is proved from the accident analyses.

So far only a few studies are available on the accident risks for the truck driver himself. In a representative evaluation of accidents the injury patterns of truck drivers dependent on collision type and accident intensity are shown on the basis of 800 truck-collisions. This is followed by a discussion of the injury causes with regard to possible safety improvements. The active safety requirements of trucks resulting from accident studies and the possible effect of technical systems such as antilock braking systems, distance warning devices etc., are discussed. In the studies the distinction was also made between the various truck categories.

The Present Situation

Truck accidents constitute an important element in the general accident scene. Each year in Germany about 30,000 accidents with personal injury involving trucks occur /1/, in which some 1,700 people are killed and 40,000 are injured (Figure 1). In addition, in 1985, about 36,000 truck accidents occurred causing serious material damage. Although in view of their mileage the involvement of trucks cannot be described as overproportionate, more attention in the future must be paid to truck accidents if the absolute accident figures are taken into account, which frequently not only involve extremely serious injuries but also an extraordinarily high level of material damage and a considerable risk to the other traffic in general.

It is chiefly the accident opponent of trucks who run the greatest risk of injury. For about every two car/car-collisions with fatalities there was one truck/car-collision with fatal injuries (Figure 2). Compared with the safety efforts up to now to reduce the risks in car/car-collisions future truck safety research must be considerably intensified in view of this fact. But neither should the efforts to achieve more protection for other road users like motorcyclists, pedestrians, and more safety for the truck occupants themselves be relaxed: even if the risks to the truck occupants are not high in relation to the trucks registered, the absolute injury figures are not insignificant and the potential safety reserves must also be used for the truck occupants.

Accidents / Personal injury resulting	Accidents with fatalities	Accidents with serious injuries	Accidents with slight injuries	Accidents with injuries/fatalities Total
Accidents in Germany 1985	7 678	320 067		327 745
Accidents involving trucks and personal injury*	**1 385**	8246	18 532	28 163
of these injured truck occupants	129	1 803	5 727	7659

*further truck accidents with serious material damage in 1985: 35 974 cases

Figure 1. Truck accidents in Germany in 1985 with resulting injuries to occupants

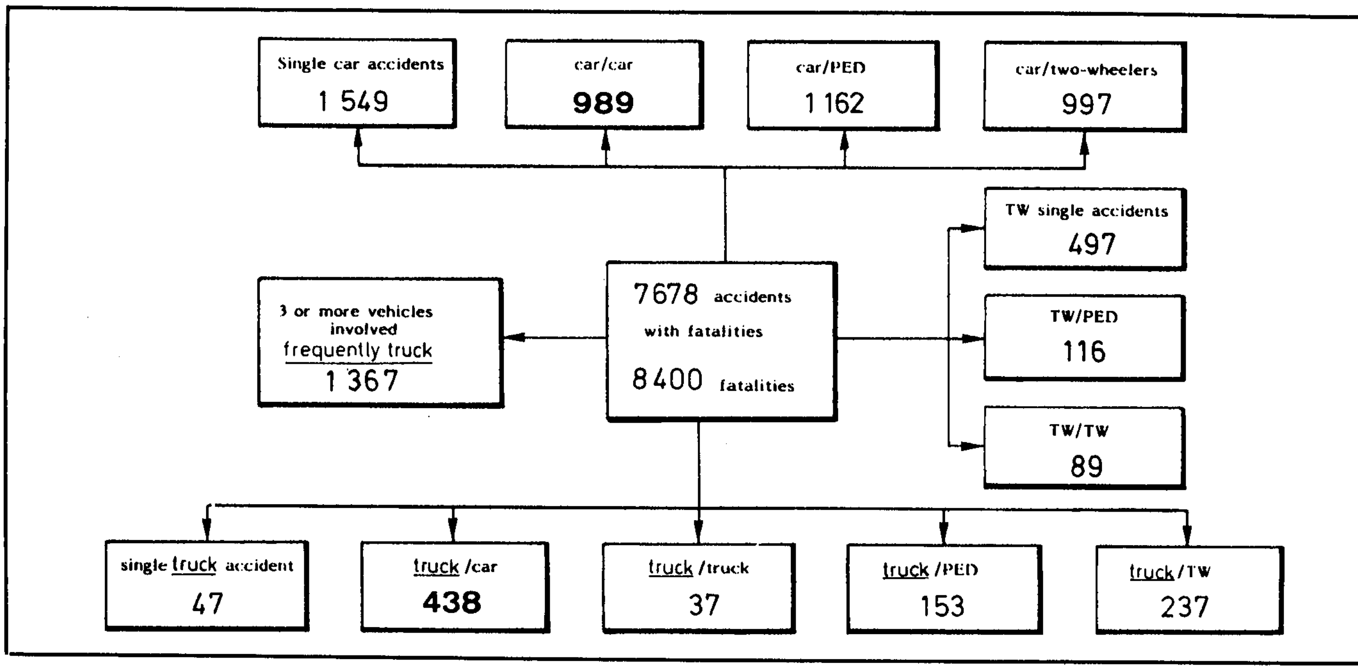


Figure 2. Traffic accidents in Germany in 1985 with fatalities

In view of the existing accident risks, it is astonishing that up to now only relatively few studies have dealt with truck accidents, emphasis usually being placed on the consequences of the accidents and only rarely on the general accident characteristics and the crisis situations in the pre-crash-phase.

It is the aim of this paper to give an overview of the accident involvement of trucks and the findings available by the HUK research on truck accidents with respect to

- partner protection in truck accidents
- self-protection of the truck occupants and
- active safety of trucks

Methodical Aspects of Truck Accident Studies

Building up truck accident material involves extremely great problems, since truck accidents are more difficult to collect by occurrence, call for supraregional evaluation because of their high percentage on highways and since the evaluated accident material has to be divided into a large number of differing truck concepts.

Further problems result from the fact that—unlike in the case of car—or motorcycle-accidents—there is no clear reference basis for criteria of representativity of the material.

The definition of the term "Lastkraftwagen" (truck) as given by the Federal German Statistical Office covers all vehicles which are suitable for transporting goods. It thus necessarily includes a large number of completely different vehicle types—vans and trucks with a tenfold variation of mass and fundamentally differing concept and use.

For this reason only trucks of over 3.5 tonnes maximum laden weight with typical truck construction were considered in this HUK accident material presented; moreover, a detailed classification by mass categories and the type of truck, trailer and articulated lorries was carried out.

Two independent bodies of accident material were available:

- **Truck material I**
  an evaluation of 1,447 truck accidents with cars, cycles, motorcycles and pedestrians with regard to problems of partner protection.
- **Truck material II**
  an analysis of 770 accidents with truck/truck- and truck/car participation and single truck accidents with regard to the problem on self protection for the truck occupants and of active safety. With regard to questions of active safety, 300 of these accidents—a representative sample of truck accidents in the Federal State of Bavaria in 1984—were analysed and examined in-depth.

For these studies which were partly conducted in cooperation with the German "Forschungsgemeinschaft Automobiltechnik" (FAT) and thus with the truck manufacturers well over 10,000 accidents had to be evaluated in order to filter out this accident material.

The accident material now available will be continuously expanded in the next few years to create an enlarged basis for special studies, for example, in the case of truck categories or accidents involving dangerous goods, and to examine the effects of typical safety constructions.

Passive Safety of Trucks—Partner Protection

Car/truck collisions must be given prominence in measures of partner protection (Figure 2), although truck collisions with pedestrians and motorcyclists also represent a high proportion of accidents with fatalities. Although ways of protecting cyclists, motorcyclists and pedestrians in the case of truck collisions are limited, yet they are existent.

Car/truck collisions (945 accidents)

Car/truck accidents account for over 50% of all truck accidents in which people are injured, and comprise about 46% of all fatalities in accidents in which trucks are involved. For the car occupants the frequency of fatal injuries in truck/car accidents is roughly four times higher than in car/car accidents.

Figure 3 gives an overview of the type of collision which occurred, at the same time defining which areas of the car and truck impacted with one another in the main collision phase.

In assessing the ranking order of the types of collision, a distinction has to be made between collision frequency and the severity of the injuries to the car occupants. Car collisions against the sides of the trucks (30.4%) and truck rear-end accidents with cars (26.0%) are the most frequent. But if the resultant injury severity is taken into consideration, it is the front/front collision (collision type A) which is by far the most important and covers about 40% of the cases recorded with serious/fatal injuries from AIS 3 and above. Accidents of cars running into the rear ends of lorries have a share of about 20% of the dangerous/fatal injuries in the accident material. The paramount importance of an optimum rear and front underrun protection of the truck can be derived from this.

The frequency distribution of each of the impact areas on the truck is given in figure 4 for all of the 945 car/truck-collisions recorded. By far the most frequent are collisions against the truck front (60%), which occur namely in the left-hand outside third of the truck front. The lorry is struck on the side in about one quarter of the cases, 40% of these occurring in the area between the axles. The impact areas on the left are far more frequent than on the right.

Rear end collisions in which the back of the truck is struck take place mainly in the left-hand third (55%). The car driver instinctively tries to take evasive action to the left, the main load to the rear underrun protection thereby being not only concentrated on one side but also quite often at an angle as well, which results in the problem of bending away the outer rail of the underrun protection and thus higher strength of the underrun protection device is needed. This risk, which is still quite considerable, is reflected by the fact that accidents in which cars strike the rear of a truck have a frequency of altogether 9.5% only, but comprise 17% of all fatal accidents in the case of car/truck accidents.

There is no doubt that even with the improvement of the reinforcing members of the truck rear underrun protection by the revision of the ECE-R 58 /2/ there is definitely not sufficient protection. This would require above all that the rear underride protection of the truck should be lowered to about 30 cm above the ground and the permissible distance from the rear edge of the loading area should be reduced. Further improvements must—as already mentioned—be made to the reinforcing support members in the event of an angled impact in the outer area and increased test-loads compared with the actual ECE-R 58 are to be required. The relative impact speed of the car on the rear of the truck is in most cases in the region of about 30 to 50 km/h /3,4/.

Type of collision			GREATEST OVERALL INJURY SEVERITY IN CAR			
car/truck			truck accidents with personal injury		MAIS ≥ 3	
			Number	%	Number	%
Front-front	A		184	19.8	83	40.1
Front-side	B		283	30.4	55	26.6
Front-rear	C		88	9.4	33	15.9
side-front	D		134	14.4	32	15.4
Rear-front	E		242	26.0	4	2.0
Total			931	100.0	207	100.0

Figure 3. Type of collision in truck/car accidents, frequency of occurrence and serious injuries resulting

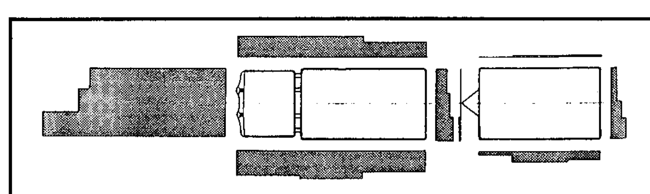


Figure 4. Distribution of the impacted areas of the truck in collisions with passenger cars (945 accidents)

In the case of the dominant front/front collisions, quite naturally serious difficulties are caused by the masses involved and the high relative speed. But studies /3,5/ showed that even serious frontal collisions were mostly not accidents with any offset, in which, of course, the necessarily high truck mass fully loads the opponent's car: heavy car-truck collisions are mostly accidents with comparatively large offset, in which, however, the front design of the truck with high mounted bumpers and rigid structures results in an unfavourable application of force against the car (Figure 5). The truck overruns the deformation structures of the car provided to absorb energy, forces the car under the truck front and wedges it there (Figure 6). This fact in an overproportionate number of cases, results in massive intrusion of the passenger cell with very serious injuries being caused to the car occupants.

Since 1978, the HUK-Verband together with the Technical University in Berlin and truck manufacturers has continually carried out car crash tests into the front of trucks to develop counter measures based on the knowledge of real-life accidents /6/.

These series of tests showed that to achieve the aim of truck front underrun protection, both measures with regard to the truck front geometry and the front deformation characteristics are necessary.

Safety improvements to the truck front design demand first of all

- a lowering of the bumper to a clearance of about 30 cm, so that the car structures are no longer driven over by the truck and the car becomes wedged underneath the truck
- a larger contact area for the car on the truck, so that a wider area of force load is given and a deflection function can take place as far as possible

Simply avoiding the car structures being overrun in itself results in an advantage, but for a truck front protection a further prerequisite is that

- the truck front is designed for a certain energy absorption and not, as is the case today, so that the deformation work has to be performed almost exclusively by the car.

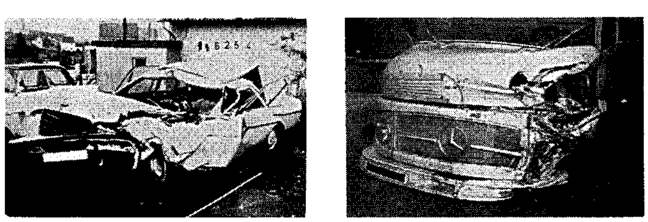


Figure 5. Typical frontal collision of a car into a truck with relatively high offset but serious intrusion of the car

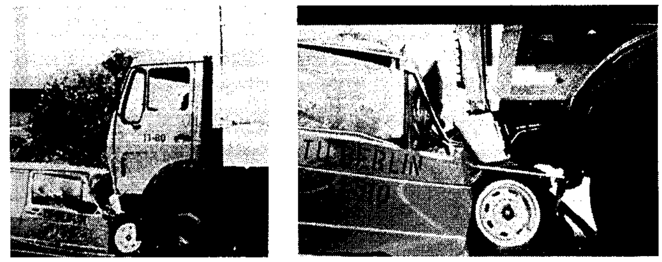


Figure 6. Car to truck crash test: typical force load to the car with the truck's front overrunning the deformation structure of the car

As a fundamental finding of the experiments to date front structures of this kind can be created—although with some expenditure—imposing no limitations on the present truck construction and potential usage and that considerable safety gains are to be expected for the car occupants /7/.

An "energy-absorbing design of the truck front" can be achieved by replacing the conventional bumper systems by a deformation structure which is mounted on a carrier construction (Figure 7a), this carrier system, being, in its turn, attached to the frame and being designed to resist deformation as far as possible.

The experiments have not yet been completed; in the light of these latest developments a front underrun protection of this kind is likely to mean an additional length of the truck amounting to about 10 to 15 cm (Figure 7b/c) and additional weight of some 150 kg—insignificant compared with the total weight of a

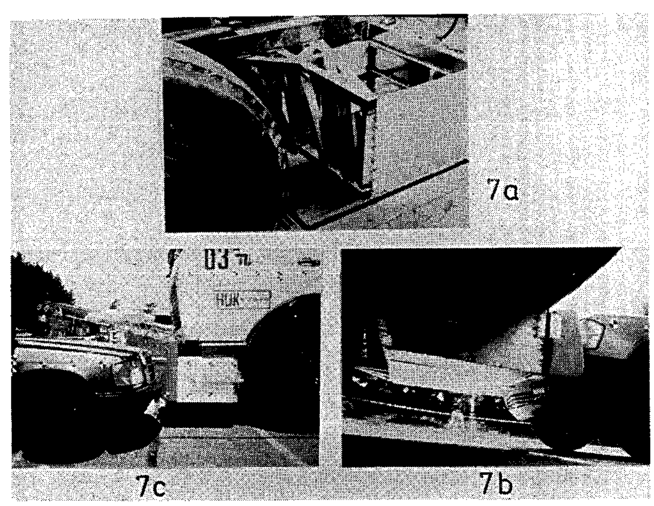


Figure 7. Possible construction of a front underride protection
a) carrier construction and deformation structure
b) truck front underride protection; view from below
c) truck front underride protection; view from the side

truck, but of not inconsiderable significance for economy calculations.

A front protection of this kind results not only in advantages for the car occupants but can also improve the safety of the truck. Even comparatively low relative speeds often cause a massive car impact against the truck's front wheel damaging its steering or even making the lorry unsteerable, so that quite often very serious secondary collisions occur as a result. This risk can also be greatly reduced by a truck front underrun protection in the way suggested.

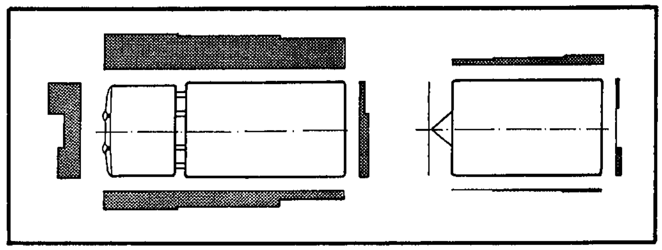


Figure 9. Truck collisions with two-wheeled vehicles; distribution of the impacted areas (390 accidents)

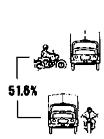
Collisions between trucks and two-wheeled vehicles (390 accidents)

In the case of these collisions with trucks of 3.5 t or more a distinction was made between the categories "motorcycle, light motorised two-wheel-vehicles (Mofas, Mokick) and bicycles".

The individual categories of two-wheeled vehicles show different accident characteristics /3,4/: in one third of the cases involving motorcycles, frontal collisions against the side of the truck take place (Figure 8). When a truck turns off, motorcyclists are apparently frequently overlooked or their speed is wrongly estimated. In comparison with motorcycles, however, "light motorised two-wheeled vehicles" and bicycles are struck far more frequently by the truck front, usually in cross traffic.

The truck's side area is involved in about 50% of the cases in collisions with motorised two-wheel-vehicles- (Figure 9); more than one-third of all collisions take place in the area between the axles (3,4), these accidents accounting for about half of all motorcyclists killed, as well as a considerable proportion of seriously injured people.

Quite a number of these involve glancing off impacts (for example when overtaking) which, accounting for a share of 20 to 33%, represent the second highest risk at all. Decisive here is not the intensity of the impact itself but the danger of a subsequent fall into the space between the front and rear axles with resulting run-over by the truck.



TYPE OF COLLISION: Truck	TYPE OF COLLISION: 2-wheeler	NUMBER OF ACCIDENTS: Motorcycle	NUMBER OF ACCIDENTS: light 2-wheelers	NUMBER OF ACCIDENTS: Bicycles	TOTAL all categories: Number	TOTAL all categories: %	serious injuries of the cyclists (MAIS ≥ 1): Number	serious injuries of the cyclists (MAIS ≥ 1): %
Front	Front	11	12	15	38	9.7	20	13.1
Front	Side	9	19	34	62	15.9	34	22.2
Front	Rear	2	7	15	24	6.1	10	6.5
Side	Front	28	37	34	99	25.4	31	20.2
Rear	Front	10	33	9	52	13.3	20	13.1
Side	Side in the same direction	17	28	57	102	26.2	36	23.6
Side	Side in the opposite direction	5	4	4	13	3.4	2	1.3
all collision types TOTAL		82	140	168	390	100.0	153	100.0

Figure 8. Accident characteristics of collision between trucks and two-wheeled vehicles, frequency of the areas impacted and serious injuries resulting

The danger of running over increases at a lower speed of the two-wheeled vehicle in keeping with the proportion of overtaking manoeuvres. For example of 186 cyclists in the truck accident material with personal injuries /4/, 14 persons were run over.

Figure 9 shows that in truck collisions with two-wheeled-vehicles, the right-hand side area is the critical impact area—a side underrun protection on the truck would influence around 50% of the serious-/fatal injuries to drivers of two-wheeled vehicles and—in particular—would totally avoid the dangerous falls between the front and rear axles. Any improvement of the present situation including refitting trucks and trailors with side protection results in advantages; an optimum side underrun protection, however, should be designed with a flat surface covering the whole space of the side especially on the right-hand side of the vehicle. Thus the fact is also taken into consideration that especially motorcycles collide relatively frequently head on with the truck side area and that therefore the driver incurs the risk of becoming entangled by the truck.

Collisions of trucks and pedestrians (112 accidents)

Here, of course, the most serious accidents are encountered: only one-third of all pedestrians get off with only moderate injuries in truck/pedestrian collisions /3,4/.

The impact of the pedestrian most frequently takes place (42%) with the front of the truck in the right hand area; in over one third of the cases an impact against the right hand side area occurs and protective measures must first of all be taken in these areas (Figure 10).

What is decisive is a closed surface and avoiding protruding edges in the vehicle's front and side especially between the truck driver's cabin and the loading area. Also by mounting a truck front protection as proposed at a short distance from the ground certain safety improvements, for example, avoiding being driven over, can be carried out.

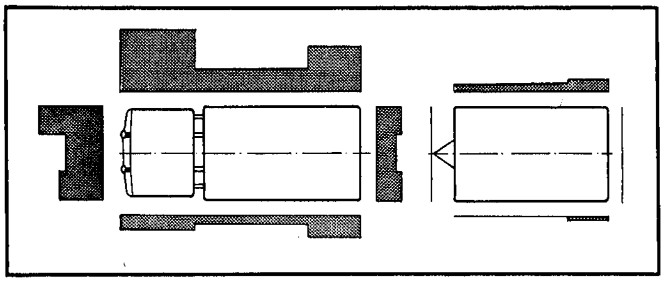


Figure 10. Impacted areas in collisions of trucks with pedestrians (112 accidents)

About two-thirds (65.9%) of truck/pedestrian accidents take place in a collision speed range of up to 30 km/h /3,4/. Because of this relatively low speed the search for safety measures, also with regard to pedestrian protection, do not seem to be completely hopeless.

The collision risk for the pedestrian could also be reduced by "active safety": since the pedestrian is frequently struck only by the outer parts of the truck /4,8/, equipping the truck with anti-locking brakes (ALB) a reduction could be expected because of the possibility of steering the truck in spite of an emergency braking.

Safety measures for the truck occupants (sample 770 accidents)

While in the last few years an increasing number of studies has been made on the subject of "partner protection" few studies are available today on the risks to the truck occupants. For this reason new accident material from 770 collisions involving trucks of 3.5 t or more was evaluated, the studies covering the whole of the Federal State of Bavaria, which means they can also be regarded as representative of the accident situation in Germany. Altogether 3,500 truck accidents from 1984 were evaluated, and from these the above-mentioned accident material II was selected according to the following criteria:

To describe the current state of truck safety, only accidents involving a truck of 3.5 t or more, manufactured in or after 1976 were included. In addition to the injuries to the occupants in the truck accidents with safety related damage to the driver's cabin but without injury to the occupants were also included in order to avoid a negative selection and thus to objectively reflect the safety risks to the truck occupants.

The study showed that the injury risk of the truck occupants is almost exclusively dominated by truck/truck collisions and single-truck-accidents (Figure 11). Car/truck collisions are, of course, frequent as far as the number of cases is concerned and can also result in considerable front damage to the truck, but the resulting injury risk of the truck occupants is nevertheless slight and the serious injury consequences are mainly due to secondary collisions.

Surprisingly not the lighter mass categories of up to 12 t were found to be overrepresented in accidents with injury to truck occupants in the accident material, but mainly the heavy trucks of 16 t or more, the truck trailers or the articulated lorries as compared with their proportion of the registrations. Thus, the

Injury severity / Accident group	uninjured	slightly injured	seriously injured		Fatalities		TOTAL truck occupants	
	Number	Number	Number	%	Number	%	Number	%
Single truck accident	19	115	43	**33.6**	4	**30.8**	181	18.8
Truck/truck	80	136	57	**44.5**	6	**46.1**	279	29.1
Truck/car	325	107	20	15.6	3	23.1	455	47.3
Truck/other	26	12	8	6.3	-	-	46	4.8
TOTAL	450	370	128	100.0	13	100.0	961	100.0

Figure 11. Truck accidents with injury or serious safety risk to truck occupants, resultant risk related to type of accident

"weight bonus" of heavy trucks involved in collisions with other accident opponents seems to represent an additional burden in truck/truck collisions and single truck accidents /9/.

Two out of three truck/truck collisions with injury to the occupants are frontal collisions with the rear-end of another truck, which in 66% of the cases occurred on motorways, and most of which involved long distance trucks (Figure 12). In 72.7% of these front/rear-end collisions the relative speed was under 30 km/h, and in 96% under 50 km/h.

Figure 13 shows the damage characteristics in truck/truck collisions in general. The intrusion in rear-end impacts of trucks—being the most predominant risk for occupants—is shown in Figure 14. Even if the driver often tries to move over towards the left as far as possible at the last moment, the outside left-hand third of the front is not less frequently loaded with direct deformation. The central problem for the stability of the driver's cabin and thus for the injury risk to the occupants is connected with an impact against the protruding, rigid and not compatible rear-end of another truck /4/.

The dangerous intrusions of 60 cm and more were not to be found in cases with partial force load, but in full force load to the whole cabin without offset (Figure 14). The main deformation here is not in the roof, but in the lower area between the bumper and lower frame of the windshield. Even if this frequently results in massive intrusion, normally a "minimal survival space" remains for the truck occupants, even in extremely serious accidents.

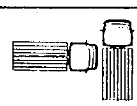
The second greatest risk of injuries to the truck occupants is encountered in single vehicle accidents, 50% of which are characterized by the vehicle leaving the road and then driving into a ditch or running into the embankment without a massive collision with an object. Only in about 25% of the cases was there a massive collision with a tree, a wall or some other object. In 71% of these cases, however, the speed was over 45 km/h and often in the critical range.

The damage characteristics in single vehicle accidents compared with truck/truck collisions show clearly that the impact is in the right-hand area in 40% of the cases, the intrusion of the driver's cabin being generally less often severe (Figure 15) in comparison with front/rear end truck collisions.

Special risks are involved when the truck tips or turns over, and this alone accounted for 50% of the fatalities in the truck and about 25% of the injuries ranging from AIS 3 to AIS 5; these cases occurred mainly in single vehicle accidents.

In 147 of the 770 cases (19%) of this sample the truck turned over with the vehicle tipping over in 134 cases (usually turning over onto the right side) and rolling over completely in only 14 cases. When it tips over to one side the intrusion into the driver's cabin is usually moderate, but the danger of ejection of the unrestrained occupant or the risk of interior impact by being thrown to the side is very great.

In this connection no major problems were observed with the rigidity of the roof—although there was considerable local intrusion in some cases, the minimum survival space was maintained even then.

Type of collision (A ↓ B ↓)		MAIS 2-5		MAIS 6		all injured truck occupants Total	
		Number	%	Number	%	Number	%
Front/front		3	7.0	1	(16.7)	24	11.3
Front/side		7	16.3	2	(33.3)	27	12.7
Front/rear		32	74.4	3	(50.0)	132	61.9
Other types of collisions		1	2.3	-	-	30	14.1
Total		43	100.0	6	100.0	213	100.0

A ≙ observed truck   B ≙ opponent

Figure 12. Highest degree of injury severity of truck occupants involved in truck-to-truck collisions—related to type of impact

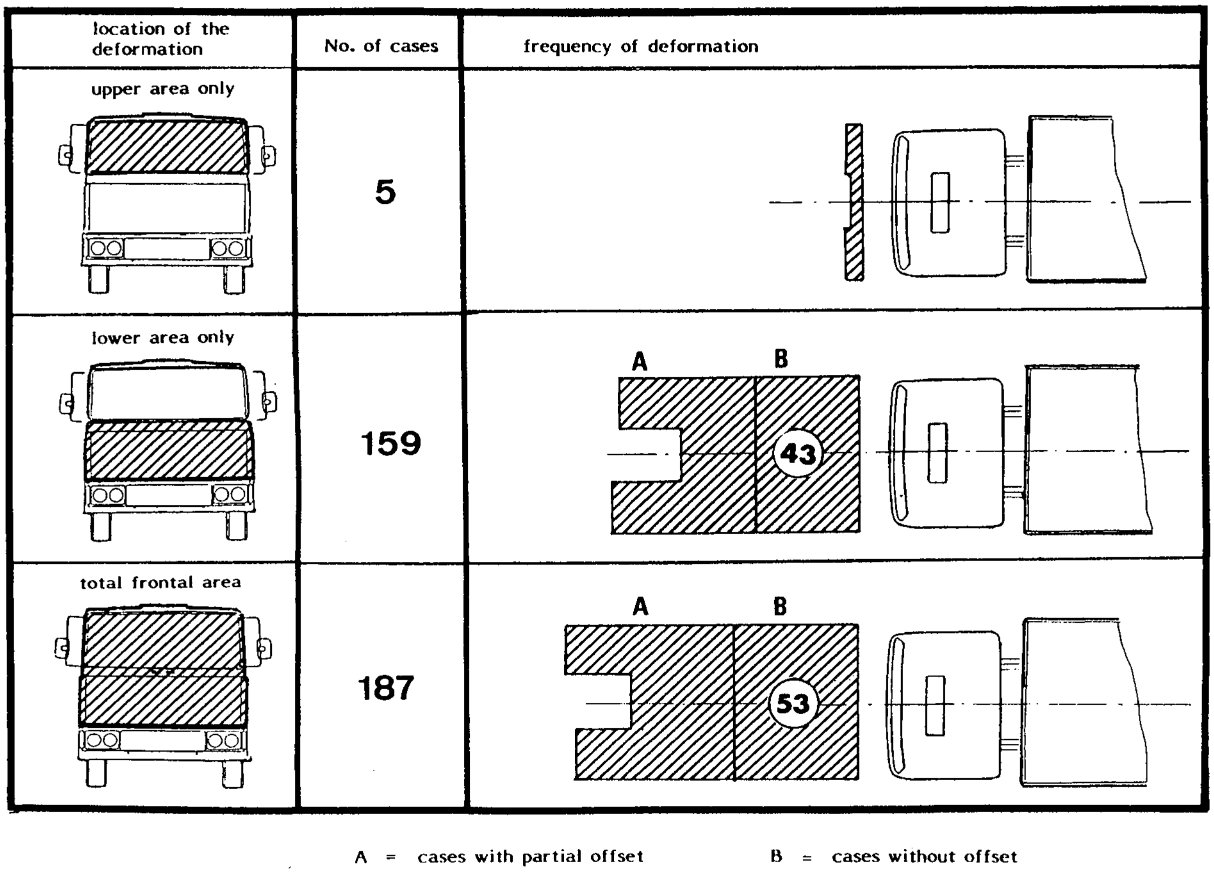


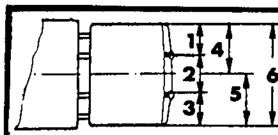
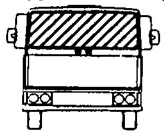
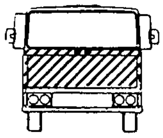
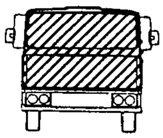
Figure 13. Frequency of damaged areas of the truck's front in truck-to-truck collisions generally

Figure 16 gives the total injury severity of the truck occupants involved and the frequency/severity of the injuries to the various parts of the body.

As many as half of all severe/fatal injuries to the truck occupants in this study are due to ejection. In 8% of the 770 cases intrusion was so great that the occupants were wedged in the vehicle (usually in the region of the legs). The other injuries occurred mainly through impact in the interior, the "steering wheel/ steering column" and deformation of the "foot/ leg-space" dominating as the cause of serious injury. The impact to the windshield is frequent but mostly without serious consequences. Occupant impact to the truck's side areas and in the roof account for a comparatively low proportion /9/.

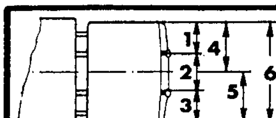
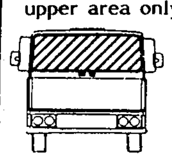
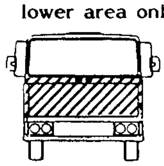
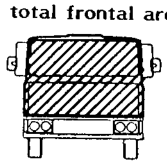
The injuries to the individual parts of the body clearly confirm the injury risks typical of trucks. Head injuries are bound to dominate in fatal injuries, but abdominal and chest injuries are almost of equal significance. Among the serious/critical injuries of AIS 3-5 injuries in the abdominal, chest and leg areas occur far more frequently than head injuries—an indication of the problems connected with occupants being wedged in and the injury causes connected with the steering wheel/steering column. In the case of intrusion the unfavourable impact conditions of the steering wheel—flat normal position—thus locally concentrated impact loading and only limited yielding— are, in addition, further aggravated by the fact that the steering wheel/steering column are frequently pushed upwards (Figure 17).

Assessments were made to find out which safety measures are suitable for reducing the comparatively low, but existing injury risks of the truck occupants. The best protective effect could be expected from a safety belt, which in approximately 50% of the cases would result in a considerable reduction of injuries. In particular the fatal injuries (ejection) and the serious injuries in the abdominal and chest region as well as the thigh fractures as a result of the occupants being thrown forward would thus be considerably reduced. Problems which were at first expected, for example caused by excessive intrusion and lack of survival space, were—except in two cases—not observed in this study and are below 1%. But it must be pointed out when discussing the question of safety belts that belt characteristics (restraint effect as with a 3-point-safety belt) and function (taking into account the truck's vibration) must not restrict the ergonometry.

 location of the damage		Damaged area: front thirds only 1	2	3	combined area 4	5	no offset 6	Total
upper area only 	INTRUSION (cm) - 20							
	20 - 40		no isolated local intrusion					
	40 - 60		only in combination with lower area					
	> 60							
lower area only 	- 20	-	-	2	2	2	16	22
	20 - 40	-	-	-	1	1	2	4
	40 - 60	-	-	-	1	-	-	1
	> 60	-	-	-	-	-	-	-
total frontal area 	- 20	-	-	-	1	1	6	8
	20 - 40	-	-	2	1	1	15	19
	40 - 60	-	-	-	-	2	22	24
	> 60	-	-	-	-	2	27	29
Total		-	-	4	6	9	88	107

Damaged area considered

Figure 14. Range of intrusion to the different areas of the truck front in truck-to-truck rear end collisions

 location of the damage		Damaged area: front third only 1	2	3	combined area 4	5	no offset 6	Total
upper area only 	INTRUSION (cm) - 20	2	-	-	-	-	-	2
	20 - 40	-	-	-	-	-	-	-
	40 - 60	-	-	-	-	-	-	-
	> 60	-	-	-	-	-	-	-
lower area only 	- 20	2	-	-	-	-	2	4
	20 - 40	-	-	-	-	1	2	3
	40 - 60	-	-	-	-	-	-	-
	> 60	-	-	-	-	-	-	-
total frontal area 	- 20	-	1	1	1	1	7	11
	20 - 40	-	-	-	-	1	3	4
	40 - 60	-	-	-	-	-	6	6
	> 60	-	-	-	-	1	4	5
Total		4	1	1	1	4	24	35

Damaged area considered

Figure 15. Range of intrusion to the different areas of the truck front in single vehicle accidents

PARTS OF THE BODY INJURED	Total Number	Total %	AIS 1-2 Number	AIS 1-2 %	AIS 3-5 Number	AIS 3-5 %	AIS 6 Number	AIS 6 %
Head	194	37.9	183	39.6	4	10.8	7	53.8
Neck/cervical spine	39	7.6	36	7.8	2	5.4	1	7.7
UPPER EXTREMITIES								
- Shoulder	33	6.4	33	7.1	-	-	-	-
- Chest	61	11.9	45	9.7	12	32.4	4	30.8
- Upper arm	26	5.1	25	5.4	1	2.7	-	-
- Lower arm	47	9.2	47	10.1	-	-	-	-
- Hand	62	12.1	62	13.4	-	-	-	-
- Back and lumbar vertebrae	34	6.6	33	7.1	1	2.7	-	-
LOWER EXTREMITIES								
- Abdomen	43	8.4	29	6.3	8	21.6	6	46.1
- Thigh	36	7.0	24	5.2	10	27.0	2	15.4
- Knee	75	14.6	69	14.9	6	16.2	-	-
- Lower leg	79	15.4	69	14.9	10	27.0	-	-
- Feet	38	7.4	37	8.0	1	2.7	-	-
Massive accident shock	32	6.2	32	6.9	-	-	-	-
Multiple bruises grazing	90	17.6	90	19.5	-	-	-	-
Total: injured truck occupants	512	100.0	462	100.0	37	100.0	13	100.0
Persons with injury AIS 1-6	512							
Persons uninjured	449							

Figure 16. Frequency and severity of injuries to the different parts of the body of truck occupants

Active safety—avoiding truck accidents (sample 300 accidents)

A representative sample of 300 accidents were selected from accident material II in which safety risks for truck drivers, including car/truck accidents were analysed. Collisions with two-wheeled vehicles/pedestrians were deliberately not included here, since these accidents reveal completely different characteristics with different safety requirements (for example, visibility) and will have to be treated in a separate study.

The criteria for inclusion in the present study material are given once more:

- trucks of over 3.5 t maximum laden weight manufactured in or after 1976
- injury to the truck occupants and/or damage to the truck driver's cabin which is of significance for safety.

The high proportion of truck-trailer combinations and articulated trucks, altogether 55.0%, (figure 18) is striking in this accident material. Even if the selection criteria had a certain influence, this does show the great significance of truck-trailer combinations and articulated trucks with respect to active safety and this confirms that in the case of these truck categories all possible technical efforts will have to be made to reduce the accident risk.

An accurate assessment of questions of active safety presupposes detailed knowledge of the individual pre-crash situations and the typical behaviour pattern—data which has up to now only been available to a limited extent and only rarely for typical truck categories.

Experience to date shows that truck accidents can be classified into a few main types which recur with similar crisis situations.

With regard to the recording criterion the following frequency of dominating "crisis situations" was to be found (Figure 19):

- rear-end accidents on Autobahns (24.3%)
- single truck accidents (20.3%)
- accidents in the area of crossings (21.3%)

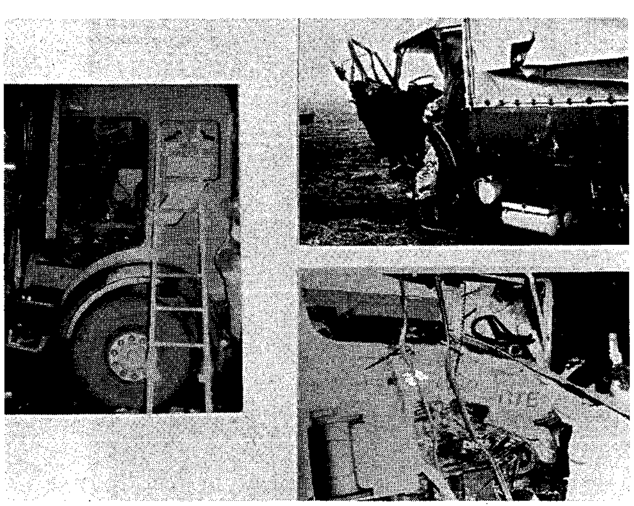


Figure 17. Examples of typical intrusion of the driver's cabin, illustrating the injury risk from the steering assembly and in the leg area

Truck type (total laden weight)	Number	%	
Small truck ÷ 7.5 t	97	24.9	
Simple truck 7.5 - 12 t	15	3.9	
Simple truck >12 t	63	16.2	
Truck with trailer ÷ 28 t	31	8.0	
Truck with trailer >28 ÷ 38 t	95	24.4	55.0 %
Articulated lorries	88	22.6	
Total of trucks involved	389	100.0	

Figure 18. Representative sample of 300 truck accidents (≧3,5 tonnes); selection criteria: cases with injury to truck occupants or safety-related damage to the driver's cabin mass category of the 397 trucks involved

		Total: 300 accidents Number	%
Rear-end collisions Autobahn		73	24.3
Rear-end collisions other roads		26	8.7
on-coming traffic accident, straight		32	10.7
on-coming traffic accident, bend		44	14.7
Collision at crossing/junction		64	21.3
Single truck accident		61	20.3

Figure 19. Frequency of critical pre-crash situations

It is typical of the pre-crash situations termed "oncoming traffic accidents straight" (14.7%) and "oncoming traffic accidents bend" (10.7%) that most of them are not characterised by wrong behaviour on the part of the truck. In three out of four cases the accident opponent—mostly a car— crossed onto the truck's side of the road through skidding, overtaking or excessive speed, unfavourable road conditions, wetness or snow, being present in about 50 to 75% of the cases.

Further details of the "crisis situation" result from the breakdown by vehicle categories (Figure 20). By far the largest proportion of the truck-trailer combinations and articulated trucks in the accident material were involved in Autobahn accidents and comparatively rarely in accidents at crossings. However, trucks in the mass category of up to 7.5 t were quite frequently involved in accidents at crossings, which is probably due not only to the use to which they are put but also frequently to the way they are driven and problems of their frequently changing loads.

"Rear-end accidents Autobahn" are in turn marked by three typical causes (if fatigue of the driver is not included as a further cause):

- The driver drives up close to a lorry ahead of him at a far higher speed, but cannot swerve out to overtake, as the lane for overtaking is occupied.
- The driver recognizes too late to avoid the lorry ahead of him, that he has wrongly estimated the speed of this lorry and his truck (trailer) gets out of control when the necessary emergency braking is made.
- Putting his trust in a clear road, the driver drives too fast and is taken by surprise by unclear situations, a traffic hold-up or fog—a typical problem in the case of mass accidents.

These rear-end accidents frequently occur in the dark, so that estimating speeds is much more difficult.

The study showed that by improving the form of rearward signals, indicating the unusually slow speed of a truck on Autobahns (caused by the load, a defect or something similar) could help to achieve a better estimation of the speed or a higher level of attention particularly at night or when visibility is low. It should be proved, if for example, by means of additional lamps, which only switch on outside of urban areas and would signalise automatically an unusually slow truck speed, a considerable number of

		Vehicle A						Vehicle B							
		Truck			Articulated lorries	Trucks with trailer		Truck			Articulated lorries	Trucks with trailer		pass. car	Bus
		÷7.5t	÷12t	>12t	38t	÷28t	28-38t	÷7.5t	÷12t	>12t	38t	÷28t	28-38t		
Rear-end collisions Autobahn		7	3	5	21	3	34	2	1	3	21	11	19	11	1
Rear-end collisions other roads		10	-	5	6	-	5	5	-	4	2	1	2	8	2
on-coming traffic accident, straight		7	1	7	8	4	5	1	-	1	-	1	2	27	-
on-coming traffic accident, bend		14	4	8	5	3	9	2	1	4	1	-	1	33	2
Collision at crossing/junction		30	1	12	8	3	9	4	1		1	2	1	54	1
Single truck accident		15	3	10	16	6	11								

Figure 20. Pre-crash situations and involvement of trucks by type and mass category

rear-end accidents caused by false estimates of slow trucks could be avoided.

The dominant occurrence of rear-end accidents on Autobahns also led to an analysis of the possible effect of a "distance warning device" for trucks. As was expected, it was shown that they would be of great benefit: 31% of the truck accidents on the Autobahn would certainly have been avoidable and a further 22% might have been avoided or at least the resulting damage could have been reduced. On country roads this figure is reduced, although it is still as high as 12.4%.

It would be a pre-requisite, however, that the truck driver observes this information. The problems with overtaking, cutting by other vehicles, etc., are obvious— but if the development of a feasible distance warning device for heavy trucks, could be successful, a considerable proportion of the most serious accidents today could be avoided.

In further studies the braking behaviour depending on the crisis situation and the possible benefit of an antilocking brake system (ALB) were examined (Figure 21). The study showed that in 52.6% of the cases an emergency braking is made by the truck driver and that—especially in accidents at crossings—there is, even today, an attempt to react by braking *and* steering. But as the wheels are locked this is not only unsuccessful but also involves the danger of secondary collisions.

In a detailed case study the accidents were evaluated with regard to the possible benefit of an anti-locking brake system (ALB) using the definition: it was assumed that the accident would certainly have been avoided if the driver tried to react by steering and braking which would have resulted in avoiding the accident but which was not possible because the wheels locked. It was assumed that the accident would probably have been avoided as the accident situation had provided a clear possibility of doing so if the driver had used a steering reaction in spite of the emergency braking. The effect of reducing the accident consequences was only taken into this category if

Accident types	No emergency braking			with emergency braking		
	Total	of these steering yes	no	Total	of these steering yes	no
Rear-end collisions, Autobahn	36	9	27	36	4	32
Rear-end collisions other roads	6	1	5	20	2	18
on-coming traffic accident straight	11	4	7	18	8	10
on-coming traffic accident bend	19	8	11	22	9	13
Collision at crossing/junction	24	3	21	40	15	25
Single truck accident	41	12	29	16	5	11
in all crisis situations	137	37	100	152	43	109
		12.8%	34.6%		14.9%	37.7%

Figure 21. Emergency reaction of truck drivers in the critical pre-crash phase

a very substantial reduction of accident intensity could have been expected with an anti-locking brake system.

It was shown that high effectivity would exist for Autobahn accidents of trucks by using an anti-locking braking system about 7% of the truck accidents would certainly have been avoided and a further 19% probably. This high rate of avoidability is above all influenced by the proportion of accidents involving truck-trailer combinations, in which the trailer frequently gets out of control if an emergency braking is made.

In the case of truck accidents on country roads the figures were lower because of the different traffic characteristics, but even so 3.1% of these accidents could certainly and an additional 10.8% could probably have been avoided. If it is considered that in about half of the accidents the crisis situation occurs so quickly and unexpectedly that an effective braking reaction cannot take place the anti-locking brake system offers a high degree of effectiveness.

These findings confirm independent studies into accidents of trucks with dangerous loads /10/ and bus collisions /11/ in which it is estimated that the use of an anti-locking brake system would certainly avoid 5% and probably 15% of the accidents.

From a technical point of view fitting heavy trucks (12 t and up) with anti-locking brakes as standard equipment—especially, of course, articulated vehicles and truck-trailer combinations—is the principle measure for reducing the existing braking problems. With regard to the accident risk, however, in a second phase the smaller trucks, too, should be fitted with anti-locking brakes as standard equipment in order to avoid or reduce accidents especially in the crisis situations in crossings—frequently accompanied by extremely serious injuries to the accident opponents.

The intention of the West German Government to make anti-locking brake systems compulsory standard equipment for trucks in the future is a decisive step towards more safety for trucks.

In further studies based on the present accident material the peripheral conditions for the demands made on the anti-locking brake systems will be analysed. A warning, however, must be made against exaggerated hopes, since an accurate recording of the different parameters, such as loading conditions, changes in friction values in a longitudinal or transverse direction is very difficult and is often even impossible. Analysis of braking marks and the pre-crash frequency of trucks leaving the road and driving on to the shoulder lead us to expect, however, that in 10% of the accidents μ—Split-conditions probably existed. For this reason, if trucks are to be fitted with anti-locking brakes as standard equipment, an optimum system of Category I in accordance with the ECE regulation R 58 /2/ should be aimed at.

Summary

Truck accidents represent a significant factor in the overall accident scene: every sixth fatal accident involves a truck; each year in Germany about 1,700 people are fatally injured in truck accidents and 40,000 are slightly/seriously injured.

With regard to passive safety (partner protection) the measures termed "truck front underrun protection" in car/truck accidents and mounting a "side underrun protection" in collisions with two-wheeled vehicles and pedestrians are of paramount importance, but the safety criteria for truck occupants should not be ignored either.

In *car/truck collisions* the problem zones are to be found in the area of the truck front, in the outer left-hand third of the truck's rear end and on the truck side between the axles.

About 60% of the serious car/truck collisions involve the truck front. By creating a truck front underrun protection system instead of the rigid bumpers, usually mounted at a high level nowadays, a considerable reduction of the risks to the car occupants can be expected.

The development work to date shows that—although with some expenditure (costs, 150 kg weight, about 10 to 15 cm additional length)—effective solutions are possible by mounting energy-absorbing structural elements onto the carrier structure of a truck front protection as proposed.

A front protection of this kind would have the advantage for the truck itself that in the event of accidents of considerable severity no direct impact would occur against the front wheel and this would avoid problems of damaging the steering and/or the steerability (danger of secondary collisions).

By being mounted at a low level, the front protection could act positively by increasing a deflection effect in collisions with two-wheeled vehicles and pedestrians who are quite often today knocked over and driven over; moreover some elastic absorption area could be attached to this deformation structure, and this might cushion at least to a limited degree the main points of impact.

The principal requirement for passive safety in collisions with *two-wheeled vehicles and pedestrians*, however, is to mount a side underrun protection and to consistently make the truck less aggressive in the region of the right-hand front and side area. The side underrun protection should be designed with a plain surface in the total area between the front and rear axles, particularly since it should be effective not only in the case of glance-off collisions, but also in the case of relatively frequent impacts at an angle by motorcyclists in this area when the truck turns off.

The rear underrun protection device in its present design in accordance with ECE regulation R 58 undoubtedly represents an improvement, although there are still problems in the case of offset impacts at an angle onto the outer areas with the risk of the outer part being bent off. An improvement in the supporting reinforcing members and increased test loads in ECE-R 58 (150 KN instead of 100 KN) would be desirable; the rear underrun protection should have a clearance from the ground of no more than 30 cm and be mounted as near as possible to the rear end.

With regard to the *passive safety of truck occupants* (self-protection) measures are possible and necessary in respect of a truck structure, the design of the interior and the use of restraint systems.

The main problems connected with intrusion do not occur in truck collisions with objects but in collisions with other trucks. The decisive intrusion zone is therefore in the region between the bumper and the lower windscreen frame—typical of the collision with the rear of a lorry. With regard to roof rigidity the present study does not reveal essential safety defects; in spite of, in some cases, massive intrusion, a necessary minimal survival space remained.

Safety tests of the structural rigidity of the driver's cabin should therefore be carried out against structures similar to the rear structures of trucks /12/. Isolated loading of the roof does not correspond to real-life risk situations. Turning over onto the side or even complete rollover, only rarely result in massive deformation of the driver's cabin and serious restrictions of the interior—the central problem is, rather, in the intrusion of the front area between the bumper and lower frame of the windscreen.

In the truck's interior, energy-absorbing padding should be improved as it provides better protection against leg injuries to the driver, and avoids, as much as possible, his being wedged in in the event of intrusion of the front structure of the driver's cabin.

The flat position of steering wheels nowadays represents a considerable injury risk to the chest/abdominal region for the unrestrained truck driver. It should be proved whether truck steering wheels can be better designed from the biomechanical point of view— at least design measures must be taken to avoid the dangerous pushing upwards of the steering assembly in the event of intrusion of the front structure.

The risks for truck drivers which dominate today as a result of ejection and impact in the interior, especially at the steering column/steering wheel, would be considerably reduced by wearing a safety belt; with safety belts—adapted, of course, to the specific requirement of trucks—a probable benefit could be expected in about 50% of the accidents recorded here, reducing in particular serious/fatal injuries to a greater extent. A negative effect, if the occupants are restrained in their seats, in the event of

intrusion did not prove to be relevant in this study and seems not to be different to the existing situation in passenger cars.

In the field of *active safety* the standard equipment of trucks, especially trucks of 12 tonnes and up, truck-trailer combinations and articulated lorries with anti-locking brakes constitutes the central safety measure of the next years. It is expected that the possibility of being able to steer and brake in emergency situations would certainly avoid 5% and probably 15% of the truck accidents and considerably reduce their consequences,—especially since these accidents frequently occur at relatively low collision speeds, but nevertheless result in extreme damage by reason of the great masses involved. But in a second, later equipment phase the lighter lorries (starting from 3.5 tonnes and up), too, should also be equipped with anti-locking brakes, since especially lighter trucks are involved in accidents with considerable injury to people as a result of their being used in built-up areas.

The great significance of accidents on Autobahns for heavy trucks was confirmed, rear-end collisions dominating. As an additional preventive measure to avoid accidents, an improved rear signalling system for trucks (warning for vehicles behind when trucks are driving unusually slowly on Autobahns) should also be examined as well as the use of distance-warning devices, especially on Autobahns. There are certainly considerable problems in the practical operation with these two possible measures; by reason of the serious risks in these frequent critical situations, however, solutions must be achieved.

The catalogue of measures described for active and passive safety must, of course, remain incomplete. But it has been possible to acquire a large body of representative material on truck accidents, and in further studies possible individual measures will be examined. The practical applications of the measures mentioned would, however, already result in a considerable improvement in active and passive safety in accidents involving trucks. In this connection internationally uniform safety regulations should be available for trucks in order to achieve the same safety standard and uniform conditions of competition in the international traffic of today which will increase in future. But these regulations must guarantee a high level of safety and must not represent a minimum common denominator.

Even with an increase in the transport requirements of the truck, economy and safety are not contradictory demands. Studies of truck accidents must be continued and intensified to present criteria for deciding on future technical developments and possibilities of further advanced truck safety.

Concluding Remarks

The authors are indebted to all the member insurance companies of the HUK-Verband in particular for making their claims records readily available for this study, and to Messrs. Heider and Zistler, members of the HUK staff who carried out accident analysis and evaluation work. The authors also owe a debt of gratitude to the various police departments.

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Typology of Traffic Accidents Concerning Cars Impacted by Trucks

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Abstract

The road traffic mixes in the same flow—light cars and heavy trucks: well then a large part of severe traffic accidents concerns trucks impacting light vehicles.

Actually, if this type of accident does not occur frequently (4.5 percent of all accidents), it is very serious and 10 percent of traffic deaths are due to it. Using the sample of our bidisciplinary accident investigation, this study, concerning 53 cases of actual accidents, tries to precise the typology of truck-to-car accidents and the influence of the type of impact on car occupants lesions. Then it determines the more representative impacts conditions.

Introduction

This study is based on 53 cases of actual accidents in which a passenger car was impacted by a truck. These data were collected over a period of five years from 1982 to 1986, by the bidisciplinary accident investigation team of the L.C.B.(Inrets France).

We can explain the limited number of cases by the fact that, on the one hand, the bidisciplinary study collects all types of traffic accidents and, on the other hand, it is often impossible to trace the truck, which leaves the scene of the accident quickly if it is only slightly damaged.

The aim of this study is to establish the origins of crash severity, for this type of traffic accident.

We consider as truck, any heavy good vehicle of which the full weight is higher than 3,500 kg and less than 38,000 kg. Among these 53 heavy trucks, we have only four coachs, the 49 others are 27 articulated and 22 non-articulated trucks. In the 53 passenger cars, we have 86 passengers involved, 53 drivers and 33 passengers.

Some Considerations Concerning Characteristics of Trucks Involved

Truck Mass and Passenger Cars Mass

Figure 1 introduces the actual truck mass at the accident time, i-e, the truck weight plus load weight.

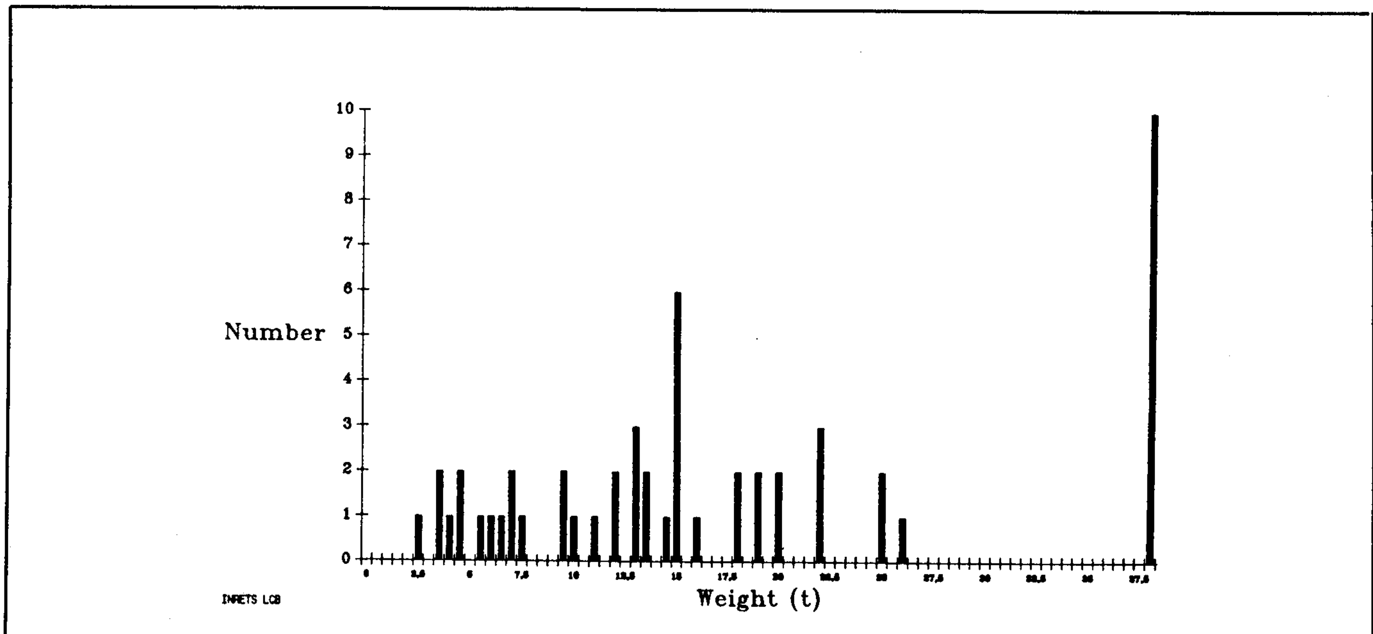

Figure 1. Weight of trucks

We can see in our sample an approximately regular distribution between 3,500 kg to 12,000 kg, concerning one-third of this sample. Then, we notice a peak of 12,000 kg and 16,000 kg truck weight, then a uniform distribution between 18,000 kg and 26,000 truck weight. Finally, we notice the highest peak (38,000 kg) usually for tractor semi-trailers.

We observe that we have no trucks between 26,000 kg and 37,000 kg.

The articulated trucks (tractor and semi-trailer) (38,000 kg) represent about 20 percent of our sample.

The absence of trucks between 26,000 kg and 87,000 kg can also be explained since this type of truck is used essentially on building sites.

When we consider the weight plus load of passenger cars (fig. 2), we notice that an important part of them are medium sized, indeed even small sized.

This distribution is, in fact, very representative of French number of cars on the roads.

Distance from Ground to Bumper for Heavy Trucks

This parameter is very important because it determines the underruning of passengers cars under the truck.

In fact, one of the most important factors is that, when a truck collides with a passenger car, it generally hits the car above the bumper i-e, the deformable part of the front end, or even the engine part.

We have measured the distance ground/truck bumper for our truck sample. These measurements were carried out on fully loaded trucks. We established two categories: articulated and non-articulated trucks as can be seen on fig. 3 and 4.

The averages are very similar and are situated around 54.5 cms. On the other hand, we note that the range of measurements is very extensive between 38 cm and 70 cm.

For the passenger cars, the distance from bumper top to ground is around 50-52 cm for recent vehicles.

Impact Speed

Clearly, this is one of the most difficult parameters to determine, and we may not therefore generalize on impact speeds for each vehicle. However, we were able to establish in 13 cases, the truck speed immediately before impact. This measurement came from the chronotachygraph logdiscs. Unfortunately, these logdiscs are difficult to read and since the time between the beginning of braking and the impact is usually extremely short, the speed registered on the disc is often higher than the actual speed on impact. Moreover, the logdisc is often missing or illegible for various reasons.

We have thus 13 speeds varying between 50 and 85 km/h, and one case of 30 km/h. The average speed is 63,8 km/h with a standard deviation of around 15,3, i-e. 67 percent of the speeds recorded are between 48 and 79 km/h. These speeds (even allowing for them being slightly higher than reality) are indeed high, and especially if we consider that for those cases where the log disc was available, the striking passenger cars were still moving.

We will have the opportunity to return to the problem of impact speed.

Different Impact Types

We categorized accidents according to impact type for each vehicle; frontal, rear on lateral, and established 5 classifications (figs. 5 and 6). By Fronto-Lateral, for example, we mean frontal shock for the passenger cars, and lateral for the HGV. The class

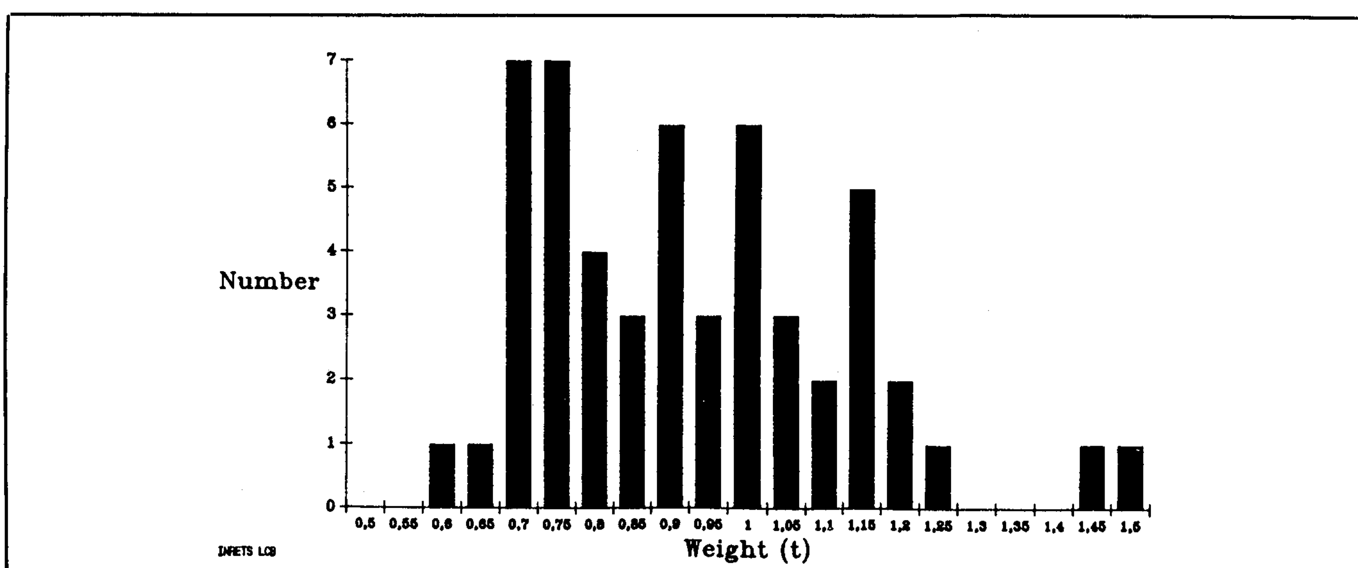

Figure 2. Weight of passenger vehicles

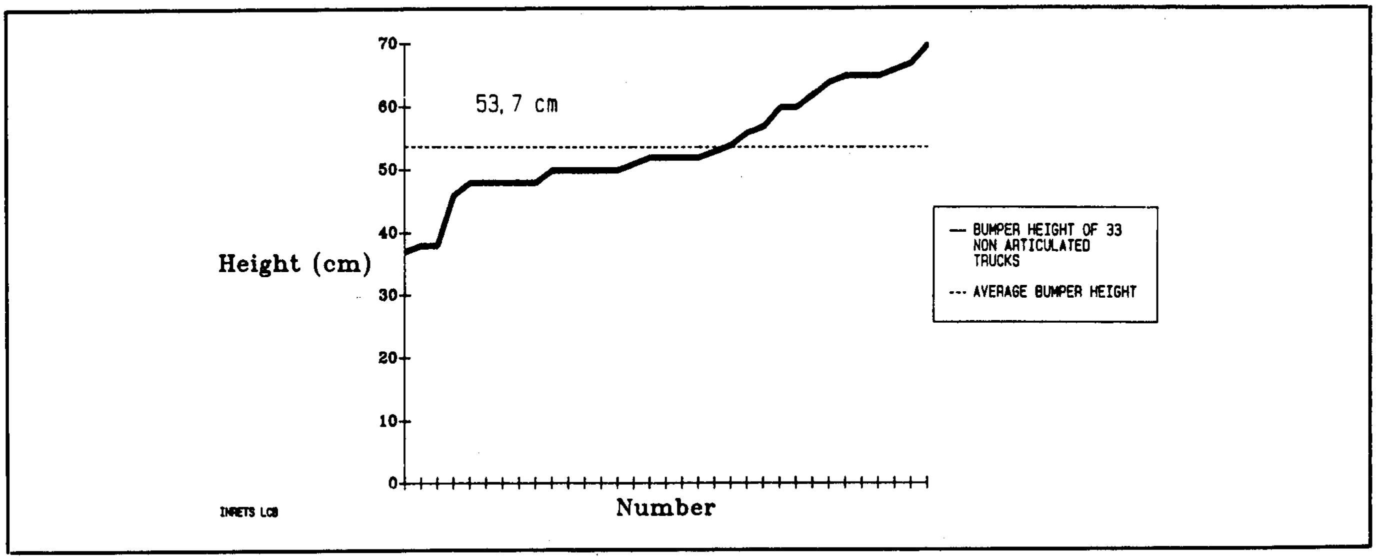

Figure 3. Bumper height of 33 non-articulated trucks

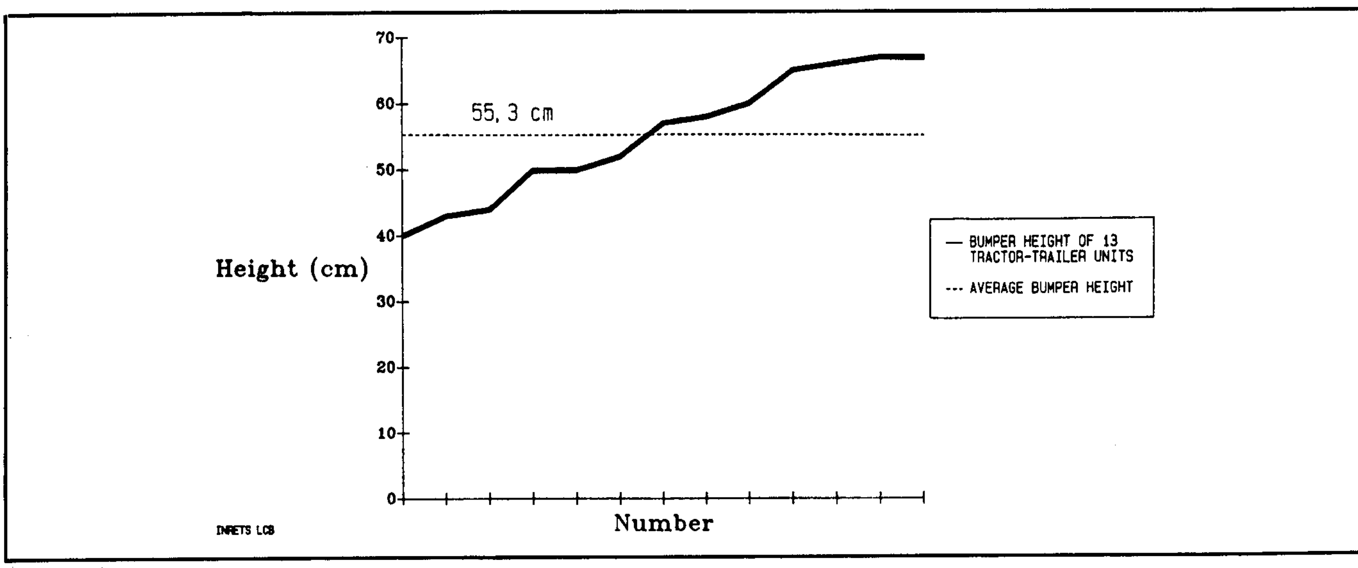

Figure 4. Bumper height of 13 tractor-trailer units

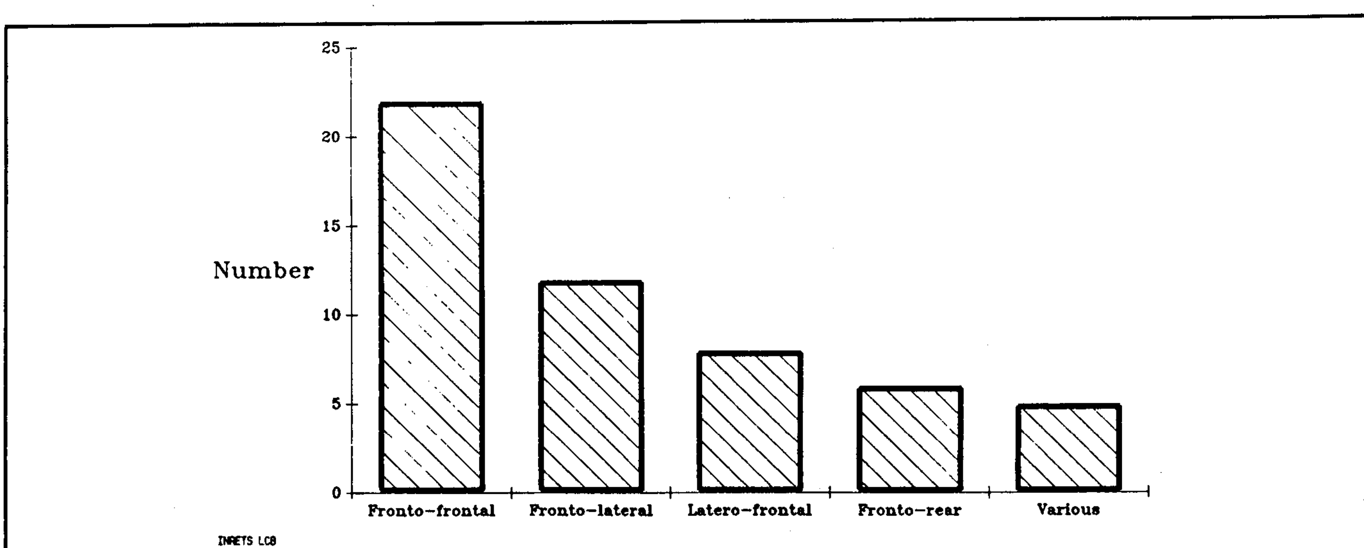

Figure 5. Truck/passenger vehicle impact types

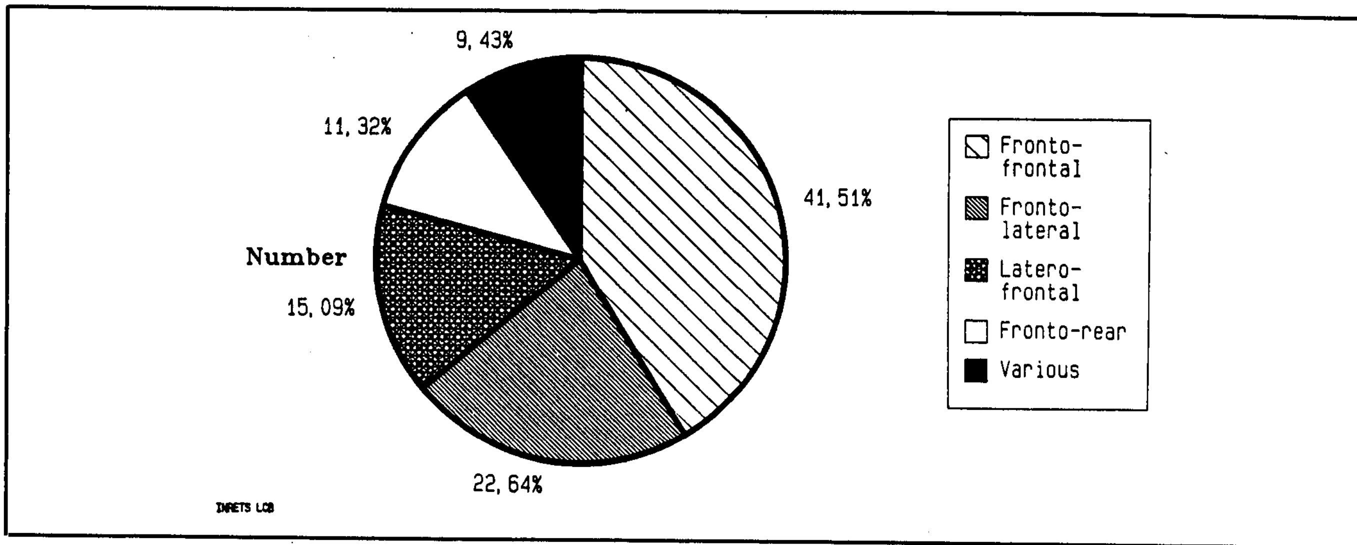

Figure 6. Trucks passenger vehicle impact types (%)

entitled "various" includes impacts difficult to classify and "rear-frontal" shocks.

The most frequent impact type is the Fronto-Frontal type (42 percent of all cases), which most frequently takes place after a loss of control by one of the drivers, or after overtaking.

Fronto-Lateral and Latero-Frontal types have been separated in our classification, since the lesion mechanisms for car occupants are completely different. However, these two types of shocks do have in common the fact that they most frequently occur at road intersections.

They represent respectively 27.6 and 15.1 percent of all cases, i-e, when combined, almost 38 percent of all cases, followed by Fronto-Rear impacts (just over 10 percent) and the "various" category)also around 10 percent). Each impact type will be studied separately.

Impact Direction

By "Impact Direction" we mean the direction of the main force applied to the vehicle during impact. This direction is indicated on a horizontal plane, by a clock face reference system "12 o'clock" representing the direction along the axis of the vehicle from the exterior to the vehicle (Diag. 1).

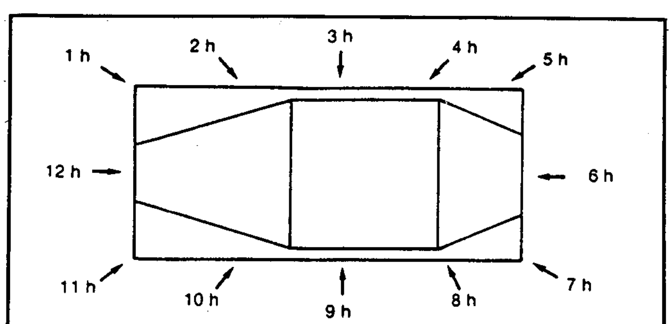

Diagram 1

However, it must be noted that this orientation of force does not allow us to anticipate impact area: it is quite possible to have direction "11 o'clock" and impact on the left-hand rear wheel.

Figs. 7 and 8 show the distribution of impact directions for different vehicle categories. In both cases, the directions 11 and 12 o'clock are in the majority, which confirms the importance of Fronto-Frontal impacts: the most frequent source of these directions. Furthermore, for trucks, we observe a high number of 3 and 6 o'clock directions, which may often indicate lateral and rear impacts.

Impact Location

This is shown on figs. 9 and 10, once again with a clear majority for frontal impacts. However, we observe a difference between trucks and passenger vehicles as far as other impact locations are concerned. For the passenger vehicle, there is a high difference between left and right-hand sides, and few impacts at rear, whereas for trucks, neither left nor right-hand sides predominate, and rear impacts are frequent.

These different accident characteristics seem to indicate the importance of Fronto-Frontal impacts truck-passenger vehicle in accidentology.

The Fronto-Frontal Impact

This type of crash involves 22 trucks and 22 passengers car (with 26 occupants).

Firstly, we examined underrun on impact, and established four types of underrun for passenger vehicles. In the case of underrun, we studied the area of the passenger vehicle which had actually been in collision with the truck. Our four categories are: all the front of the car, two-thirds, one-third, side-swipe. Side-swipe produces low underrun, since the two

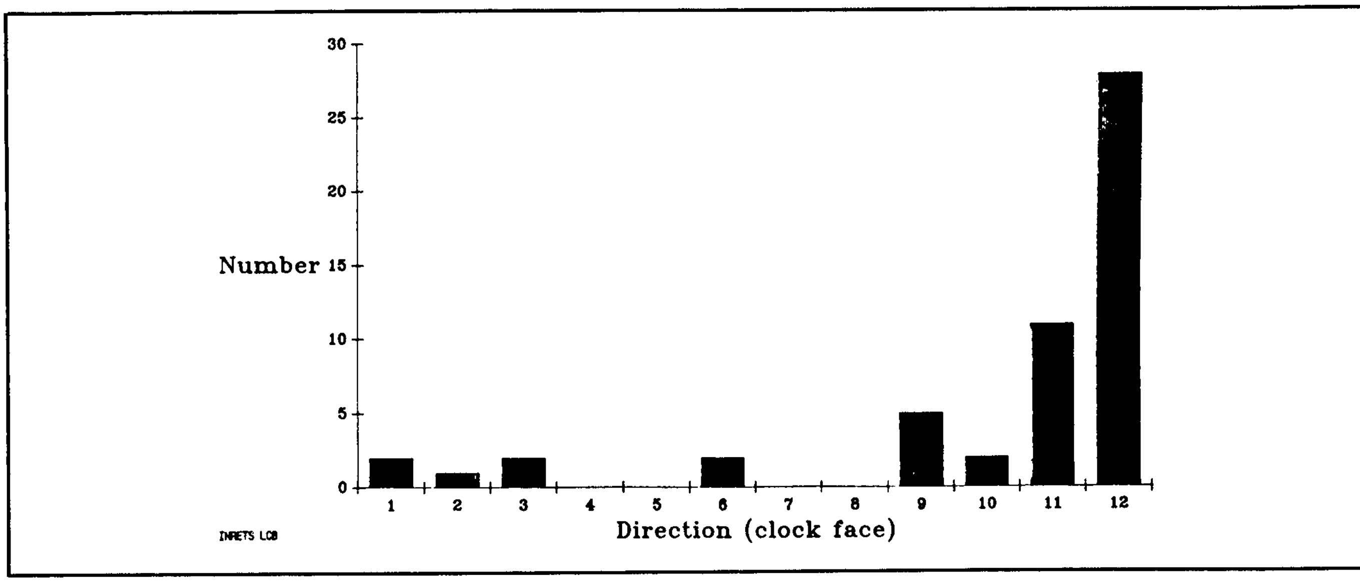

Figure 7. Passenger vehicle impact direction

vehicles tend to move along at a tangent. We refer to figs. 11 and 12, where for approximately 1/3 of all cases, underrun for the passenger vehicle was total, and similarly for the category "two-thirds", whereas side-swipe represents only 14 percent of all cases. This may be explained by the fact that these accidents most frequently originate from a loss of control by the truck or passenger vehicle drivers: and in all the cases, we examined in the "all-front" and "two-thirds" underrun categories, there were never traces of the driver trying to avoid collision, except for some cases of last-minute braking. This is not the case for side-swipe, in so far as this type of accident is usually the result either of a loss of control, and an unsuccessful, too-delayed to avoid collision or else an error of judgment in distances when overtaking or at crossroads.

To study the actual damage caused to passenger vehicles, we use a system of coding called VDI (Vehicle Damage Index), the last parameter—crushing is used below, and is distributed as follows:

VDI	1	2	3	4	5	6	7	8	9
N° of cases	0	2	0	4	5	1	5	2	3

VDI Code no. 5 corresponds to damage reaching windscreen. Thus we see that our sample is mostly made up of seriously damaged vehicles. In fact, damage Code nos. 6-7-8-9 which correspond to some crushing of the survival space area, make up 50 percent of our sample.

As far as occupants are concerned, driver death rate is exceedingly high at 60 percent. We would point out,

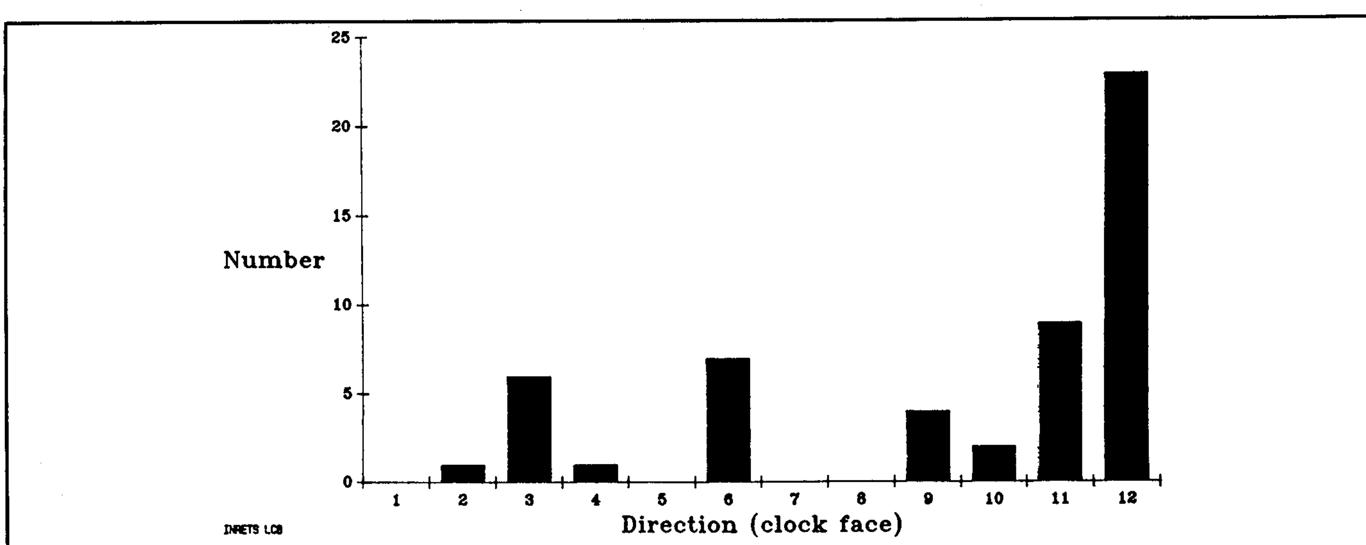

Figure 8. Truck impact direction

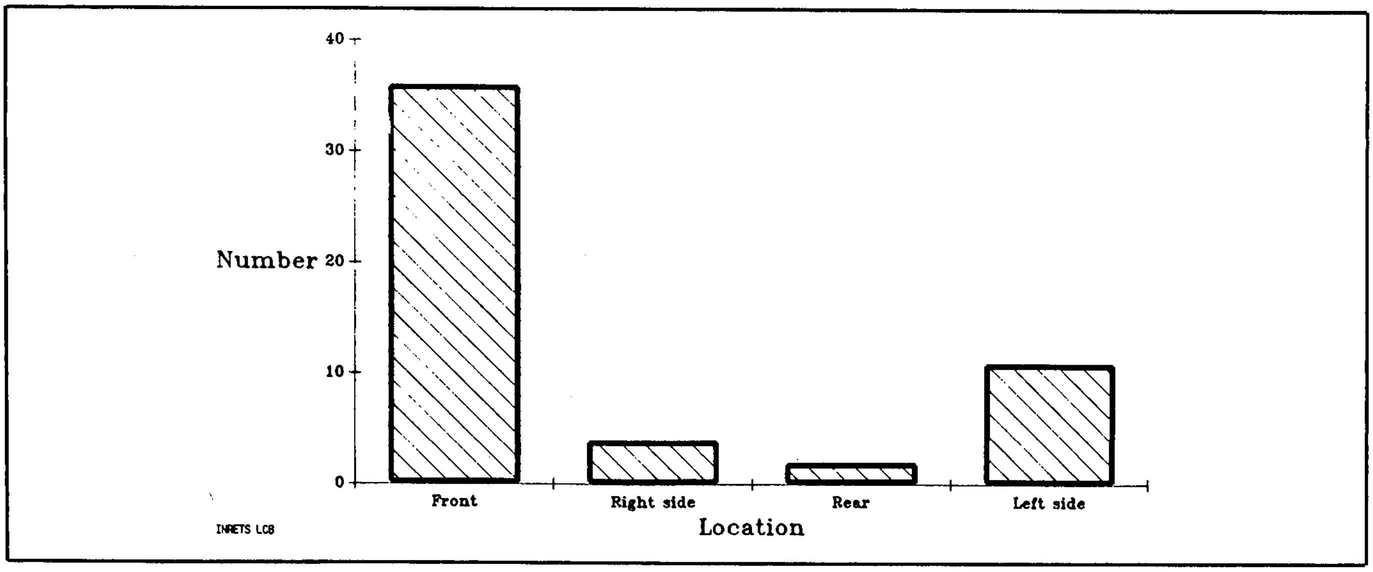

Figure 9. Truck impact location

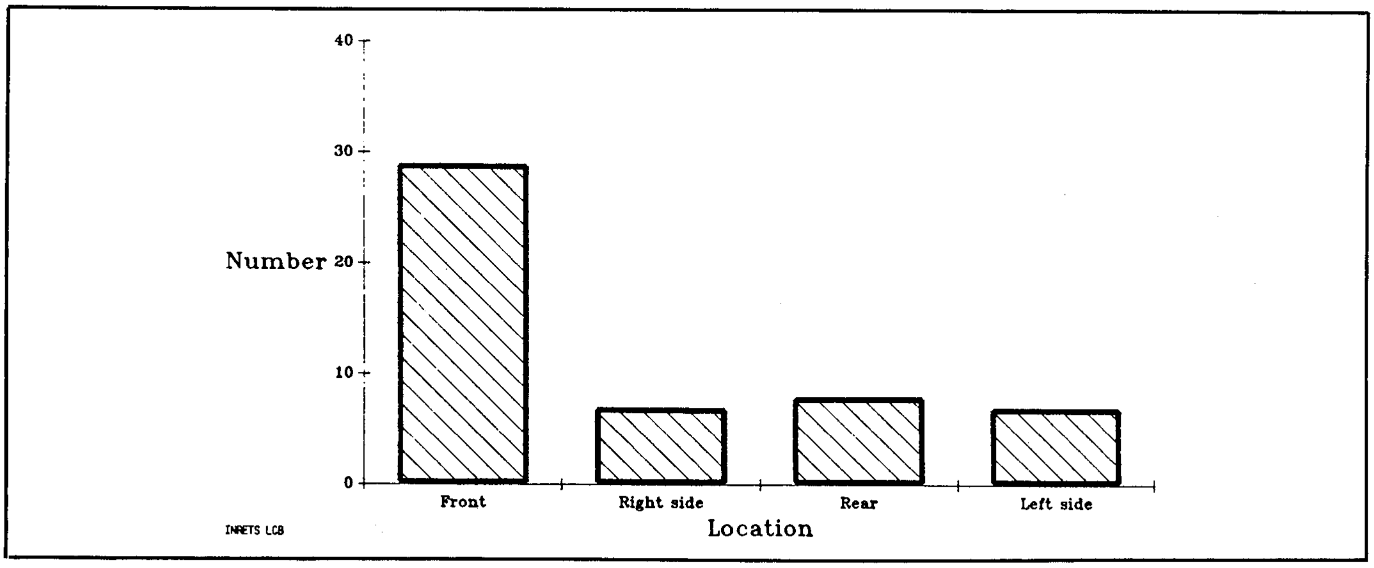

Figure 10. Passenger vehicle impact location

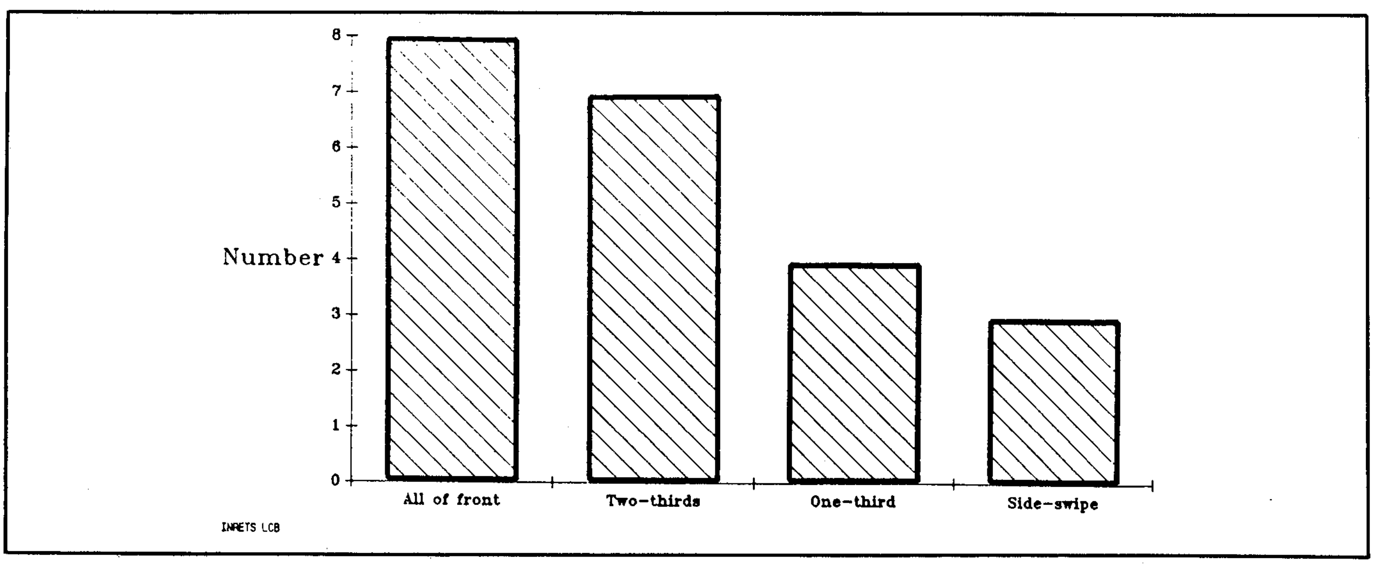

Figure 11. Truck/passenger vehicle underrun

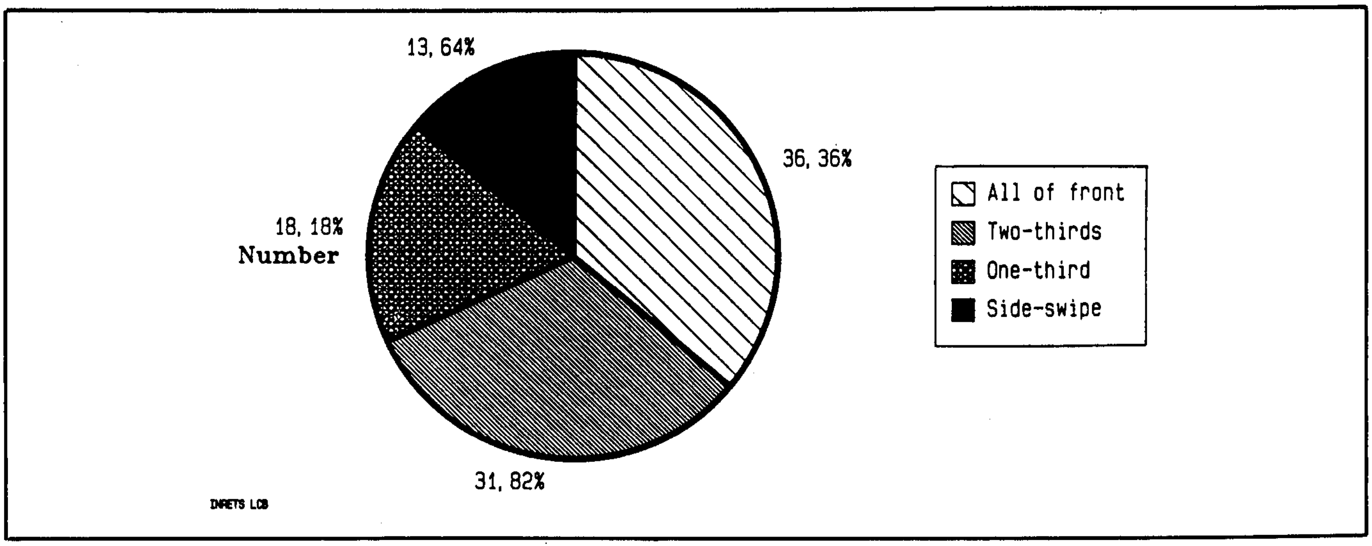

Figure 12. Truck/passenger vehicle underrun (%)

however, that only 20 percent of these drivers were restrained, a very low figure, even considering our study was carried out before the recent French Campaign to enforce seat-belt wearing.

If we compare vehicle damage with occupant injury, we see that out of 11 drivers with vehicle damage Code 6-7-8 or 9, only 2 survived (i-e 82 percent death rate).

These 2 cases merit further study. In the first one, the driver was restrained, and the VDI coded 8. This is a high rate of damage, but in this case, was limited widter since underrun was only one-thirds. The OAIS[1] code for the driver is 3.

In the second case, the driver was unrestrained, and the VDI also 8. However, this was a case of side-swipe, and since there was delayed impact. Furthermore, the passenger vehicle was not overrun by the truck: so the energy of the 2 vehicles was not totally used by the collision, which partly explains the OAIS of 2 (slight injuries). It would thus seem that the highest degree of damage is a criterion of impact severity, but which must be examined in association with underrun.

If we observe the less-damaged vehicles, we find five deaths (50 percent). There is thus a correlation between VDI and injurity severity. Furthermore, for injured surviving drivers (6 drivers), there are 6 cases of side-swipe, thus no underrun. For passengers, there are no severe injuries, but the collisions in question all occurred with delayed impact on left-hand side, i-e, the cabin was not impacted on their side.

[1]The OAIS is the international coding system for occupant injury on a scale of 0 (uninjured) to 6 (deceased).

Fronto-Lateral Impact

This type of impact involves 12 trucks and 12 passengers cars (with 21 occupants).

In this type of impact, underrun is less severe, and the impacting zone of the truck becomes a determining factor. Some zones, such as wheels, and spare wheels, seem particularly aggressive, others less so, such as some types of underrun side-guards. Tractor fuel tanks, on the other hand, appear in our sample to be a relatively favourable zone: we have three cases in which the fuel tank undoubtedly absorbed the shock. It is also of interest that damage to passenger vehicles in this type of collision is clearly less severe than in the case of Fronto-Frontal impact. This is demonstrated in the following table, which indicates VDI distribution: maximum level is 4.

VDI	1	2	3	4	5	6	7	8	9
N° of cases	1	3	5	3	0	0	0	0	0

In confirmation of this result, we find a lower occupant death rate (22 percent), even though only one driver was restrained. Similarly, amongst surviving occupants, we find only one injured person OAIS 4, and 3 OAIS 3, which means that severely injured occupants also represent only 22 percent of total. As it is also the case with Fronto-Frontal impact, avoidance of underrun, when applicable, is a beneficial factor.

Other Impact Types

We have chosen to group together other types of shocks since our sample was only a small one. In spite of the difficulty in the drawing conclusions for this category, we are able to point out that lateral impact is always severe for the passenger vehicle (as is also the case for collisions between 2 passengers vehicles).

Indeed, in our sample, death rate for occupants on impact side, when the cabin is damaged, is 75 percent. Survival rate for occupants on the opposite side depends on two factors: if they are restrained, and the speed of truck impact.

As far as wearing a belt is concerned, our observations are drawn from studies carried out on collisions between passenger vehicles, since none of the occupants in our survey were restrained.

For the second point, as soon as truck speed can no longer be qualified as "slow" (or even "very slow"), we have no survivals. By slow speed, we mean speeds resulting from heavy braking (ground marks) or from an initial shock (which slows the truck considerably).

Fronto-Rear impacts may be assimilated to Fronto-Frontal impacts with one notable difference: the speeds of the two vehicles are to be substracted rather than cumulated.

Truck Occupants

Only two truck occupants were injured. One, at the wheel of a tractor semi-trailer was in Fronto-Lateral collision with a CX, suffered a slight hand wound (OAIS 1); the other sole occupant of a 10 t coach was hit at high speed by a R 25 in a Fronto-Frontal impact, and had a fractured leg (OAIS 2). This second case may be explained by the fact that in a coach, runlike other trucks, the driver's seat is relatively low. Furthermore, the bodywork of coaches is different to that of trucks, and more easily deformable.

Conclusion

Our study included 53 pairs of vehicles involved in accidents: this figure is obviously a very small one. But we would point out that, in 1,985, trucks involved in accidents resulting in corporal injuries represented only 5.5 percent of all vehicles involved.

What may we deduce? First of all, it seems that the most frequent type of impact is the Fronto-Frontal; therefore this type will be the most interesting source of study for Truck/Passenger vehicle collisions.

We have also seen that underrun is an important parameter in impact severity. Our survey shows that the most frequent type of underrun is the two-thirds one. Indeed, this type represents almost one-third of all cases, one other third being complete underrun; the other configurations together making up the remaining third.

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Seat Belt Effectiveness for Heavy Truck Occupants During a Collision

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Abstract

In spite of the abundance of passenger car seat belt studies in various related fields, seat belt tests and studies for heavy trucks, which are substantially different in body structure from passenger cars, have been the minority.

To obtain basic data for evaluating the seat belt effectiveness of heavy trucks, the following were studied:

1) The analysis of prediction of ratios by which the effectiveness of seat belts, based on a statistical study of accidents involving heavy trucks.
2) Sled tests were conducted by simulating the barrier impacts of heavy trucks, and the magnitude of impact on the occupant wearing seat belts and not wearing them was compared.

3) And the tests were conducted with different sled floor acceleration, impact speed, and their influence on occupant behavior was studied.

The results of the above 1) - 3), and their analysis results are reported as the first step in safety study for heavy truck occupants.

Introduction

Many studies have been made so far on the seat belts, those of passenger cars in particular, by leading nations of the world. These studies cover a wide scope, including (1) the evaluation of seat belt effects based on accident survey results, (2) the development of seat belts designed to improve occupant protection characteristics, and (3) the development of passive seat belts.

On the other hand, many nations have made statistical studies of accidents involving heavy trucks, and after analyzing the study findings, have reported that the following are important for the heavy truck safety(1)-(7)

1) The necessity of seat belts
2) Reinforcement of cab strength to secure survival space
3) Rollover countermeasures
4) Prevention of under-ride in front and rear
5) Improvement of energy absorption by the steering wheel and dashboard.

In the case of experiments concerning safety during heavy truck collisions, reports and data are primarily related to 2) and 4) above; that is, the reinforcement of cab strength and the prevention of under-ride(8)-(10).

As for studies of the seat belts of heavy trucks where body construction, dimensions, specifications, etc. differ from those of a passenger car, we can cite a single example made in 1980 in West Germany at the Battle Institute,(11) in which they studied the performance of passenger protection under various belt conditions.

In spite of clamorous appeals for the necessity of seat belts, studies dealing with the seat belts of heavy trucks are seldom found among survey analysis and the results of heavy truck accidents.

To investigate the effectiveness of seat belts when a cab-over type heavy truck (GVW, 20 tons) has a collision, we made a systematic study of the scale of impact on the occupant and the scale of injury to the occupant in relation to impact speeds using and not using seat belts through (1) the analysis of prediction of ratios by which the effectiveness of seat belts, based on a statistical study of accidents involving heavy trucks are shown, (2) sled tests under various seat belt systems and (3) a full scale collision test as a preliminary test. Reports thereof follow below.

In the sled test, experiments were made so as to generate a nearly equal impact to that of a heavy truck when it makes a frontal collision with a flat barrier. To establish the impact conditions for the sled test, a test of frontal collisions of heavy trucks with the flat barrier was made first to determine the scale of impact on the body during collision.

On the Effectiveness of Heavy Truck Seat Belts Seen From Accident Survey Findings

Figure 1 shows the heavy truck accidents (9,710 cases) involving heavy trucks in 1983 in Japan by collision areas. Accidents involving pedestrians and motorcycles are excluded.

The front is the overwhelmingly dominant collision area with 72.2% of the total suggesting a considerable level of seat belt effectiveness as long as a survival space has been secured in the cabin.

Figure 2 shows the occurrence ratio of the part of total accidents which consists of accidents where the cab front was hit and single accidents (roll-overs, collision with structures, etc.) where occupant injury levels would have been lowered by seat belts.

The hatched area in the graph shows the number of drivers killed or seriously injured in spite of little cab deformation (that is, the survival space in each cabin was secured) in the rear-end collisions, frontal collisions and single accidents. In other words, this data shows that seat belts would have alleviated the injuries suffered by this number of drivers.

In the case of heavy truck accidents involving deaths and serious injuries, it is predicted that such injuries would be alleviated by about 25% if secondary impact in the cabin is eased, or prevented, by using seat belts.

The number of the dead or seriously injured in the graph (78 persons) are for the drivers alone and the number will increase if the passengers are included.

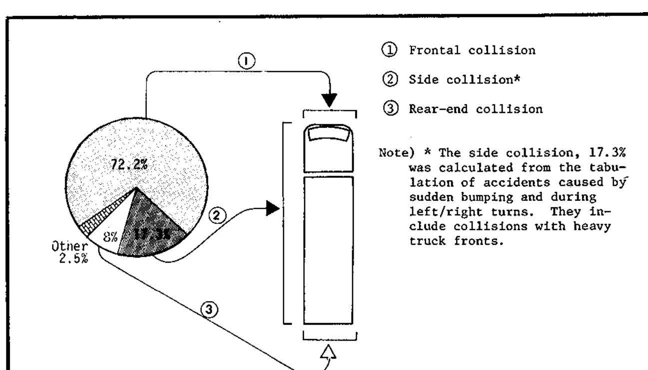

Figure 1. Heavy truck accidents by collision areas (In Japan, during 1983)

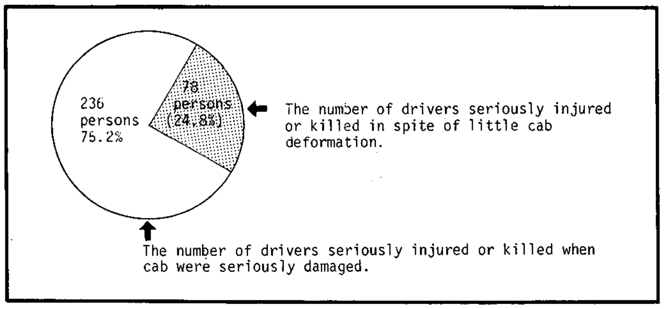

Figure 2. Prediction of seat belt effectiveness in heavy truck accidents

As seen, a rough picture of the degree of the effectiveness of seat belts in heavy truck accidents was envisaged from accident survey findings. Though accident surveys can clarify statistical ratios, they cannot explain how impact velocity and degrees of acceleration affect drivers' injuries. Hence, full scale collision tests and sled tests were made to draw conclusions concerning these effects.

A report of these tests follows:

Frontal Collission Test of a Heavy Truck With a Flat Barrier

To establish impact conditions for a sled test required to review the effectiveness of seat belts when a heavy truck makes a frontal collision with the barrier and to investigate the passenger behavior during impact condition, a preliminary frontal barrier impact test of a heavy truck was made to study the scale of impact applied to the cabin, the duration time, etc.

A special emphasis was put on the degree of cabin floor acceleration which greatly affects an occupant's injury level in a collision.

In the current test, the collision speed was determined to be 20 mph (32 km/h) based on foreign studies(5),(11),(12) made so far on

(1) Driving speeds
(2) The composition of impact speeds in heavy truck accidents and
(3) Impact speeds used in actual test.

In addition, to study dummy behavior in a collision, three Part 572 dummies (HYBRID-II) were put on the driver's seat, the center seat, and the passenger's seat with a 3-point seat belt (NLR), a 2-point seat belt, and a no belt, respectively. A measurement of injury values was also carried out.

Figure 3 shows a test scene. To examine dummy behavior during a collision, the door panel was cut open for the test but it was reinforced to provide the same longitudinal crush strength of the door in a collision as before the cutting (the standard condition).

Figure 3. A test scene

[Test Results]

Figure 4 shows the deformation around the cabin, after the test.

i) Cabin deformation

As the main frame is highly rigid, the deformation of the cabin after collision is slight, roughly 60% of the passenger car level if absolute values of deformation are compared. The crush stroke of cab front-panel is approximately 180 mm and the survival space in the cabin is secured sufficiently.

ii) Acceleration of the Cabin

The waveforms of cabin floor acceleration, measured in the neighborhood of the driver's seat and the

Figure 4. Cab deformation after the test

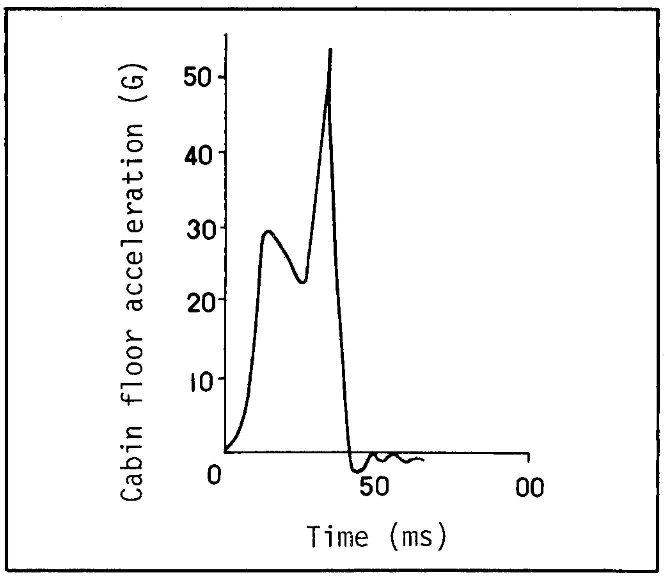

Figure 5. Waveform of cabin floor acceleration

passenger's seat, have two peaks of duration time of acceleration of 40 to 45 ms and the maximum acceleration of 55 to 60G as shown in Figure 5. As a whole, the waveform can be patternized as a triangle wave.

Compared with those of passenger cars, this waveform features a short duration time, roughly 50% shorter, and a steep rise.

Irrespective of the occupant's use of a 2-point seat belt or no belt, ride-down effects are not usually realized unless the duration time is longer than 100 ms. Therefore, in the case of an impact where the duration time is 40 to 45 ms at best, as in this test, the occupants are caused secondary impact with interior structures and devices at the initial speed maintained before the collision after the vehicle is stopped without respect to the shapes of the acceleration waveforms of the cabin. This short time waveform therefore implies critical conditions for the driver.

iii) Dummy behavior

Figure 6 shows the behavior of a dummy during impact condition by the time series.

- Driver's seat (3-point belt)
  The knees come in contact with the lower part of the instrument panel as the belt extends. However, other parts do not come in contact with the panel.
- Center seat (2-point belt)
  The torso fell forward with its hip in position to allow the head and face to hit the instrument panel.
- Passenger's seat (without belt)
  After the knees hit the instrument panel, the torso fell forward while the hip was raised upward, and the head went out through the front glass to reach the barrier.

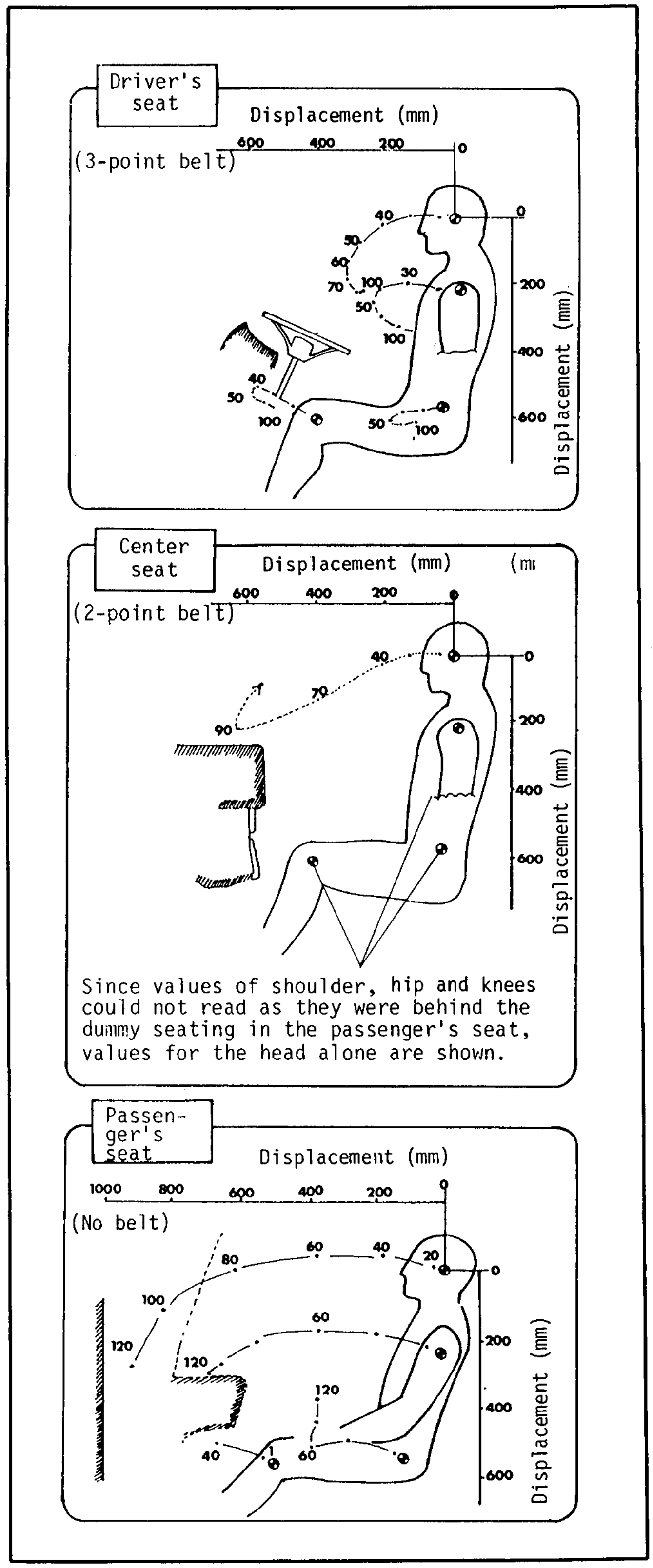

Figure 6. Dummy behavior during impact condition

iv) Values of passenger injury

Figure 7 shows the injury values obtained from this test as reference value.

1) If a 3-point belt is worn, impact on the head, the chest and the femur are all below the injury criteria of the human tolerance prescribed by FMVSS 208, but
2) If a 2-point belt is worn or no belt, the head comes in contact with the instrument panel or with the barrier front, the head penetrates into the windshield, respectively, pushing up head injury values much higher than HIC = 1000.
3) As for the chest and the femur, injury values are less than the injury criteria of FMVSS 208 irrespective of the use of a belt.

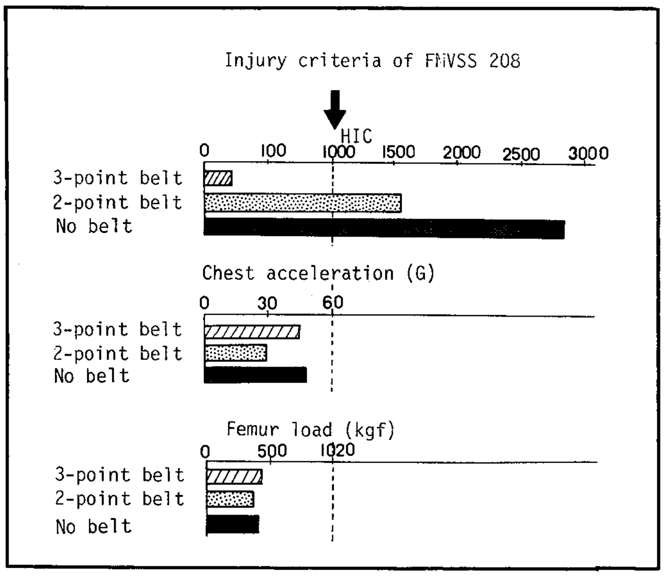

Figure 7. Passenger injury values (frontal collision of a heavy truck with the flat barrier at 32 km/h)

Sled Test

Outline

To study the effectiveness of a seat belt when a heavy truck makes a collision, the HYGE impact test equipment, hereafter called "the sled", was generates when it makes a frontal collision with the flat barrier. In the test, a device modelled on the compartment of the driver's and passenger's seats of a cab-over type heavy truck was fixed on the sled and a given level of impact was administered to it after loading belted dummies or a beltless dummy on it

Figures 8 to 10 show the test conditions.

Part 572 dummies were used as human body dummies and seat belts were fastened in the following manner:

(1) Driver's seat	1) 3-point belt with NLR 2) 2-point belt with NLR 3) No belt
(2) Assistant's seat	1) 3-point belt with ELR (that senses floor acceleration of 0.4G) 2) 2-point belt with NLR 3) No belt

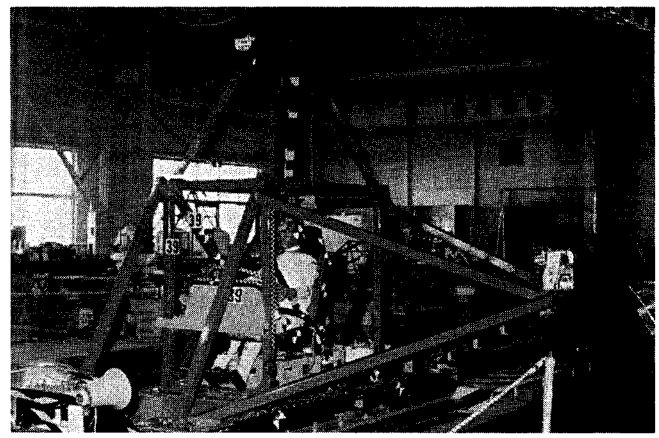

Figure 8. A sled test scene

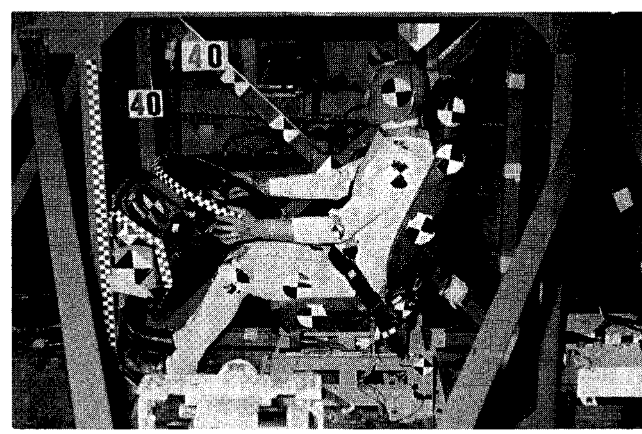

Figure 9. Driver's seat (3-point belt with NLR in use)

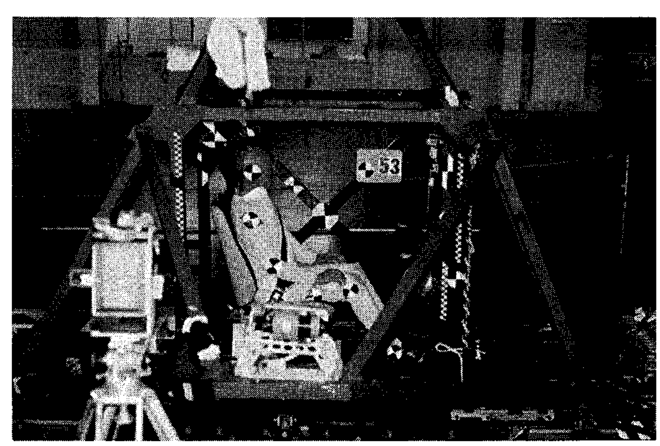

Figure 10. Assistant's seat (3-point belt with ELR in use)

Impact Conditions

Generally, the duration time of acceleration applied on the body floor in a frontal collision with the barrier remains almost constant irrespective of the impact speed and, in the case of a heavy truck, is roughly 50 ms.

In the sled test, the duration time of the floor acceleration of the sled was set constantly at around 50 ms to give a similar impact to that generated by the heavy truck when it makes a frontal collision with the flat barrier and impact were given under various acceleration degrees, that is, different impact speeds.

Test Results and Review of Results

1) Comparison of the results of a full-scale collision test and of a sled test

Figure 12 compares the waveform of cabin floor acceleration in the full scale frontal collision test (32 km/h) with that of the sled floor acceleration in the sled test (at 37 km/h). In the case of the full scale test, the waveform shows a very steep rise but, in the case of the sled test, it is as if the waveform of the former is simply patternized into a triangle wave, or a half sine wave.

Figure 13 compares the full scale test results with the sled test results in terms of injury values incurred at different sections of the dummy seated on the driver's seat with a 3-point belt.

The black round marks in the drawings indicate the results of the full scale test. The sled test results relatively correspond to the full scale test results. This means that the behaviors of dummies correspond in both tests.

2) Dummy behavior

Figure 14 shows the dummy behavior during impact in the passenger's seat by the time series.

When a 3-point belt was worn, neither the head nor the chest come in contact with the instrument panel. In the current test, however, the knees come in contact with the lower part of the instrument panel if the speed exceeded 35 km/h, restricting the knees behavior.

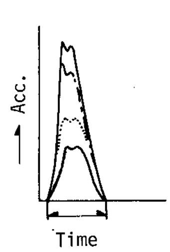

Figure 11. The waveforms of floor acceleration of the sled

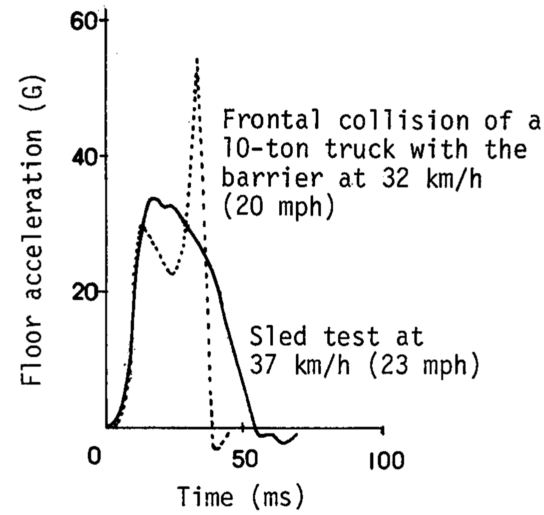

Figure 12. Comparison of the acceleration waveforms in the full scale test and the sled test

If a 2-point belt was used, the torso fell forward with its pelvis in position to allow the head to hit the instrument panel top.

In the test, the speed ranged between 17 and 42 km/h, but a certain area of the instrument panel top was hit by the head without exception. If the capacity of impact energy absorption of this area is improved, it is possible to reduce the impact on the head.

If no belt was used, the dummy moved forward in its initial position at first, hit the lower part of the instrument panel with the knees, then hit the instrument panel front with the chest. Then, while the pelvis was raised upward, the head hit the instrument panel top, and then hit the windshield. The dummy was not thrown out of the vehicle in the test as a preventive net was fixed on the windshield. In an actual accident, however, it is likely that the passenger would be thrown out.

3) Head injury Criteria (HIC)

As an example, Figure 15 shows HIC changes in impact speeds derived from the test results with and without a belt.

The straight and dotted lines in Figure 15 show the driver's and the passenger's value, respectively.

HIC = 1000 is reached around 35 to 45 km/h when either a 2-point belt is used or no belt, while it is reached at around 55 km/h* (on the passenger's value) when a 3-point belt is used.

If these are evaluated by the impact speed at which a heavy truck makes a frontal collision with the flat barrier, HIC = 1000 will hypothetically be reached around 30 to 40 km/h if a 2-point belt or no belt is used and around 50 km/h* (on the passenger's seat position) if a 3-point belt is worn.

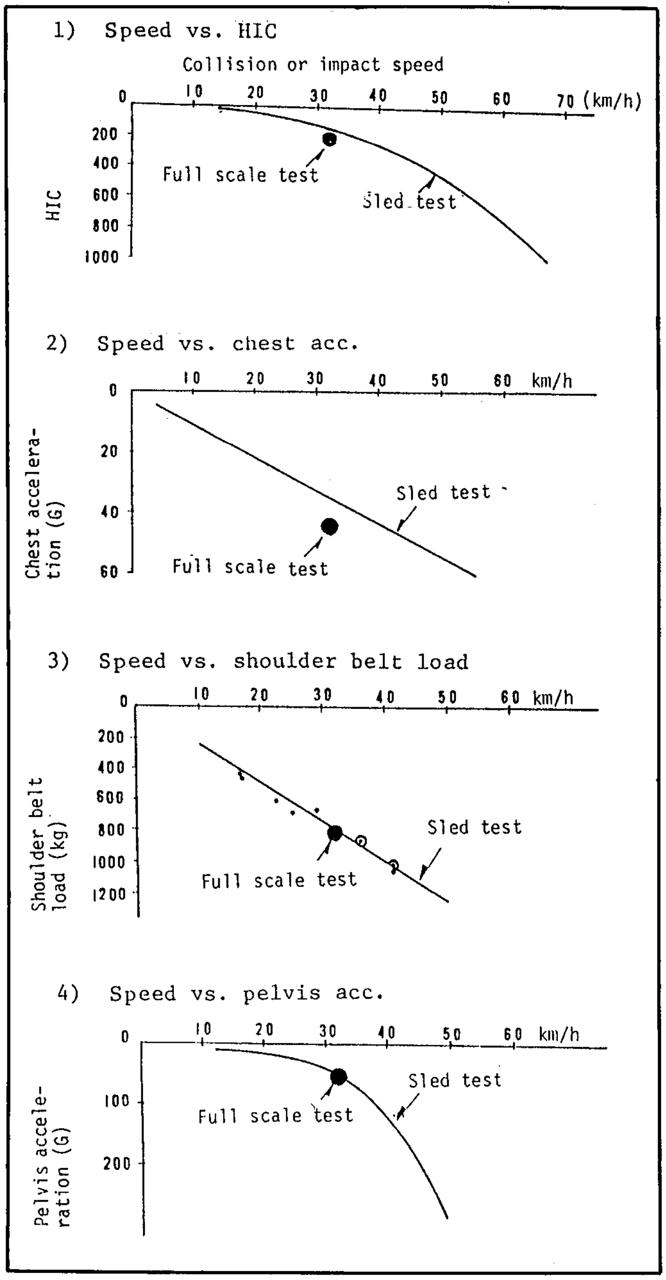

Figure 13. Comparison of test results between full scale test and sled tests

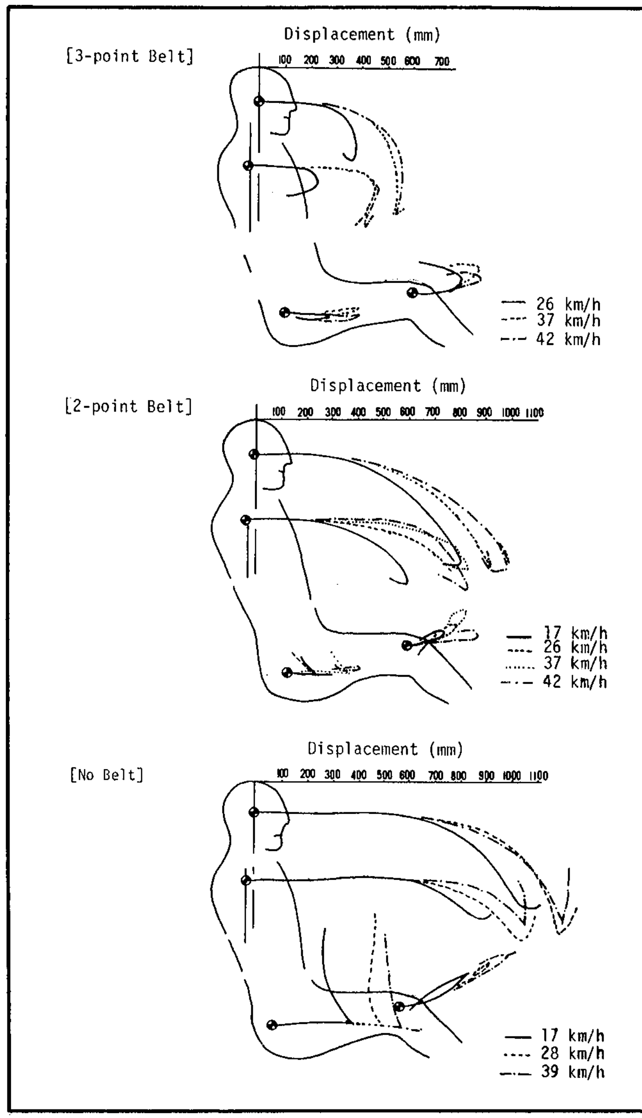

Figure 14. Dummy behavior during impact

Note: The corresponding speed on the driver's seat position to the asterisked speed is predicted to be over 60 km/h but, as the degree of extrapolation is large enough as shown in Figure 15 1), the evaluation was made on the passenger's seat position.

The report thus far has been a prediction of marginal speeds based on test results and only shows the results of an example. If whether or not this represents the general tendencies of a heavy truck should be judged after examining the results of many tests to be made in the future. The same applies to the chest injury value, etc. explained to the right.

4) Chest acceleration

Figure 16 shows the relation between impact speeds and chest injury values (3 ms G).

For the same impact speed, the injury value on the driver's seat is different from that on the passenger's seat but the impact speed at which the injury criteria of FMVSS-208 of the chest of 60G (3 ms cut G) is reached is roughly 35 km/h if no belt is used and roughly 45 to 65 km/h if a 2-point or 3-point belt is used. This clearly suggests the use of a belt is effective.

5) Femur load

Figure 17 shows changes in the femur load by impact speed when a belt is used and when it isn't.

While the femur load remains less than $\frac{1}{2}$ of the limit value within the scope of the current test only if a belt is worn 1) and 2) of Figure 17, the human

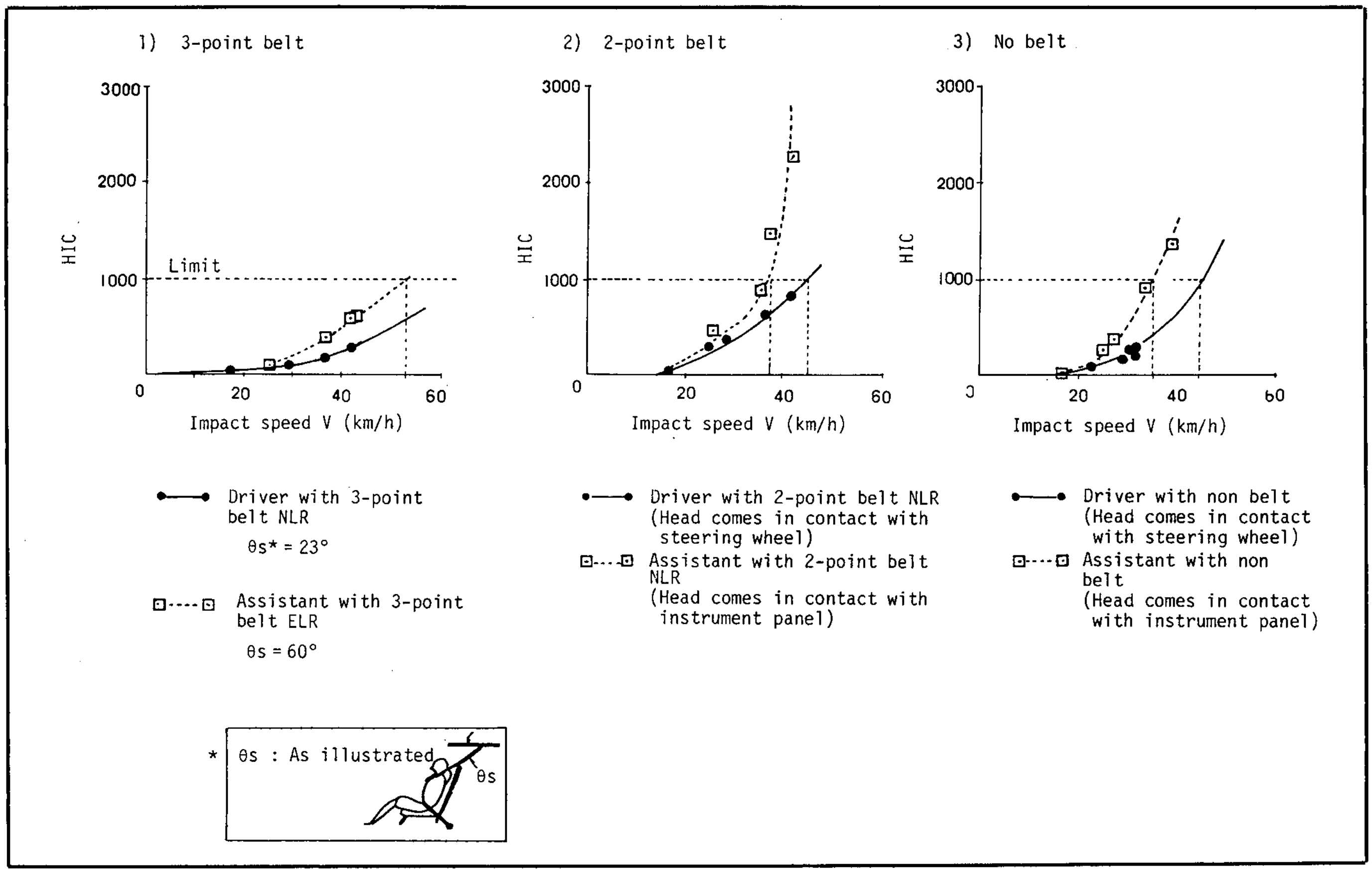

Figure 15. HIC changes in impact speeds

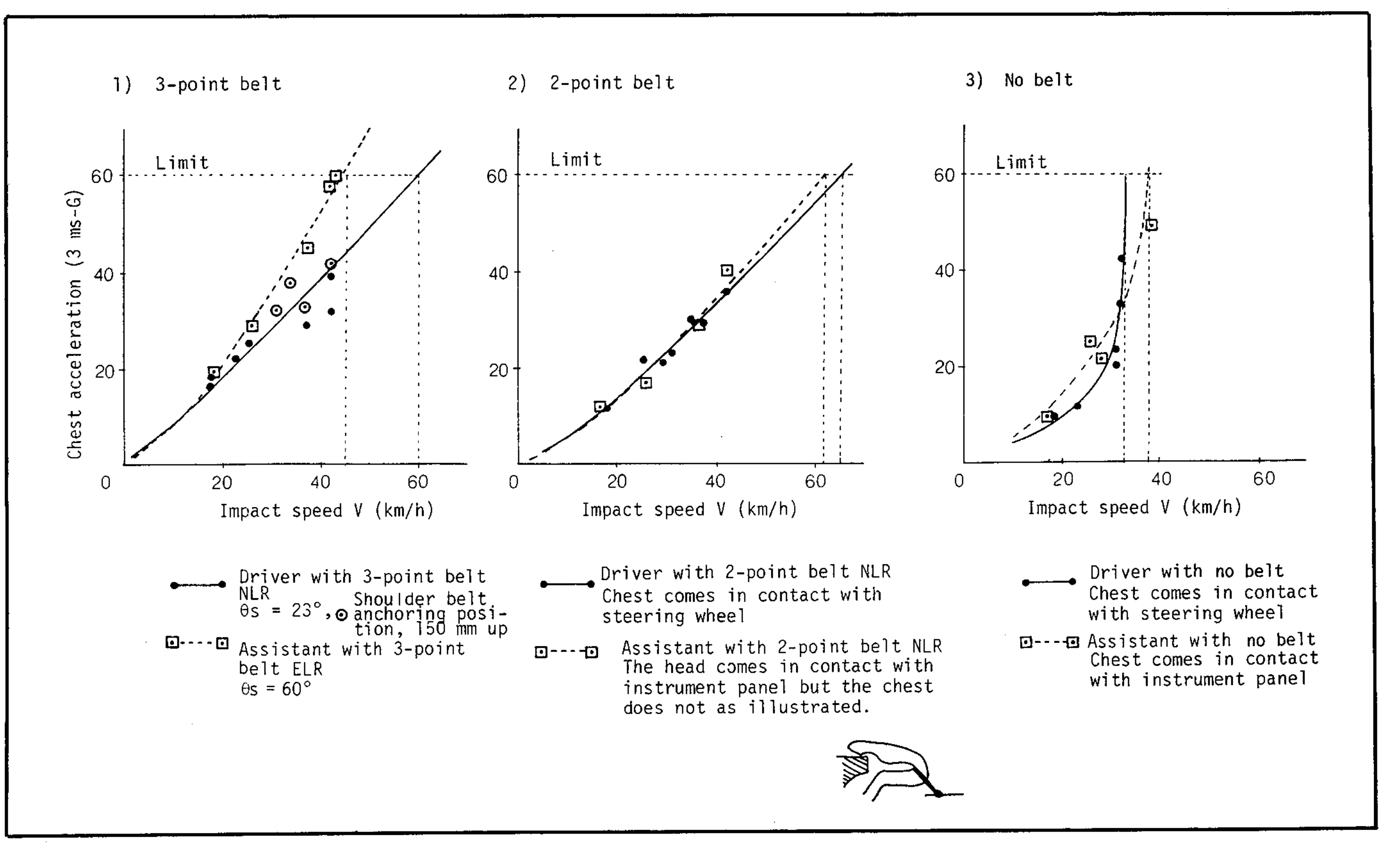

Figure 16. Changes in the chest acceleration by impact speeds

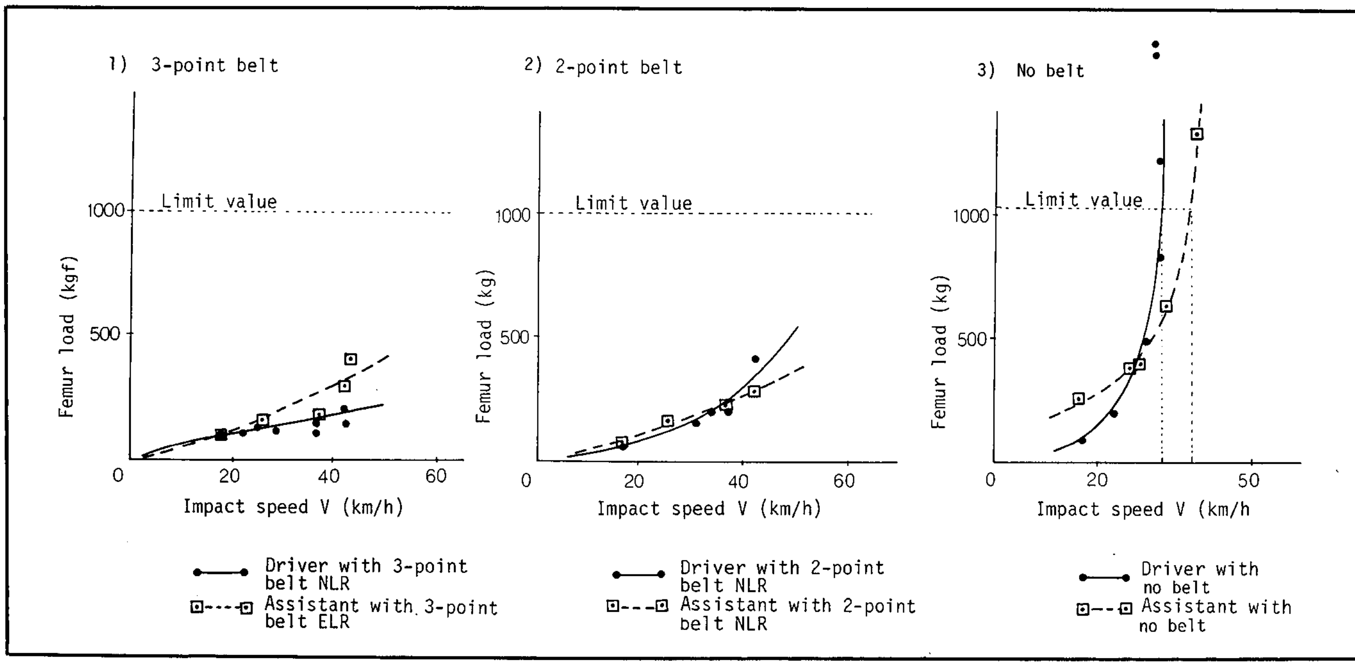

Figure 17. Changes in femur load by impact speeds

tolerable level (1,021 kg) is already reached at an impact speed of roughly 35 km/h, if no belt is used 3) of Figure 17, as the dummy's knees come directly in contact with the instrument panel, clearly suggesting the effectiveness of seat belts.

6) Occupant injury values by impact speeds

So far, occupant injury values by impact speeds in the sled test have been analyzed. In Figure 18, the above speeds are converted into the impact speeds at which a heavy truck makes frontal collisions with the barrier based on the results of

Plotted speeds in the graph show the limit speeds at which the injury criteria prescribed by FMVSS-208 are deemed to have been reached.

As the conditions of the driver's seat are different from those of the passenger's seat with respect to the following, comparison cannot be made directly.

1) Seat belt anchorage position.
2) Retractor type (NLR for driver, ELR for passenger).
3) Secondary impact position of passenger and respective absorption of energy.
4) Alloted rate of loads on the shoulder and lap belts, etc.

If seat belt effectiveness is reviewed in general, one 3-point belts, two 2-point belts and three no belts are effective in protecting the passenger, in that order.

The limit speeds on the driver's seat position when a belt is used and when it isn't are as follows:

Non belt	at around 25 to 30 km/h, the chest or femur will reach its limit
2-point belt	at around 40 km/h, the head will reach its limit
3-point belt	at around 50 km/h, the chest will most likely reach injury criteria prescribed by FMVSS-208.

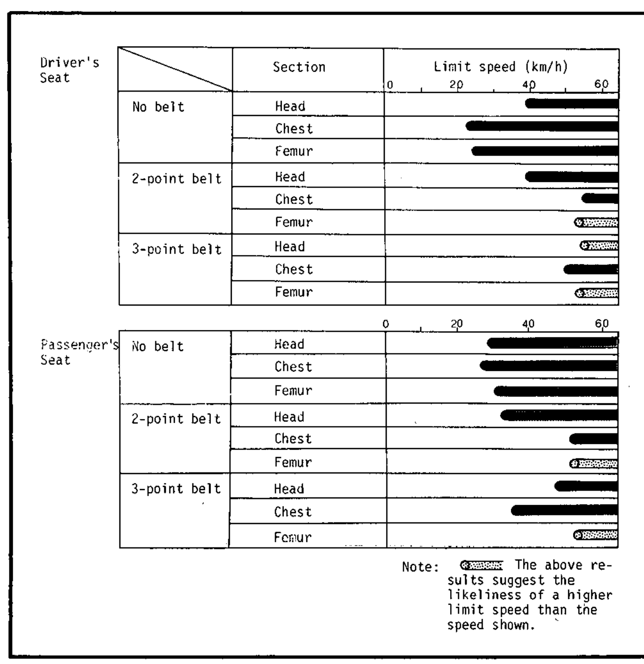

Figure 18. The collision speeds at which the human tolerance (injury criteria prescribed by FMVSS) are reached

Similarly, limit speeds on the assistant's seat position are:

Non belt	at around 25 to 30 km/h, the chest will reach its limit.
2-point belt	at around 30 to 35 km/h, the head will reach its limit.
3-point belt	at around 35 to 40 km/h, the chest will reach its limit.

7) Occupant injury values by seat belt fastening angles

Figure 19 shows the relationship between HIC and the impact speed of a 3-point ELR on the passenger's seat by using shoulder belt fastening angles as parameters.

In the test, the angle of belt fastening (θs) was changed from 26° to 60° but HIC values changed little if impact speed remained the same.

Figure 20 shows a similar relationship with respect to the acceleration of the chest. In this case, the chest injury values tend to be influenced by shoulder belt angles.

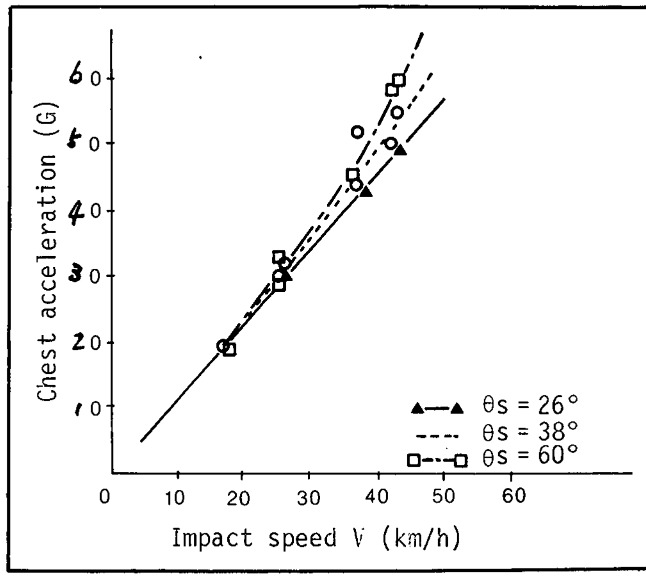

Figure 20. Impact speed vs. chest acceleration; where the angle of the shoulder belt fastening is used as a parameter

Conclusion

The findings of a survey of the effectiveness of a heavy truck's seat belts derived from accident survey results, the results of full-scale heavy truck collision tests, and the results of sled tests made to review the effectiveness of a heavy truck's seat belts explained so far can be summarized as follows:

1) The effectiveness of a heavy truck's seat belt derived from accident survey results

As a result of investigations into the effectiveness of a heavy truck's seat belts by using the statistical data of heavy truck accidents that occurred in Japan during 1983, it is reasonable to predict that the number of fatalities and seriously injuries could be reduced by roughly 25% if occupants of heavy trucks were wearing seat belts.

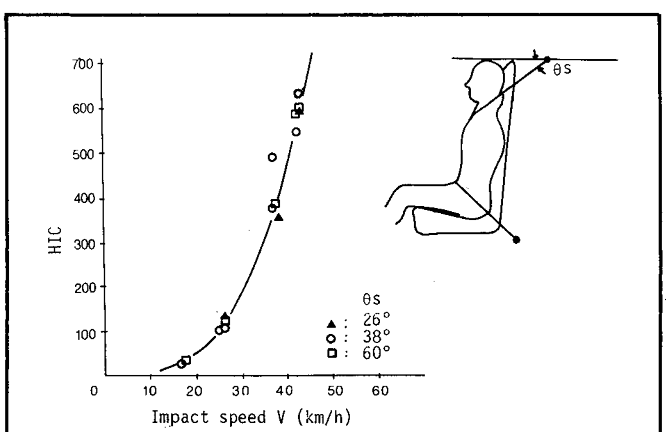

Figure 19. Impact speed VS HIC, where the angle of the shoulder belt fastening is used as a parameter

2) A full-scale heavy truck collision test

As a result of a front collision test with the flat barrier of a cab-over type truck in the GVW 20-ton class at an impact speed of 32 km/h, it was discovered that

1. The deformation of each section of cabin was minimal and the cabin's survival space after tests was secured sufficiently.

2. The acceleration of the cabin floor, which is reproduced as the floor acceleration of the sled test, featured a waveform where the duration time was roughly 50% shorter and the rise was steeper than that of a passenger car. This waveform is critical in that it suggests secondary collisions of passengers.

3. While the injury values of the head, chest, and femur do not reach the human tolerance prescribed by FMVSS 208 if a 3-point belt is in use, the corresponding HICs of the use of 2-point belts and no use of belts are well over 1000.

3) Sled tests

1. By the sled tests, a dummy behavior similar to that in the collision of a heavy truck with the barrier was reproduced.

2. As a result of sled tests similar, it is clear that the use of belts is effective for passenger protection. On the driver's seat, an impact speed corresponding to the human tolerance prescribed by FMVSS 208 is predicted as follows:

25 km/h or so if no belt is used.

40 km/h or so if a 2-point belt is used.

50 km/h or so if a 3-point belt is used.

3. In the test, attempts were made to identify the influence of the angle of the shoulder belt fastening (θs) on the passenger injury value. As a result, it was found that the HIC had almost nothing to do with the value of θs, but chest injury value tend to be influenced by shoulder belt angles.

Recommendation

This time, we made various tests and studies to examine the effectiveness of the seat belt as part of our study of a heavy truck's occupant protection and could clarify its effectiveness to some extent. The following, however, are conceivable future problems.

1) When a 2-point belt is used, the head and chest comes in contact with the steering wheel in the driver's seat and the head hits against the instrument panel in the passenger's seat, making damage more serious. The energy absorption by the steering wheel and instrument panel during secondary collision will have to be clarified further from the viewpoint of occupant protection.

2) When an ELR-type seat belt is installed in a heavy truck, attention should be directed to the following:

The range of standard acceleration sensed by the current retractor locking mechanism should be set between 0.45 and 0.85 by each country in the case of an acceleration sensing formula for the vehicle body. But the level of sensed acceleration varies from country to country, and Japan which approved 1.5 G products for automobiles produced after September 1987.

In addition to a study of the inconveniences caused by seat belts, including excessive constriction of the stomach during rough road driving, a reasonable standard acceleration value should be agreed to internationally in the future.

3) In this test, rigid seats without suspension were used, but the use of suspended seats is possible for a heavy truck and suspension seats will also have to be examined in the future.

Acknowledgement

Lastly, we would like to express our deep appreciation for the cooperation of JAMA's members in the heavy truck study committee and the heavy truck cab impact W/G committee during this study.

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Front Underrun Guards for Trucks

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Abstract

This paper presents research which is a continuation of earlier work on front underrun guards for trucks, carried out at the Transport and Road Research Laboratory in Great Britain.

Results are given from impact tests between European small cars and trucks, where both vehicles are moving towards each other. Comparisons are made with earlier tests at similar closing speeds but where the truck was stationary before impact. It is concluded that there is little difference in the results from either method of test, particularly for the case when an underrun guard is fitted.

The results suggest that an energy absorbing truck front underrun guard with a ground clearance of 300 mm is able to provide protection from fatal or severe injuries to seat belted occupants in a small car (750 kg) at closing speeds up to about 65 km/h.

Impact tests have been carried out between a rigid faced 250 kg trolley and three different types of energy absorbing truck underrun guard. The test procedure thus developed could form the basis of a legislative test to determine whether front underrun guards have adequate energy absorption. Limits are suggested for the proposed test procedure to cover speed of impact, input energy, size of trolley face and trolley decelerations during impact.

Introduction

This paper describes work carried out at the Transport and Road Research Laboratory in Great Britain and is a continuation of research presented at earlier Experimental Safety Vehicle Conferences in 1980[1] and 1985[2]. On this occasion it is concerned only with underrun guards fitted to the fronts of heavy goods vehicles to reduce the severity of injury to car occupants.

Accident data

Table 1 shows the number of car occupants killed each year in GB in accidents involving goods vehicles of gross weight over 7.5 tonnes for the period 1974 to 1985. The number of car occupants killed annually in all other road accidents is also shown.

There was a general decline in the numbers of all categories of road user killed in truck accidents until around 1980. This may possibly be attributed to the gradual diversion of heavy goods vehicle traffic to the safer motorway roads in this period. Since 1980 the number of occupants killed each year has levelled off to around 320. Approximately two-thirds of this number are killed when their vehicles impact the fronts of trucks, most frequently when the two vehicles are travelling in opposite directions and the front of the car hits the front of the truck.

Heavy goods vehicles are very aggressive towards cars in collisions for several reasons. The large mass ratio results in the velocity change of the car being much greater than that of the truck. The height of the truck structure is such that, in a collision, the car may run under the truck, often to the extent that the truck structure comes into contact with the car occupant compartment or in extreme cases into direct contact with the occupant. By this mechanism, the important energy absorbing zones of a car, which tend to be below the truck structure, are not used to protect the occupant. Finally, because of the rigidity of the truck structure, most of the impact energy is dissipated in the car rather than in the truck.

Truck front underrun guard design

The first essential step in providing protection for car occupants is to lower the front structure of the truck so that effective use may be made of the energy absorbing structure of the car. It is also desirable to make the lowered structure of the truck energy absorbing. This has the effect of increasing the maximum survivable closing speed for the car occupants, which is particularly beneficial in car front to truck front impacts where closing speeds tend to be higher.

The amount of energy absorption that may be provided by the truck frontal structure as it yields is limited by two factors. Firstly, the total longitudinal crush of both the car and truck structure must not exceed the original length of the car bonnet. Secondly, the force level at which the truck structure yields must be compatible with the forces at which a car front collapses so that at the end of a severe impact both have crushed. This latter point is important. If a guard or other form of protection is very soft, it will

Table 1. Car occupants killed in road accidents in Great Britain.

Year	1974	1976	1978	1980	1981	1982	1983	1984	1985
Accidents involving HGVs	471	391	337	296	323	335	300	352	325
All other accidents	2238	2129	2232	1982	1964	2108	1719	1845	1736
Total	2709	2520	2569	2278	2287	2443	2019	2197	2061

just bottom out and absorb little energy. Similarly, if it is too stiff (for a small weak car) it will not yield and again will not absorb much energy.

One effective method of increasing the amount of possible energy absorption would be to bring the face of the guard forward of the truck front. An increase in stroke, even as little as 200 mm, introduced in this way would make a substantial difference to the energy absorbed. Changes in the measurement of vehicle length for legislative purposes would be needed to allow the many vehicles already operating at maximum length to make use of this concept.

The limiting stroke and force levels that can be used at present suggest that impacts between small cars and trucks provide the important design case for the required truck structure.

Previous TRRL work on front underrun guards

The TRRL research on this subject which was presented at the 1985 Experimental Safety Vehicle Conference[2], was based on a series of test impacts between the fronts of cars of various sizes and the front of a truck fitted with an energy absorbing front underrun guard. In these tests the truck was stationary and the car was travelling into it at about 65 km/h. The guard used was one developed jointly between TRRL and TI Tube Products Ltd and its energy absorption is based on the plastic deformation of mild steel in the form of two invertubes. The guard may be seen in Figure 1. Each invertube collapses at a near constant load of just under 50 kN over a stroke of 200 mm giving a total energy absorption of about 20 kJ. Because of the geometry of the guard it needs a horizontal bumper force of about 70 kN to start collapsing which rises to about 130 kN at the end of its stroke[3]. The ground clearance of the guard in all tests was 300 mm.

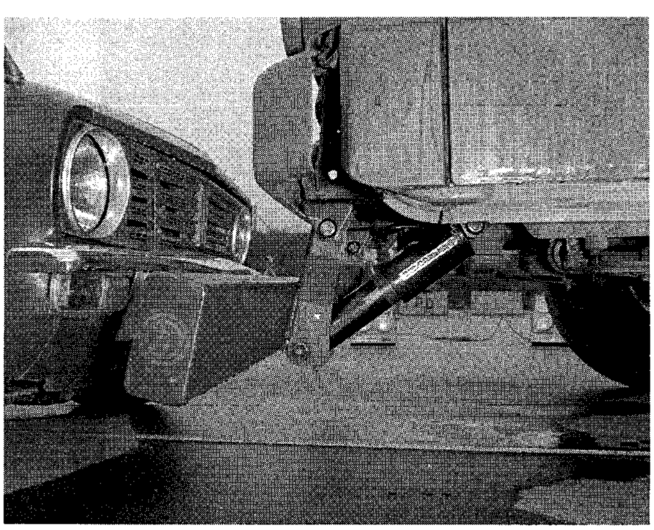


Figure 1. Invertube front underrun guard

The results from this earlier programme of work suggested that energy absorbing truck guards should offer protection to seat belted car occupants at closing speeds up to 65 km/h. It was also considered that ground clearance should not be much greater than 300 mm and certainly no more than 400 mm, otherwise the structural parts of smaller cars would be overridden.

It was estimated in Reference 2 that with this order of protection it should be possible to prevent at least 60 car occupant deaths each year in Great Britain.

Objectives of current work

All the earlier impact tests have been carried out with the truck stationary. In a study of fatal road accidents involving trucks[4] it was observed that when the truck was moving forward at the time of initial impact, it tended to ride up onto the car as the latter was pushed backwards. At the end of the main impact the truck then settled down onto the car causing further distortion of the passenger compartment.

Therefore one objective of the present research has been to investigate the effect on the protection offered to the car occupant when the truck is moving. To this end, impacts have been carried out between trucks and small cars with both vehicles moving towards each other at the closing speeds that were used in the earlier programme of work. The tests have been carried out with trucks both fitted and not fitted with a front guard.

The second objective is concerned with the capability of a truck front underrun guard to absorb energy. If legislation is introduced to require trucks to be fitted with front guards, it is essential, in the authors' opinion, that they should be capable of absorbing energy.

Because some designs of energy absorbing guards may be speed sensitive (for example, hydraulic systems) the test should be dynamic and at a speed representative of severe but survivable impacts. The test procedure developed is similar to that proposed in Reference 2, using a trolley impactor. Three different types of energy absorbing guard have been used in the tests.

Moving Car to Moving Truck Impact Tests

Method of testing

Two tests were performed with both the truck and the car moving at the same speed. A steel cable was attached to the rear of the lorry and, via a pulley fixed to the ground behind it, to the front of the car. Thus the car was pulled towards the truck as the truck moved forward, resulting in both vehicles moving towards each other at the same speed. The truck was driven by an experienced stunt driver: the car contained a dummy driver and passenger.

The intended position of impact of the car was into the centre of the truck guard and at right angles to it. To facilitate this, the left side front and rear wheels of the car were placed in a guide rail fixed to the ground. Just before impact with the truck the car left the guide rail and the towing cable was disconnected by means of a bomb release. The car was therefore free running when impact took place. Both the car and the truck speeds were measured just before impact.

In these tests the lorry had an all up mass of 5300 kg, and the cars used were British Leyland Minis, each of total mass 750 kg. Anthropometric dummies were placed in the car front seat positions and were restrained by three point inertia reel seat belts. No accelerometers were placed in the dummies. However one was placed at the base of each of the two 'B'-pillars of the car. The seat belt loads across the chests of the dummies were also measured. The deceleration of the lorry chassis was measured and each collision was filmed with high speed cameras running at approximately 400 frames per second.

The intended speed was 32 km/h per vehicle to give a closing speed of 64 km/h which had been used in the moving car to stationary truck tests.

The first impact was carried out with no underrun guard fitted. The test was then repeated with the truck fitted with a TI Tube Products invertube energy absorbing guard described earlier in the introduction and shown in Figure 1.

Results from the test with no truck guard fitted

In this test the height of the standard bumper of the truck was 580 mm from the ground. The impact closing speed was 64 km/h. The deceleration traces for each 'B'-pillar were found to be similar as regards general shape and maximum values. One of the two traces is shown in Figure 2, together with a note of important events at the times they happened. The latter were obtained by matching the deceleration-time trace with the events observed on the high speed film.

The peak deceleration of the car was found to be 38 g. This peak occurred when the standard bumper of the lorry made contact with the base of the 'A'-posts of the car (see Figures 2 and 4). The maximum seat belt load was found to be 2.47 kN. However this low value was due to the truck bumper reaching the windscreen of the car and penetrating the occupant compartment, thus causing contact between the upper parts of the dummies (including their heads) and the truck bumper and reducing forward movement and belt forces. In such a case, in spite of the restraining

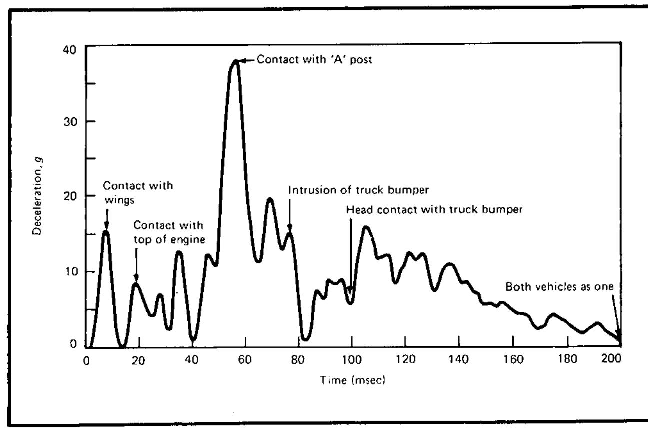


Figure 2. Car 'B' pillar deceleration pulse for test with no guard fitted

action of the seat belts, there would be little hope of saving the front seat occupants.

The length of the vehicle crushed by the truck was 1.32 m (43 per cent of the total length of the Mini). The damage to the car may be seen in Figure 4.

Results from the test with a truck guard fitted.

This test was undertaken with a TI Tube Products invertube front underrun guard fitted to the truck. The lowest edge of the bumper was 300 mm from the ground as can be seen from Figure 1.

The impact closing speed in this case was 62 km/h. One of the 'B'-pillar deceleration traces for the car is shown in Figure 3. The peak deceleration was found to be 33 g. From the high speed film analysis this peak occurred when the bumper made contact with the engine block as noted on Figure 3.

The peak belt load recorded was 5.56 kN which is considered to be a survivable belt load.

The maximum amount of crush was 0.65 m (21 per cent of the original length of the Mini) and this gave

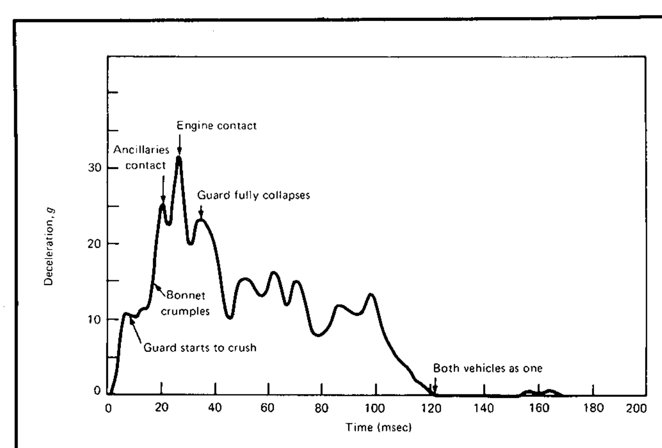


Figure 3. Car 'B' pillar deceleration pulse for test with guard fitted

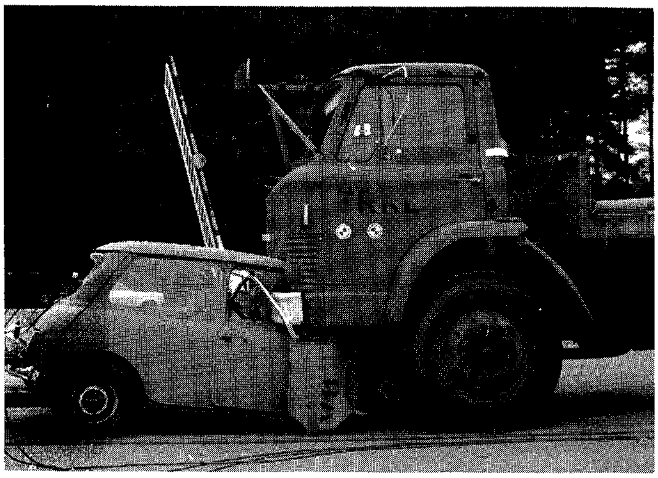


Figure 4. Impact tests without truck guard fitted. Both vehicles moving before impact

no intrusion into the occupant compartment by the truck front. There was however slight deformation of the car footwell and at the centre of the car facia.

The truck guard invertubes collapsed fully in this test. Also, bending of the bumper beam occurred which suggests that at least 20 kJ of energy was absorbed by the guard system. The results from this impact can be seen in Figure 5.

Comparison With Moving Car to Stationary Truck Tests.

Provided the two vehicles remain together after impact, the decelerations and resulting damage to the car should be dependent only on the closing speed of the vehicles. It should not matter whether the truck is moving before impact.

Comparisons of energy changes

The energy changes that occurred in the four impact tests, two with both vehicles moving and two

Figure 5. Impact test with truck guard fitted. Both vehicles moving before impact

with the truck stationary, are shown in Table 2. Energy change here is the difference between the total energy of the vehicles just before impact and that remaining immediately after impact is completed. The theoretical speeds after impact have been calculated using conservation of momentum theory. In general, there is good agreement between theory and practice for these speeds after impact.

There is also generally good agreement between the energy changes that occurred in the impacts where both vehicles were moving before impact and in the impacts where the truck was initially stationary. The exception is the test without a guard fitted and truck stationary, where the initial speed of the car was greater than intended.

Figures 6 and 7 show the two impacts, one without and one with the guard fitted, for the stationary truck tests. These figures are directly comparable with Figures 4 and 5 and generally show very similar results from the two methods of testing.

Comparison of impacts where no truck guard was fitted

In comparing the two types of testing procedure in detail, the more important differences occur in the pair of tests where the truck guard is not fitted.

Firstly, the amount of car crush in the stationary truck test was 1.17 m, while that in the moving truck test was 1.30 m. It can be seen from these results that there is a substantial difference between the two,

Table 2. Vehicle speeds and energy changes in impacts. Comparison of actual and theoretical results

Impact test		Speed* before impact m/s		Speed after impact m/s		Change in kinetic energy
Both vehicles moving		Truck	Car	Truck	Car	kJ
Without guard	Actual Test	- 8.9	+ 8.9	- 7.0	- 7.0	91
	Theory.	- 8.9	+ 8.9	- 6.7	- 6.7	104
With guard	Actual test	- 8.6	+ 8.6	- 6.6	- 6.6	92
	Theory	- 8.6	+ 8.6	- 6.5	- 6.5	97
Truck stationary						
Without guard	Actual test	0	+20.3	+ 2.4	+ 2.4	137**
	Theory	0	+20.3	+ 2.5	+ 2.5	135**
With guard	Actual test	0	+17.8	+ 2.7	+ 2.7	97
	Theory	0	+17.8	+ 2.2	+ 2.2	104

* Positive speed is in the original direction of motion of the car.
**Comparatively larger energy change due to greater closing speed.

although both amounts of crush mean that the intrusion into the car has extended to well behind the 'A'-posts.

The reason for the greater crush in the moving truck test can be seen in the high speed film. During the impact the truck bumper follows a horizontal path, contact occurs between the top of the engine and the bumper, causing the engine and subframe to twist so that the top of the engine pivots towards the rear of the car. This causes the car to sag downwards until the bottom of the 'A'-posts and sills touch the ground. At this point the truck meets a resistance and cannot crush downwards any more. However, after the main impact the truck still has forward momentum and therefore overrides the structure of the car, thus driving the front of the truck up and into the passenger compartment of the car.

This 'jacking' effect was observed in investigations of road accidents and is described in Reference 4.

Seat belt loads are not available for the stationary truck test but as in the test with both vehicles moving, contact was made between the dummies and the front of the truck. In such circumstances, as stated before, seat belt loads are largely irrelevant for survival of the car occupants.

The peak deceleration in the stationary truck test was 45 g for the car compared with 38 g obtained when both vehicles were moving. Bearing in mind that the closing speed of the former test was higher, these values are in good agreement.

The pitching motion of the car during the impacts, as observed in the film, varied slightly between the two tests. With the truck stationary the only pitching of the Mini is as the front gets wedged underneath the truck. The same situation occurs when both vehicles are moving. However in this case the truck appears to have a greater vertical displacement (that is riding over), and as has already been mentioned the amount of crush on the car was greater.

Comparison of impacts where a truck guard was fitted

With the underrun guard fitted the amount of crush of the car in the stationary truck test was 0.60 m, while that with both vehicles moving was 0.65 m, giving close agreement for the two test methods.

A maximum seat belt load of 5.10 kN was measured in the stationary truck test compared with 5.56 kN when both vehicles were moving. Both these loads are survivable and are meaningful results since there was no contact between the dummies and the truck structure as occurred in the tests with no underrun guard fitted.

A peak deceleration of 40 g was obtained in the stationary truck test compared with 33 g with both vehicles moving. The small differences in speeds of impact and amount of average crush of the car would account for some of the variation in the peak decelerations, but not all.

There were again some differences in the pitching motion of the car during and immediately after impact but the effect on recorded decelerations would be small. From the high speed film it can be seen that with the underrun guard fitted and the truck stationary then during the impact the car itself remains almost horizontal thus giving true deceleration values. However, immediately after the impact the truck is moved rearwards by 2 m, and the car itself takes up severe pitching motion with the rear wheels being lifted approximately 0.5 m off the ground.

When both vehicles are moving it can be seen that during the impact, the car pitches more than with the stationary truck. However after the main impact the car appears to start to pitch, as in the stationary truck

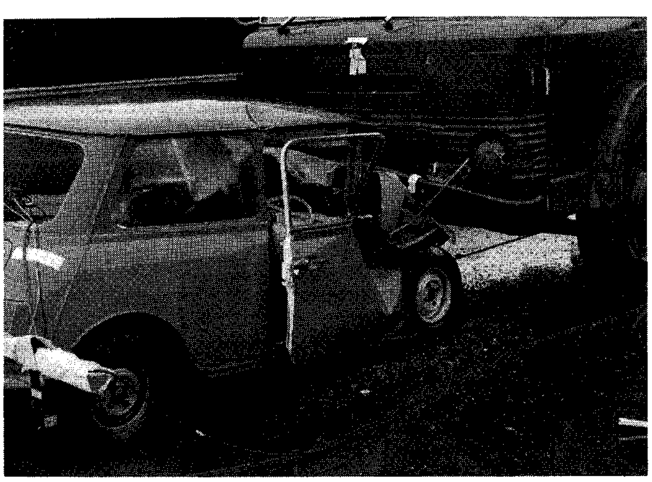


Figure 6. Impact test without truck guard fitted. Truck stationary before impact

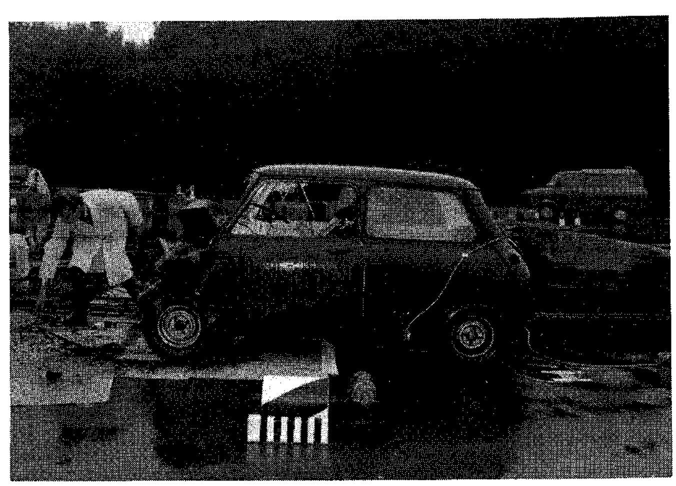


Figure 7. Impact test with truck guard fitted. Truck stationary before impact

case, but the car then becomes trapped between the underrun bumper and the standard bumper and thus remains reasonably horizontal.

Summary of comparisons

Table 3 is a summary of closing speeds, the amounts of car crush, peak decelerations at the car 'B' pillar and seat belt loads in the four impact tests.

The conclusions to be drawn from these tests are as follows:

1. The tests suggest that a truck front underrun guard with a ground clearance of 300 mm and with a capability of absorbing about 20 kJ of energy provides protection from fatal or severe injuries to seat belted occupants in a European small car at closing speeds up to about 65 km/h.
2. The energy change in an impact between a truck and a car is largely dependent only on closing speed, irrespective of whether or not the truck is moving before impact.
3. When no truck guard is fitted, the moving truck to moving car impact test gives a more realistic indication of the dangers of underrun, particularly regarding the degree of car crush.
4. When a truck guard is fitted there is little difference in the resulting car crush, decelerations or dummy seat belt loads whichever method of test is used. Because the test procedure with the truck stationary before impact is much simpler and cheaper to perform it is recommended as a satisfactory method of developing truck front underrun guards and as a means to demonstrate their benefits in use.

Trolley Test to Measure the Energy Absorption of Front Underrun Guards

It has been stated earlier in the paper that in the opinion of the authors it is essential that front underrun guards should absorb energy in order to raise the maximum survivable closing speed for car occupants. The tests carried out using one type of truck guard have shown that it is feasible to incorporate at least 20 kJ of controlled energy absorption into the guard.

Table 3. Summary of Mini car performance in impact tests.

Impact test		Closing speed m/s	Peak deceleration g	Car crush m	Peak seat belt load kN
Both vehicles moving	Without guard	17.8	38	1.30	2.47
	With guard	17.2	33	0.65	5.56
Truck stationary, car moving	Without guard	20.3	45	1.17	Not available
	With guard	17.8	40	0.60	5.10

This section of the paper describes the development of a dynamic test procedure to measure the energy absorbed by front underrun guards. To encourage future guards to have more energy absorption, an input energy change of 25 kJ has been used.

The aims of the test are to monitor the force levels and horizontal stroke of the guard to ensure that the energy absorption is used satisfactorily in impacts between small cars and trucks.

The test procedure

The mobile impactor or trolley is shown in Figure 8. It is basically a 6 mm thick piece of square section steel tube which measures 200 mm by 200 mm and is approximately 2 m long. It runs on wheels and has extra mass bolted to it to bring its total mass up to 250 kg. The impact face is a 10 mm thick piece of steel which is 200 mm high by 400 mm wide and has a 50 mm thick piece of plywood bolted to it. This in effect forms a non-deformable impactor. The impact face on an earlier version of the trolley was 200 mm square, to represent a car engine block. However, in a trial impact into the centre of an underrun bumper beam, it caused considerably more bending of the beam of the guard compared with that occurring in a similar impact with a car.

The underrun guard to be tested was mounted, in accordance with the manufacturer's instructions, to the rear of a truck of similar mass to the one used for the car impact tests. Thus any limitations in the effectiveness of the guard by the truck itself should be demonstrated.

The trolley was propelled into the guard at a speed of about 50 km/h to give the required energy change of 25 kJ. The method of doing this was similar to that described earlier for the moving car to moving truck tests except that the trolley was towed up to

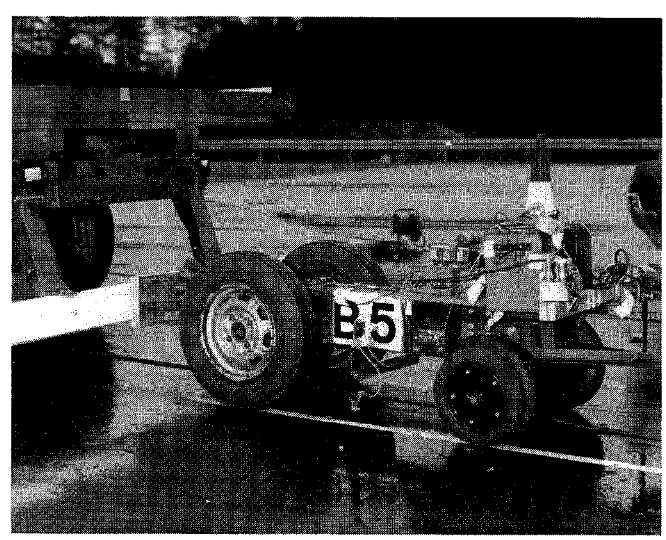


Figure 8. Test trolley

speed by a powerful car using a towing cable which passed beneath the truck. Points of impact were into the centre of the guard bumper beam and also in line with one of the drop arms. It was arranged for these tests that the trolley body centre line was at the same height from the ground as the middle of the bumper beam. As in the case of the car tests, the trolley was free running at impact.

The speed of the trolley was measured just before impact and transducers were fitted, one each side of the body of the trolley, to measure deceleration. The tests were also filmed by high speed cine cameras.

The underrun guards tested

Three commercially available guards, normally fitted to trucks to satisfy British requirements for rear underrun protection, were tested. Each has built-in energy absorption working on different principles. They all employ similar structures which consist of a cross-beam welded to two vertical drop arms. The drop arms are pivoted from brackets which are attached to the truck's chassis. The cross-beam or bumper bar is a rectanglar hollow steel section 150 mm by 75 mm with 6 mm wall thickness. The energy absorbing mechanisms used in each guard are described below.

The TI Tube Products 'Rearguard', shown again in Figure 9(a), has been described briefly in the introduction to this paper but in more detail, it works as follows:

It uses two invertube cartridges 450 mm long and capable of being compressed by 200 mm. These join the lower ends of the drop arms to brackets mounted on the truck chassis approximately 300 mm forward of the drop arm pivot brackets. In the event of an impact the drop arms swing forwards, causing the invertube cartridges to operate when the compressive force in either reaches about 50 kN. The function of these cartridges is to absorb the energy of an impact by turning a steel tube 'inside out'. The system may be tuned by changing the dimensions of the invertube cartridge to enable it to absorb the required amount of energy. In the form tested the pair of cartridges could absorb about 20 kJ of energy. Additional energy may be absorbed by bending of the bumper beam in severe impacts.

In the Quinton Hazell 'Underider' guard shown in Figure 9(b) the cartridges used operate on a different principle. Instead of using invertubes, the cartridges contain a special hydraulic oil. When the cartridges stroke in an impact, the oil is forced through a profiled slot in the cylinder wall and a resisting force is produced which is dependent on the speed of impact. Thus its energy absorption increases with increasing speeds. Externally the cartridges have strong coil springs which are designed to return the cartridges (and thus the drop arms and the cross-beam), back to their original state after an impact. Tests on this guard were carried out with two pairs of cartridges having differing internal profiled slots.

Finally, the 'CURB' underrun guard, made by Lostock Hall Fabrications Ltd, is shown in Figure 9(c). It does not use impact absorbing cartridges as in the two previously described systems. Instead, at the top of each drop arm there is a butyl rubber block placed between the arm and the bracket that it pivots from. In an impact the drop arms pivot and meet resistance from the rubber. Because of the type of rubber used, energy is absorbed. After impact the guard is designed to return to its original position.

Trolley test results

A total of eight impact tests were performed. Complete results are not available for two of the guards. Filming was not possible due to weather conditions for the centre impact into the invertube guard and there was insufficient time to carry out the CURB centre impact.

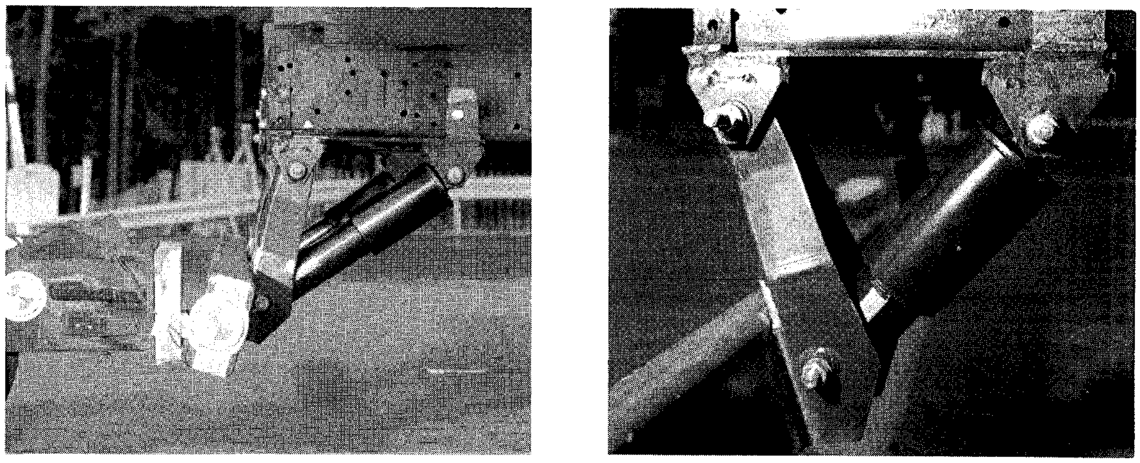
The full results for the remaining six tests are given in Table 4. The force during impact was derived from the mass of the trolley times its deceleration. Trolley displacement relative to the truck during impact was measured from the film and the total energy absorbed by the guards was determined from the area beneath the force displacement curves.

Overall, the energy absorbed by the guards increased with increase in maximum displacement. The largest displacement occurred in test B6, into the drop arm of one of the hydraulic guards but was caused mainly by a partial failure of one of the mounting brackets.

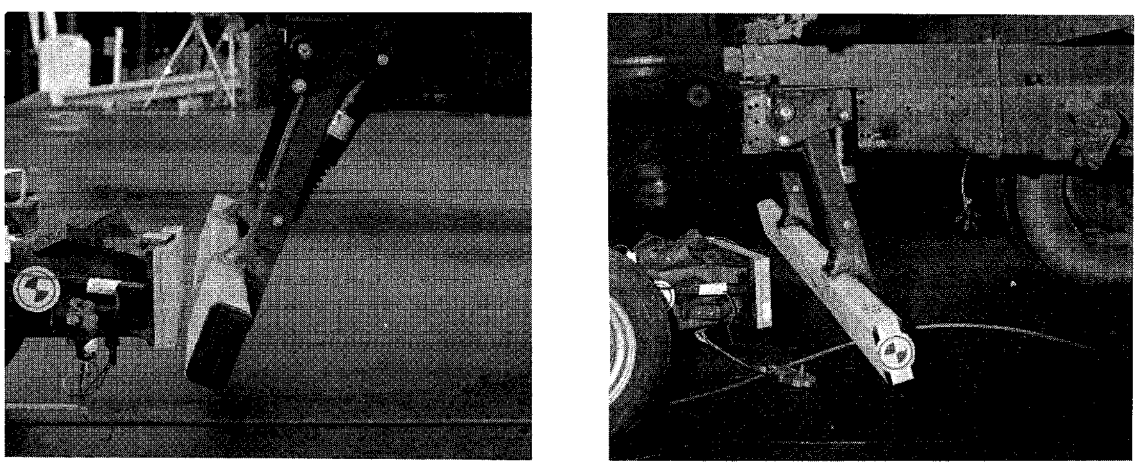
The peak forces varied with the type of guard. The Tube Products invertube guard and the CURB produced the largest forces. However, in the impacts into the Quinton Hazell guards with modified hydraulic units, the forces increased nearly to the same level.

It is only in the tests with the hydraulic guards that a full comparison may be made between impacts in line with a drop arm and into the centre of the bumper beam. In both pairs of tests (one pair with modified hydraulic units) the force levels in the centre impacts were about 15 per cent greater. This would indicate that even in asymmetric impacts, work is being done by both energy absorbing cartridges. This was also found in the impact in line with the drop arm of the invertube guard, where the stroke of the cartridge at the drop arm away from the trolley impact point was at least half that of the other unit, showing that it had done work.

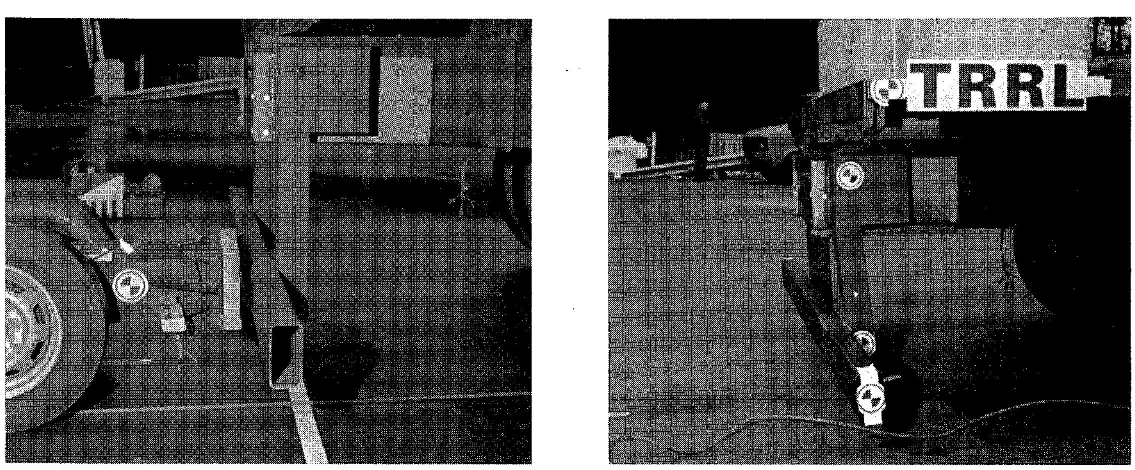
In both the centre impacts carried out the bumper beam deformed, thus absorbing some energy. The amount of bending is less than that produced in the impact between the small car and the invertube guard but bearing in mind the higher impact speed and energy in the car impact, the dimensions of the trolley impacting face would appear to be a satisfactory compromise. Virtually no beam bending occurred in any of the impacts in line with the drop arms.



(a) Invertube energy absorber



(b) Hydraulic energy absorber



(c) Rubber energy absorber

Figure 9. Three types of underrun guard before and after impact

Table 4. Trolley test results

Test number	Point of impact	Type of guard	Speed of impact m/s	Energy absorbed by guard kJ	Maximum force during impact kN	Maximum displacement of front of trolley ** m
B1	Drop arm	Tube Products Invertube	13.7	10.9	123	0.19
B4	Drop arm	Quinton Hazell Hydraulic	14.0	16.8	64	0.40
B5	Centre	Quinton Hazell Hydraulic	14.7	18.0	74	0.47
B6*	Drop arm	Quinton Hazell Hydraulic	14.8	19.1	90	0.53
B7*	Centre	Quinton Hazell Hydraulic	14.2	16.2	111	0.28
B8	Drop arm	CURB Rubber	13.6	14.3	121	0.29

* Different profiled slot in hydraulic units compared with those used in tests B4 and B5.
** Relative to truck

The results of the impacts in line with the drop arms, for each type of guard, may be seen in Figures 9(a), (b) and (c). Their force displacement curves are shown in Figure 10. The shapes of the curves are different for each guard. The force in the invertube guard increases with stroke up to a maximum of just over 120 kN to give a total energy absorption of about 11 kJ. The hydraulic guard force rises to a lower maximum of just over 60 kN but maintains that level of force over considerably longer stroke to absorb about 17 kJ of energy. In the CURB guard result, the force level is quite low over the initial 0.15 m of stroke. The analysis of the film suggests that these low forces occur while the rubber insert is compressing. The force then rises rapidly as the drop arms deform and the stroke increases to 0.29 m. The total energy absorbed was about 14 kJ but most of

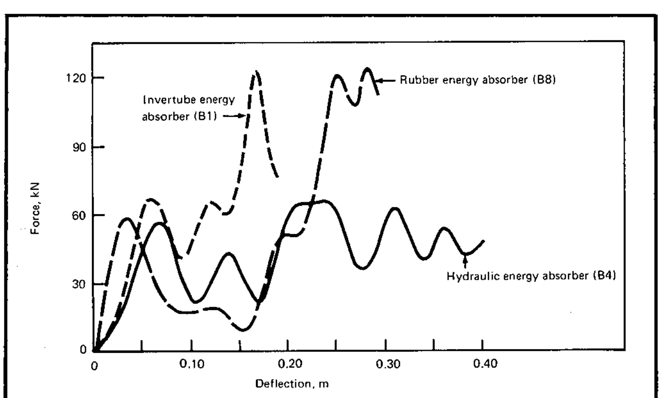


Figure 10. Dynamic force-deflection curves for three types of energy absorbing underrun guard

this was dissipated in drop arm bending (see also Figure 9(c)).

It is interesting that in the six tests, none of the guards absorbed more than about three-quarters of the trolley energy change. The lowest absorption was obtained with the invertube guard when around half was absorbed. In this test neither of the invertube cartridges stroked fully. However, good agreement was obtained between the energy absorption calculated from the linear compression of the cartridges and that measured from the force displacement curve. It does, therefore, appear to be a genuine result.

Detailed analysis of the deceleration traces and high speed film did not yield any obvious paths for the remaining energy. Trolley rebound velocities were very low in all cases and truck movements during impact were also negligible. It is possible that energy was absorbed by the temporary bending of various components of the truck such as chassis members and the cab structure and in wind up of the vehicle suspension.

Implications for future legislation

If underrun protection at the front of trucks becomes mandatory in the future, it is suggested that legislative requirements for two aspects of its performance would be necessary. The first should be concerned with its energy absorbing properties and the second with its ultimate strength.

Energy absorption. Considering first the energy absorption requirements, a dynamic test should be specified similar to the trolley test described in this paper. The test speed should be at least 50 km/h and it is suggested that the energy input should be at least 25 kJ. Limits should be placed on the force levels and maximum stroke during the impact. As stated earlier the force level should lie within a range which will allow the frontal structure of a small car to crush. In the car to truck impacts described in this paper maximum forces of between 240 kN and 300 kN (33 to 40 g) were obtained when the trucks were fitted with one type of guard.

Relating these values to a 250 kg impactor would suggest that the trolley or impactor deceleration should be limited to a maximum of 100 g (equivalent to a force of 245 kN). It may also be advisable to have a lower limit, say 50 g, in the initial stages of collapse of the guard, for example in the first 100 mm. This would extend the benefits of the energy absorption properties of the guards to lower energy impacts between cars and trucks.

It is suggested that the maximum stroke of the guard in the proposed test should not exceed 400 mm. This should prevent the cab structure of the truck from reaching the passenger compartment of the car after allowing for the crush of the bonnet of a small car. The only way of increasing the allowable stroke would be by permitting the guard to project forward of the front of the truck.

Test impacts should be carried out both at the centre of the beam and in line with one of the guard drop arms. The latter test should ensure that the torsional strength of the bumper bar and its joints with the drop arms are sufficient to utilise most of the guard's total energy absorbing capability even in an offset collision. It may prove necessary to specify a third impact position, close to the outer end of the guard, to prevent weak bumper beam ends being used. These can allow the truck structure to strike the car passenger compartment in collisions close to the lateral extremities of the truck. The required guard strength in this area is not easy to specify. If the guard is too stiff it may tear into the footwell of a weak car. One test carried out at TRRL with a car impacting into the outer end of an invertube underrun guard produced reasonably satisfactory results. Although the beam end bent to some extent, and therefore absorbed some energy, it did not allow the upper structure of the truck to strike the car passenger compartment. Further work is needed in this area to finalise a requirement.

The top edge of the impactor face should be no more than 400 mm from the ground to ensure that the guard, when fitted to the truck, is low enough to be effective.

It may be questionable whether such a test procedure should be carried out with the underrun protection mounted on a truck because of possible damage to the vehicle. It does however have the important advantage that it is representative of how the guard will be used in practice on the road. For underrun protection which forms an integral part of the truck cab structure, it may be the only way to test it.

An alternative test method would be to mount the guard via a rigid subframe to a large concrete block but further research would be needed to compare the results from the two methods of test.

Ultimate strength. The guard or protection must be ultimately strong enough so that when it is struck by a large car it does not break off and allow the car to underrun the truck structure. It is obviously not justifiable to make the structure strong enough to survive a large car impact at very high speed since it is unlikely under such circumstances that even seat belted car occupants would survive.

In an earlier 64 km/h test impact at TRRL between a heavy car (a 200 series Volvo) and the front of a truck fitted with a Tube Products underrun guard, the guard survived the impact. Since this guard meets the European Community Directive EEC 79/490 (which stipulates maximum forces and deformations for rear underrun protection) in order to satisfy the British Construction and Use Regulation 49, it is suggested that this requirement would form an adequate criterion for ultimate strength for front underrun guards.

Testing other types of front underrun protection. In this paper the only type of protection that has been considered in depth has been the 'add-on' type of guard. It may, in the future, be possible to provide front underrun protection as an integral part of the cab structure or as an extension to it, in the same way that energy absorption is provided in a car frontal structure. This could prove to be both lighter and cheaper.

The testing of this type of protection for legislative purposes may provide some problems. It would almost certainly have to be tested 'on the truck'. Whether a trolley test would be satisfactory for measuring energy absorption performance is not easy to say without further research. Ultimately the true test of its effectiveness would be demonstrated only by impacts using a car or car sized mobile barrier having a deformable face.

Conclusions

Car impact tests to demonstrate the effectiveness of truck front underrun guards

1. A truck front underrun guard with a ground clearance of 300 mm and capable of absorbing 20 kJ of energy can provide protection from fatal or severe injuries to seat belted occupants in a European small car at closing speeds up to about 65 km/h. (Equivalent, in this case, to a car velocity change of 57 km/h).
2. A test procedure, involving a small car impacting a truck fitted with a front underrun guard where both vehicles were moving towards each other before impact, gave similar results to those of an alternative procedure in which a similar car was impacted into a stationary truck at the same closing speed.
3. The stationary truck test is therefore a satisfactory method of developing front underrun guards and demonstrating their effectiveness.
4. The test without a front underrun guard fitted showed the possibility of additional crush of the car occurring when the truck has initial speed. This can be caused by the truck 'climbing over' the car after the basic impact is completed.

Trolley test to demonstrate the effectiveness of the energy absorption of front underrun guards or other underrun impact protection

A series of impact tests have been carried out at 50 km/h between a rigid faced trolley impactor of mass 250 kg and three different types of energy absorbing underrun guards fitted to a truck. The results suggest that the following procedure could form the basis for a legislative test to determine whether a front underrun guard has adequate energy absorption. The procedure might also be suitable for testing alternative forms of protection such as integral low front structures on trucks although this has not been verified.

1. The test should consist of three impacts. Two of them should be at a speed of 50 km/h and with an energy change of about 25 kJ between a rigid face impactor and the underrun guard. The face of the impactor should be approximately 400 mm wide by 200 mm high and its top edge should be no more than 400 mm above ground level. The first impact should be in line with the centre of the bumper beam and the second in line with one of the drop arms. Both these impacts should be perpendicular to the guard. A third impact into one end of the guard has not yet been finalised.
2. The deceleration of the impactor should be measured and, for the first two impacts proposed, should not exceed 100 g during the test. For the first 100 mm of guard movement the deceleration should not exceed 50 g. The total horizontal movement of the guard during impact should not exceed 400 mm.
3. To ensure sufficient ultimate strength, the front guard should satisfy European Community Directive EEC 79/490 (which is the Directive normally used to stipulate maximum forces and deformations for rear underrun protection.)
4. At present, it is considered that the impact tests should be carried out with the guard fitted to the truck.

Acknowledgements

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The Benefits of Energy Absorbing Structures to Reduce the Aggressivity of Heavy Trucks in Collisions

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Abstract

Because of the large weight differences that now exist between heavy trucks and cars, car occupants are at risk of serious injury when they collide with large trucks. However, although there is little that can be done to reduce this disparity in weight, it is possible to modify trucks so that the effects of the impact between a heavy truck and a car could be lessened. This paper estimates the effects of modifying the fronts of heavy trucks to incorporate crushable structures with stiffness characteristics similar to the fronts of cars. Equations of motion are developed that show that equipping trucks with crushable zones would increase the deceleration distance available to car occupants in car-truck collisions by 40 percent and

reduce the average deceleration to restrained occupants by a factor of 1.4. A method is provided that transposes this reduction in acceleration to a reduction in fatality risk using Fatal Accident Reporting System data for 1977-1985. An example is given that shows a crushable zone could reduce the likelihood of fatal injury to car occupants by as much as 33 percent.

Introduction

In 1985 there were nearly 40,000 fatal accidents in the United States and 4,211 or 11 percent of these involved heavy trucks (>26,000 lbs.). While attention has been given to improving the crashworthiness of cars to reduce the risk of serious or fatal injury, virtually nothing has been done to reduce the risk to other road users in collisions with large trucks. Seventy-nine percent of all fatal truck accidents involve other vehicles, and 42 percent involve a large truck and a passenger car (Figure 1). Truck-car collisions are the single most common type of fatal truck collision and the deaths are almost always to the car occupants.

Table 1 gives a breakdown of car occupant deaths in car-truck collisions by the impact area to the car and to the truck. Of the 2,187 car occupant deaths in crashes with large trucks in 1985, 29 percent occurred in frontal collisions, 32 percent occurred from the truck hitting the side of the car, 9 percent from the car hitting the side of the truck, 12 percent from the car hitting the rear of the truck, and 5 percent from the truck hitting the rear of the car. Engineering design improvements directed toward reducing the aggressiveness of trucks could reduce the severity of many of these crashes. For example, it has been established for some time that truck rear underride guards can reduce fatalities by effectively preventing cars from underriding trucks when cars strike the rear-ends of trucks. Crash protection devices on the sides of trucks similar to those for rear underride guards could also reduce car occupant fatalities in side impacts. Ways to protect car occupants in collisions where the front of the truck hits the front, side or rear of the car are less obvious, but, in theory at least, they can be developed. This paper concentrates on ways of improving crash protection for car occupants in collisions in which the fronts of trucks strike the fronts of the cars.

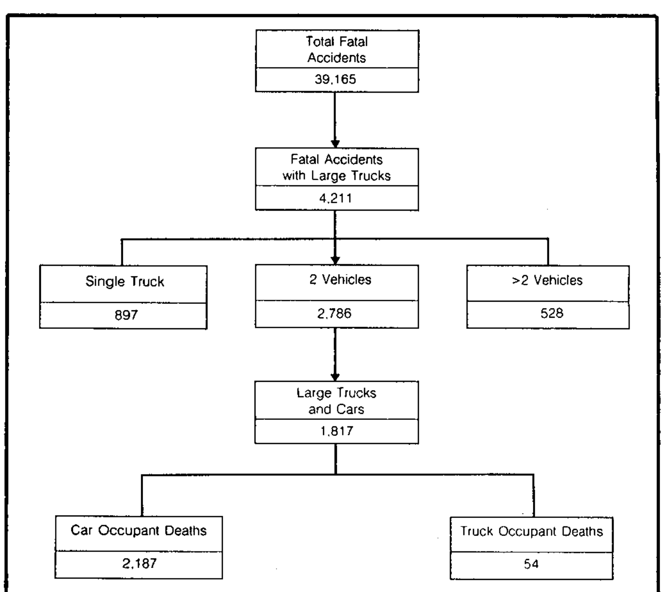

Figure 1. Distribution of fatal truck* crashes by number of vehicles involved—1985 FARS data

Table 1. Characteristics of car occupant deaths in car-truck crashes—1985 FARS data

Impact Point: Truck	Impact Point: Car	Car-truck Crashes with Car Occupant Fatality: N	Car-truck Crashes with Car Occupant Fatality: Percent
Front	Front	624	29
Front	Side	690	32
Side	Front	192	9
Rear	Front	263	12
Front	Rear	118	5
Other		280	13
Total		2,167	100

Theoretical Development

It has been claimed that, because of the large mass difference that now exists between heavy trucks and cars, for a given collision it is difficult to do anything that would reduce the velocity change that the car experiences.[1] Although little can be done to reduce the disparity in mass, it is possible to modify the fronts of trucks so that the injury potential of the impact between a heavy truck and a car could be lessened. The velocity change experienced by the car in a car-truck collision cannot be changed, but the distance and time over which it takes place can be increased by modifying the front of the truck. Increasing the time or distance of the velocity change has the effect of decreasing the deceleration experienced by the car occupants and consequently their risk of injury. This can be accomplished by putting a structure on the front of the truck with energy absorbing characteristics similar to those of a car.

Consider a symmetrical head-on (or central) collision between two vehicles as shown in Figure 2. The following notation is used:

E_1 (E_2) = energy absorbed by vehicle 1 (vehicle 2);

K_1 (K_2) = crush stiffness of vehicle 1 (vehicle 2);

m_1 (m_2) = mass of vehicle 1 (vehicle 2);

d_1 (d_2) = crush distance on vehicle 1 (vehicle 2);

F_1 (F_2) = force on vehicle 1 (vehicle 2; and

a_1 (a_2) = deceleration of vehicle 1 (vehicle 2).

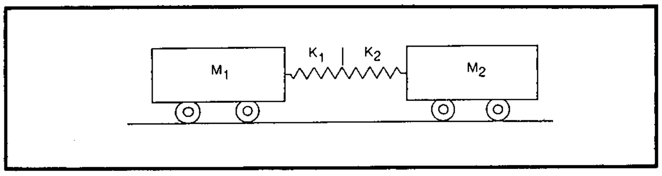

Figure 2. Central collision between two vehicles

During the collision the forces on each vehicle are equal and opposite so that $F_1 = F_2$. If it is assumed that the force on the vehicle is proportional to the vehicle crush, then

$$F_1 = K_1 d_1 = F_2 = K_2 d_2. \tag{1}$$

The energy absorbed in crushing vehicle 1 is given by

$$E_1 = K_1 d_1^2/2.$$

The energy absorbed in crushing vehicle 2 is given by

$$E_2 = K_2 d_2^2/2.$$

Then the total energy (E) absorbed during the collision is given by

$$\begin{aligned} E &= E_1 + E_2 \\ &= K_1 d_1^2/2 + K_2 d_2^2/2. \end{aligned}$$

In a truck-car collision, the crush on vehicle 2 (the truck) is zero and the total energy absorbed is given by

$$E = K_1 d_1^2/2.$$

Alternatively, the crush on vehicle 1 is

$$d_1 = \sqrt{2E/K_1}.$$

If the truck is modified so that its stiffness is equal to that of the car, the truck absorbs a proportion of the energy and the total energy absorbed is given by

$$E = K_1 d'^2_1/2 + K_2^2/2.$$

where d'_1 is the new crush of vehicle 1. However, Equation (1) shows that if the vehicles have equal stiffness the amount of crush on each vehicle will also be equal such that $d'_1 = d_2$ so that

$$E = K_1 (d'_1)^2. \tag{2}$$

Rearranging Equation (2), the crush d'_1 is given by

$$d'_1 = \sqrt{E/K_1}.$$

However, the total crush between the two vehicles is $2d'_1 = 2\sqrt{E/K_1}$.

Note that the energy E, absorbed in the collsion, is the same irrespective of whether the truck is modified. Also the velocity change, $\triangle V$, that the occupants of the car undergo is the same, but when the truck is modified the car occupants are decelerated over a longer distance so that their average deceleration is less. The ratio of the two deceleration distances is $2d'_1/d_1$. Then

$$2d'_1/d_1 = 2\sqrt{E/K_1}/\sqrt{2E/K_1} = \sqrt{2}.$$

Thus, if the truck is modified so that its stiffness is equal to that of the car, the occupants of the car derive about 40 percent more ride down, i.e., the distance in which they are decelerated to rest is increased by 40 percent. To determine what effect this increased ride down has on the likelihood of injury for the car occupants, the most straightforward approach is to see how their average deceleration is reduced. Assuming a force is proportional to crush relationship, the average force $\overline{F}$ on the car during the impact is given by

$$\overline{F} = \int_0^{d_f} Kxdx/d_f,$$

where x is the crush distance at any time during the impact, and d_f is the final crush distance. Then $\overline{F} = Kd_f/2$.

Comparing the average force on the car in the unmodified truck collision ($\overline{F}$) with that in the modified collision ($\overline{F}_m$) we have

$$\overline{F}_m/F = \sqrt{E/K_1}/\sqrt{2E/K_1} = 1/\sqrt{2} = 0.7. \tag{3}$$

Table 2. Number of car occupant and truck occupant fatalities in fatal truck crashes by size of car involved—FARS data 1977-1985

Car Mass-lbs.	Truck Occupant Fatalities	Car Occupant Fatalities	Ratio Car/Truck Fatalities	Mean Car Mass-lbs.
1,500-1,999	17	595	35.0	1,851
2,000-2,499	35	912	26.1	2,275
2,500-2,999	34	1,022	30.1	2,723
3,000-3,499	49	1,601	32.7	3,240
3,500-3,999	52	1,273	24.5	3,741
4,000-4,499	29	756	26.1	4,211
4,500-4,999	14	245	17.5	4,750

Regression analysis results: fatality ratio = -0.0043 (mass) + 41.42; r = - 0.8.

Table 3. Number of car occupant and truck occupant fatalities in fatal frontal car-truck crashes by size of car involved—FARS data 1977-1985

Car Mass-lbs.	Truck Occupant Fatalities	Car Occupant Fatalities	Ratio Car/Truck Fatalities	Mean Car Mass-lbs.
1,500-1,999	3	188	62.7	1,854
2,000-2,499	4	328	82.0	2,267
2,500-2,999	4	322	80.5	2,707
3,000-3,499	7	441	63.0	3,233
3,500-3,999	7	329	47.0	3,736
4,000-4,499	6	169	28.2	4,204
4,500-4,999	2	75	37.5	4,731

Regression analysis results: fatality ratio = -0.0163 (mass) + 110.1; r = -0.824.

However, because the mass of the vehicle is unchanged the ratio of modified versus unmodified average force also represents the change in the deceleration that the car occupants will experience providing they are restrained, i.e., modified deceleration = 0.7 (original deceleration).

Thus, equipping the fronts of trucks with a crush zone of stiffness equal to that of cars will reduce the deceleration that restrained car occupants experience by a factor of 1.4. Unrestrained car occupants will derive less benefit from the ride down.

Estimated Reduction in Fatality Risk From Modified Truck Front-End Design

If the deceleration of the occupant can be reduced by 1.4, injury severity will also be reduced particularly if the occupants are restrained. In effect, the increased crush space gives the occupant more distance in which to decelerate while the velocity change for the car remains the same. Another way to look at this reduction in deceleration is as an increase in the effective mass of the car.

This can be seen by rearranging Equation (3):

$$\overline{F}_m = \overline{F}/\sqrt{2} \text{ so that } \overline{F}_m/m = \overline{F}/m\sqrt{2}$$
$$\text{or } a' = F/m\sqrt{2}.$$

where a' is the reduced deceleration. Hence for the purposes of calculating the effect of the crushable zone on reducing the risk of fatal injury, the reduced deceleration is effectively achieved by increasing the mass of the car by a factor of $\sqrt{2}$, i.e., the effective mass $m' = m\sqrt{2}$.

There is evidence that the risk of fatality in car-truck collisions increases as the mass of the car decreases.[1] If this effect can be quantified, the reduction in fatalities that could be achieved by increasing the effective mass of the car could be assessed.

Using Fatal Accident Reporting System (FARS) data for years 1977 to 1985, Table 2 gives the ratio of car occupant to truck occupant fatalities for fatal accidents involving large trucks (>26,000 lbs.) analysed by subcategories of vehicle weight in increments of 500 lbs; the mean mass of the cars involved is given for each car weight category. Figure 3 shows the fatality ratio plotted against the mean car mass for each weight category. The increasing fatality risk with decreasing car size is clearly evident, and the strength of the relationship is confirmed by the regression analysis correlation coefficient of -0.8 ($p<0.001$). Because of the already large disparity in mass between cars and trucks, the increase in the number of car occupant fatalities cannot be attributed to the increase in the mass ratio. A primary cause for the increase in severity is that as cars are downsized

Table 4. Regression equation for fatal injury odds for car occupants in fatal head-on car-truck crashes

Independent Variable	Beta*	Std. Error	Chi-square	p value
Intercept	-4.810	0.187	664.0	0.0001
Car mass	6.8×10^{-4}	$2. \times 10^{-2}$	9.8	0.002
Accident year	0.079	0.070	1.3	0.26
Mass x year	-9×10^{-7}	8.2×10^{-5}	1.4	0.24

*$\log e\,(1-p/p) = B_0 + B_1/m + B_2\,y + B_3\,m/y$;
p = probability of fatal injury.

their crush stiffness is increased[2] and the amount of crush for a given velocity change is reduced. This means that the deceleration that an occupant experiences is increased with a corresponding increase in the risk of injury. Another reason for the increased risk is the higher likelihood that small cars will suffer occupant compartment intrusion. However, given the wide range of severities that occur, there will be a significant number of crashes that occur without compartment intrusion in which increased crush space would be beneficial.

To establish whether frontal crash protection would work, it is necessary to further restrict the analysis to collisions involving the front of the truck and the front of the car. Table 3 gives the ratio of car occupant to truck occupant fatalities as a function of car weight for head-on collisions involving the front of the car and the front of the truck. Figure 4 gives the car occupant to truck occupant fatality ratio plotted against mean car mass for each weight category. The correlation coefficient of -0.8 suggests a strong empirical relationship between the ratio of car occupant to truck occupant fatalities and mass car.

However, although there is a clear increase in the ratio of car occupant to truck occupant fatalities with decreasing mass, a number of possible confounding effects must be considered. Since the 1970s, there has been a progressive downsizing of cars and a steady annual increase in the ratio of car occupant to truck occupant fatalities.[1] This means there is the possibility of simultaneous effects of car mass and accident year. Also larger cars could produce more severe car-truck crashes, which would increase the likelihood of fatal

Table 5. Regression equation for fatal injury odds for car occupants given a truck occupant death in head-on car-truck crashes

Independent Variable	Beta	Std. Error	Chi-square	p value
Intercept	-0.082	0.235	0.12	0.73
Car mass	4.1×10^{-4}	3.0×10^{-3}	1.51	0.22
Accident year	0.086	0.086	0.59	0.32
Mass x year	-8.0×10^{-5}	0.01	0.49	0.48

injury to the truck driver. This would increase the denominator in the fatality ratio and produce an anomalous effect of increasing fatality ratio with decreasing car mass. One would intuitively expect the effect to be small because, at closing speeds of 60 mph, the increase in velocity change experienced by a 40,000 lb. truck in a frontal collision with a 2,000 lb. car compared to 4,000 lb. car is less than 1 mph.

To examine these effects, multiple logistic regression procedures were used.[4] To fit the regression models the CATMOD procedure from the SAS Institute was used.[5] The dependent variable in each observation was the presence or absence of a fatal injury, and the independent or predictor variables were car mass, year of accident, and the interaction of car mass with year of accident. Car mass was centered about its mean by recoding car mass as the difference between the mean mass and the mass of the accident involved car. Regression equations were used to predict the *log odds* of occupant fatality for the following situations: death of a car occupant given a fatal head-on car-truck accident; death of a car occupant given a fatal head-on car-truck accident in which the truck driver died; death of a truck occupant given a fatal head-on car-truck accident. The results are given in Tables 4 through 6.

The odds of fatality for car occupants (Table 4) increased significantly with decreasing car mass ($p<0.002$), they were not dependent on accident year, and the interaction of car mass and accident year was not significant. The odds of fatality for car occupants (Table 5), given a truck occupant death, is dependent neither on car mass nor accident year. This is most likely because accidents in which truck drivers are killed are so severe that their outcome is not dependent on the size of the car. This means that the denominator of the fatality ratio of car occupant to truck occupant death, in Figures 3 and 4, acts as a surrogate for truck exposure.

The odds of fatality for truck occupants (Table 6) also does not appear to depend on the mass of the car, although the parameter estimate for the mass effect does have the appropriate sign, i.e., truck occupant death increases with increasing car size. It is reasonable to conclude that the effect of increasing risk of car occupant death with decreasing car mass is predominantly a car mass effect and is not caused by anomalous effects of truck occupant deaths or accident year. However, considering the mass effects shown in Tables 4 and 6 simultaneously in terms of the ratio of car occupant deaths to truck occupant deaths, the two mass effects are additive. The mass effect in Table 4 shows that the probability of car occupant fatality decreases with increasing car mass, and the mass effect in Table 6 (although not statistically significant) shows an increase in the probability of fatality for truck occupants with increasing car mass. This means that, although the ratio of car to truck occupant fatalities (Table 3) is likely to be controlled predominantly by the car occupant/mass effect of the numerator, there could be some truck occupant/mass effect in the denominator. This would have the effect of reducing the ratio for large cars and increasing it for small cars.

Table 6. Regression equation for fatal injury odds for truck occupants given a car occupant death in head-on car-truck crashes

Independent Variable	Beta	Std. Error	Chi-square	p value
Intercept	4.645	0.165	793.4	0.0001
Car mass	-3.0×10^{-9}	0.02	3.6	0.06
Accident year	-0.01	0.063	0.0	0.87
Mass x year	5.0×10^{-5}	8.1×10^{-5}	0.4	0.53

The results in Figure 4 provide a means of estimating the reduction in fatality risk that could be achieved by equipping trucks with crushable zones. The parameter estimates for the regression analysis give the following equation:

$$\frac{\text{No. Car Occupant Fatalities}}{\text{No. Truck Occupant Fatalities}} = 110.1 - 0.0163 \text{ (mass of car).}$$

The mean mass of the car population in car-truck fatal accidents in 1985, was 2,921 lbs. Putting a crushable zone on all trucks would effectively increase this mean mass by a factor of $\sqrt{2}$ to 4,131 lbs. Substituting this value in the regression equation from Figure 4 gives a reduced fatality ratio of 42.8 compared to the original value of 62.5. This suggests that crushable zones would reduce the risk of fatality to car occupants in frontal car-truck collisions by 32 percent.

Discussion

The results of this analysis suggest that putting crushable zones on the fronts of trucks could reduce

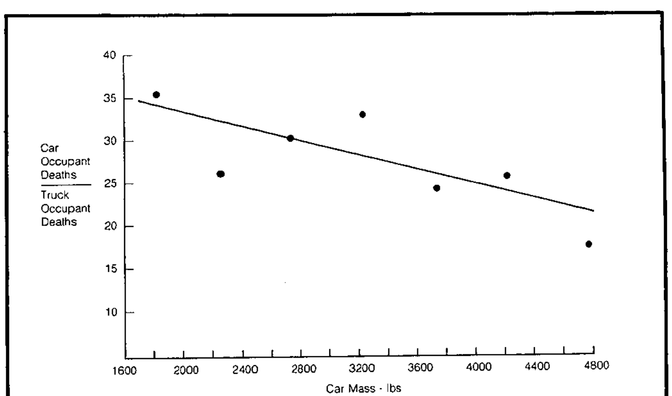

Figure 3. Ratio of car occupant to truck occupant deaths versus car mass—all crashes

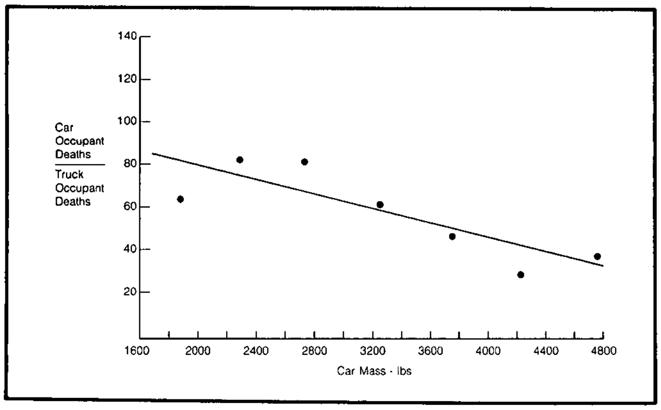

Figure 4. Ratio of car occupant to truck occupant deaths versus car mass—frontal crashes

the risk of fatality to car occupants in car-truck collisions by about one-third. Although the concept of crushable zones or front guards has received little attention in the United States, their feasibility has been assessed in Europe. A United Kingdom study estimated that about 25 percent of car occupants killed in car-truck collisions would have survived with a suitably constructed front energy absorbing guard.[6] A prototype guard was constructed with a crushable distance or stroke of 40 cm (15.7 in.). This was based on the fact that 50 to 60 cm is the maximum crush distance available in the smallest mini-cars before serious intrusion starts to occur. The characteristics of the prototype bumper allowed it to start to stroke at a load of 146 kN (approximately 10 tons). However, crash tests of a 1,000 kg car into a 5,100 kg truck at 50 km/h (angled offset) and 64 km/h (perpendicular offset) showed that the bumper did not stroke fully, although it proved effective in preventing underrun and passenger compartment intrusion. A central perpendicular test of a 1,550 kg car into the truck at 56 km/h also failed to stroke the bumper fully. To optimize the energy absorption the load required to stroke the bumper was reduced to 68 kN. The modified bumper stroked fully, in a central impact with a 1,000 kg car at 65 km/h (40 mph). This study clearly demonstrated that protection for car occupants at closing speeds of 65 km/h was possible providing occupant restraints are used. In the absence of passive restraints or if seat belts are not worn, protection cannot be provided for unrestrained occupants at relative speeds much above 40 km/h (25 mph).

A German study that looked at the possibilities of reducing the consequences of car-truck accidents found that more than half of car-truck fatalities occur to car occupants involved in frontal collisions with trucks.[7] The study concluded that, although the relative speeds in frontal collisions were high, reducing the agressitivity of the front end of the truck would also reduce the risks in truck collisions. In fact, if the results of the United Kingdom study (i.e., fatal collisions below 65 km/h are survivable with front energy absorbing guards) are applied to the German accident data, more than 40 percent of the fatal frontal collisions between cars and trucks would be affected.

The results of all three studies lead to the conclusion that energy absorbing bumpers on the fronts of trucks could provide a substantial saving of life. Over 2,000 accidents in the United States each year involve a car-truck collision in which there is fatal injury to the car occupant. Table 1 shows that in 1985, 624 of these collisions were frontal impacts so that 32 percent or about 200 fatal accidents per year could be ameliorated by energy absorbing bumpers.

Conclusions

1. If trucks were constructed with front energy absorbing guards so that their energy absorption characteristics were similar to that of cars, the severity of injuries to car occupants involved in car/truck accidents could be substantially reduced.
2. Energy absorbing zones on the fronts of trucks could increase the ride down distance available to car occupants by 40 percent and reduce their average deceleration by a factor of 1.4.
3. Energy absorbing zones on the fronts of trucks could reduce the risk of fatal injury in car-truck collisions by as much as 33 percent.

Acknowledgments

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The Global Approach for Safety in the V.I.R.A.G.E.S. Project

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Introduction

During the 10th ESV conference (Oxford) RENAULT V.I. presented its Research Programme V.I.R.A.G.E.S. (Industrial Vehicle Improving Energy Consumption and Safety) and the test results of a first stage experimental vehicle (VE 10).

This research programme directed by RENAULT V.I. has engaged also related industries such as Fruehauf and Trailer companies for the semi-trailer.

This programme is planned from 1982 to 1988 and is partially aided by French Administration :

Ministère des Transports
Agence Française pour la Maîtrise de l'Energie.

V.I.R.A.G.E.S. is a project with multicriteria objectives, which tends to satisfy new general specifications in all components :

for mobility and energy
for safety
for goods transportation
for comfort, ergonomy and live conditions on board
for environmental conditions
for industrial building.

The aim of this project is a new global economical optimum in the industrial field and for the performances.

We present here the main elements of the second experimental vehicle (VE 20) essentially on the SAFETY aspect.

V.I.R.A.G.E.S. VE 20 and Primary Safety

The unit has a total streamlining body and presents a non-aggressive outline.

The width and height conform with the new European regulation but its bigger length shows the necessity of a better adapted regulation to satisfy the use of new long trailer (13.4m) and to maintain a good level of comfort in the cab.

The maxi gross weight is 44 tonnes, and the unit conforms with the European turning circle, despite its important length (17.4m). For the tractor two possible solutions are in test—a 6 × 2 outline and a 6 × 4 one.

Handling—Roll-Over Comportment

The vehicle is a 6 axles unit—3 for the tractor and 3 for the semi-trailer—with new tires and wheels with little diameter (wheel 19″5; height under charge, 937 mm).

This choice, organised with the general architecture, and the level controlled hydraulic suspensions permit to have the gravity center 173 mm down and the roll axle up.

The use of single wheel on each side of axles makes the real gauge larger.

The direction control system is realised with a two-way hydraulic safety unit.

In association with a very front axle, all these elements permit a better handling of the vehicle particularly to avoid the roll-over situation.

Braking

The energy absorption for the modulation of the running speed is realized by a hydro-mechanical retarder installed between the engine and the gear box. It permits to eliminate 170 KW in a permanent retarding phase and 300 KW during short phases.

The braking is obtained by disc brakes with high pressure hydraulic control on the tractor and pneumatic control on the semi-trailer. There is a double control circuit on each axle, and an anti-slipping device on each wheel.

In the 6 × 2 version an original device derived of the use of hydropneumatic suspensions, permit the transient overloading of the motorised axle to increase the limit adherence for the starting phase.

This micro-process piloted device acts automatically and uses certain components of the anti-slipping system.

Visibility—Lights

During night the visibility of the unit is reinforced by a reflecting painting belt.

At the driving place, the visibility is very improved because of the front and lateral windshield which come down the level of the cab floor—the visibility distance is divided by 2 in relation of the actual cabs.

The lateral rear visibility is realised by classical large mirrors.

The Driving Place is an important component of the SAFETY

In the objective to obtain a large field of visibility the dash-board is dimensioned only for the minimum number of instruments asked by the regulation. The other informations for diagnostics or vehicle management (fuel consumption for example) can be obtained on a satellite display, only when the driver asks for them.

Less Splash and Spray

The streamlining of the whole vehicle is a good solution to minimize splash and spray. More an original box between each wheel collects water and pours it out, under the body, out of the track of the wheels.

An electronical device to survey the pressure of tires complete this set of systems improving in a full manner the different factors of active safety.

V.I.R.A.G.E.S. VE 20 and Passive Safety

In this area the specifications aim to lower seriously the consequences of a front-to-front shock between a personnal vehicle and VE 20.

In this type of collision, the priority is to obtain a good compatibility with the front structure of the two vehicles, avoiding the usual underrunning of the passenger car and permitting the structures of the car to work in the same manner as in a front-to-front, car-to-car shock.

This is obtained by a special front structure of VE 20. The planes of the strengths during the shock are compatible on the two vehicles. More, deformable elements of the front axle system are active between the bumper and the rigid body of the tractor.

Shock tests at a speed of 57 km/h of a car again the simulated fixed front parts of VE 20 have given similar results as the normalized shock of the car at speed 50 km/h on the European wall.

For the lateral and rear collisions the very low and rigid belt gives a better protection as the actual devices.

For the persons on board, a lattice structure of the cab, a very high position and a large inner space (9 m3) aim to obtain a minimisation of the importance of the injuries when the cab is concerned during the accident.

Safety and Use

When the vehicle is stopped for activities related with goods transportations, original solutions with the aim of SAFETY have been built.

The access of the cab is realised only on the right side of the vehicle by an inner stair-case, protected by sliding doors, to avoid down on the traffic side and put an end to the risks of sliding.

However, these choices claim for a long cab not possible with the actual European regulation and the use of long semi-trailers.

Tractor and semi-trailer are equipped with a system permitting the automatical coupling and uncoupling operations. This manoeuvre is initialized at the driving place. During the uncoupling operation, for example, down the stands, unlocking the king-pin, down the tractor suspension, putting the park brake, out electrical and pneumatical connections are automatically realized and the tractor is authorized to run, just only all necessary sequences are ended.

Conclusion

The experimental vehicle VE 20 is a technical compromise in regards the aimed multicriteria objectives.

It is not a commercial version, but the essential industrial and economical constraints have been taken in account to build a realistic unit.

Despite of these constraints, the vehicle presents in *all areas of the SAFETY*, solutions permitting important improvements but these gains are the results of a global approach, that is to say a new global design of this type of unit in opposition of margin partial usual responses.

More, these new solutions are often possible only in the perspective of new regulations better adapted for the goods road transportation of the years 2000.


AULT
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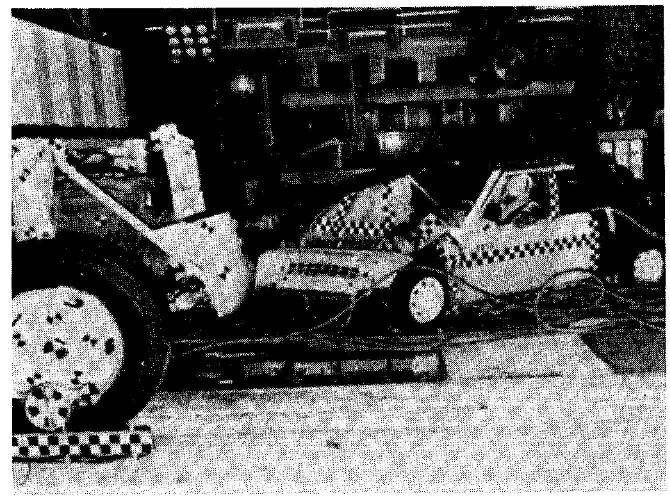

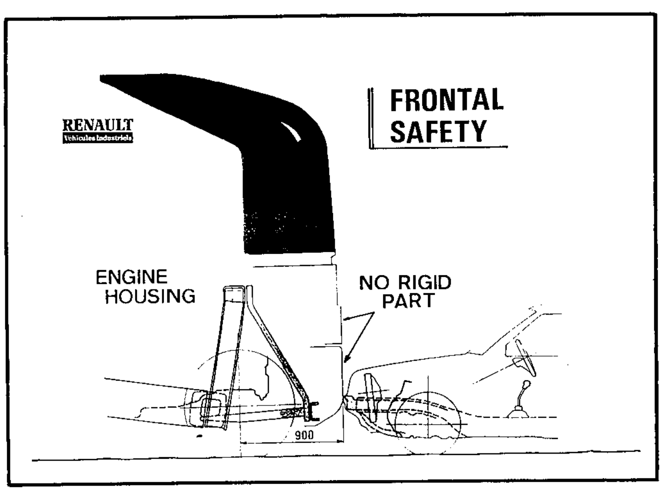
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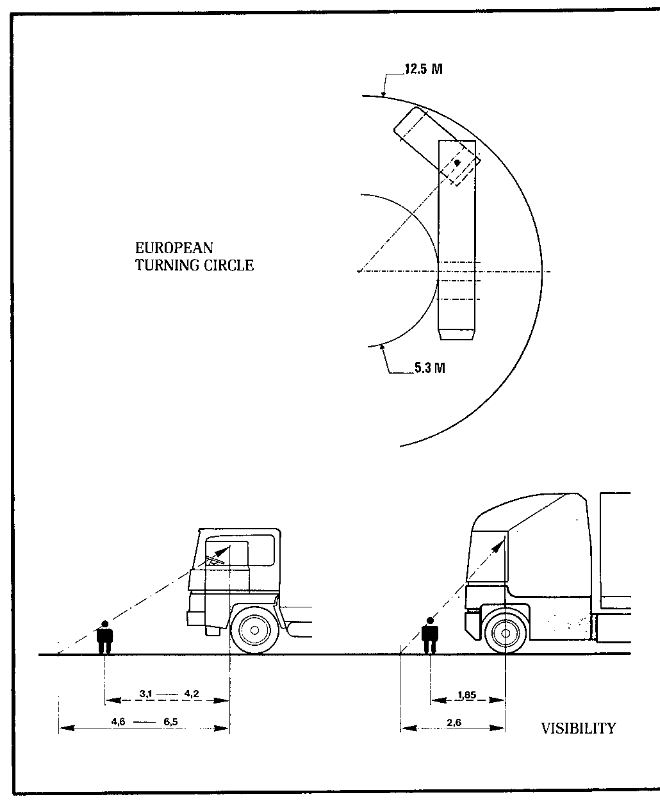
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1,85
2,6
VISIBILITY

Side and Rear Marking of Trucks With Passive Materials

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Abstract

Starting with an investigation about the distribution of the reflection factor of trucks in Europe and about the background luminances around the trucks during nighttime, the influence of different markings on the conspicuity is tested. Out of these, in the first step static experiments an optimal side and rear marking of trucks is developed.

Introduction

During nighttime driving, the trucks are marked not too conspicuous, so a lot of side- and rear-impacts can occur. In down-scaled experiments special markings of trucks should be developed to improve the conspicuity.

Luminances of Trucks During Nighttime

In Figure 1 the situation of recognition of trucks during nighttime driving is plotted schematically.

The truck is only seen in the contrast to the surrounding luminances L_H, L_R, L_L, L_S in the side situation, L_1, L_2, L_3 in the rear situation. The dotted area describes the luminance area necessary for the detection of a truck.

The influence of the rear position lamps is neglected in these experiments because this marking in Europe is the same for all vehicles.

In Figure 2 the test results for the threshold luminances for trucks seen from aside and from the rear are plotted. In addition the curves for seen luminances in the low beam situation are shown for the area for the chassis (L_3 in Figure 1) and body work (L_1 in Figure 1).

This Figure shows that for example the body work can be seen from the rear at a distance of 70 m, the chassis at 120 m. These values are for the threshold case without glare etc. In the normal traffic situation these distances are much smaller.

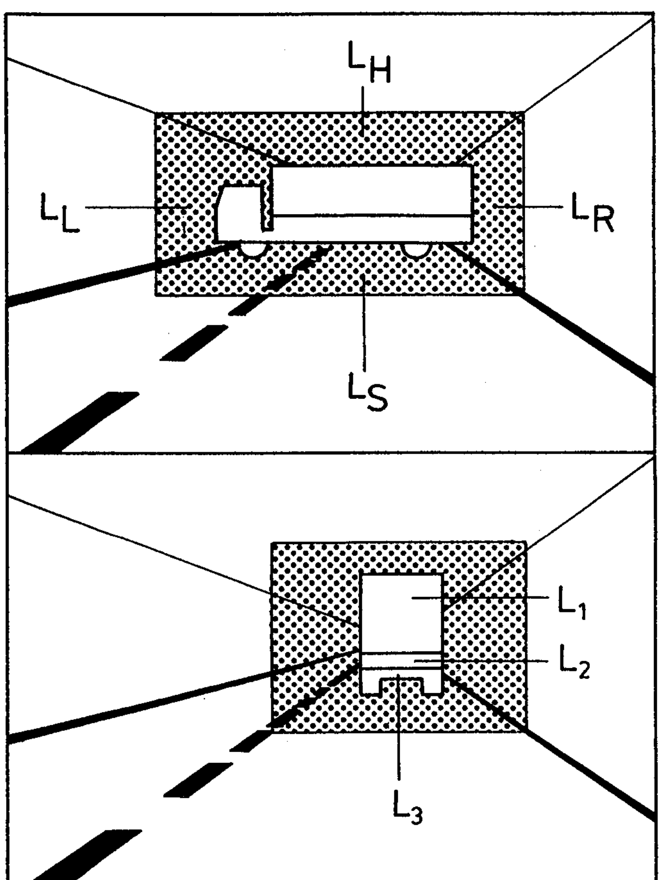

Figure 1. Side and rear side of a truck in the street L_R, L_L, L_H, L_S: background luminances L_1, L_2, L_3: background luminances

Side Marking of Trucks

In a 1:10 down scaled experiment the optimal side marking of trucks was investigated. In Figure 3 and 4 (annex) these 15 different markings are shown.

In an assessment experiment (9-step-scaling) the optimal luminance of the 15 different side markings was derived. The results are plotted in Figure 5.

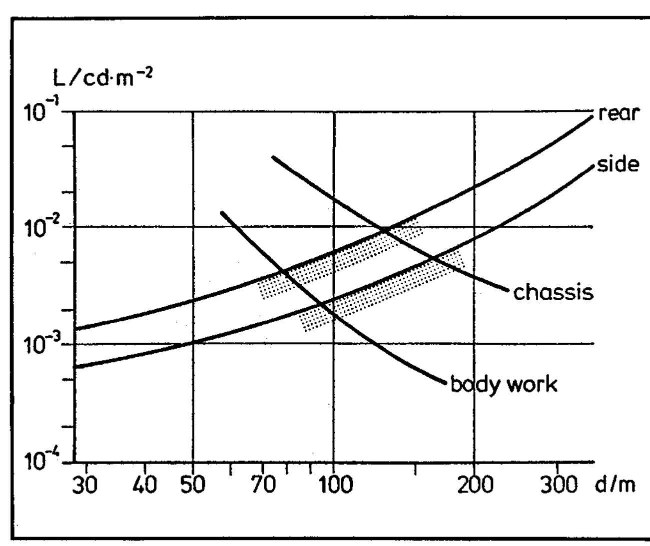

Figure 2. Threshold luminances of trucks seen from aside, chassis, body work: mean luminances of these areas

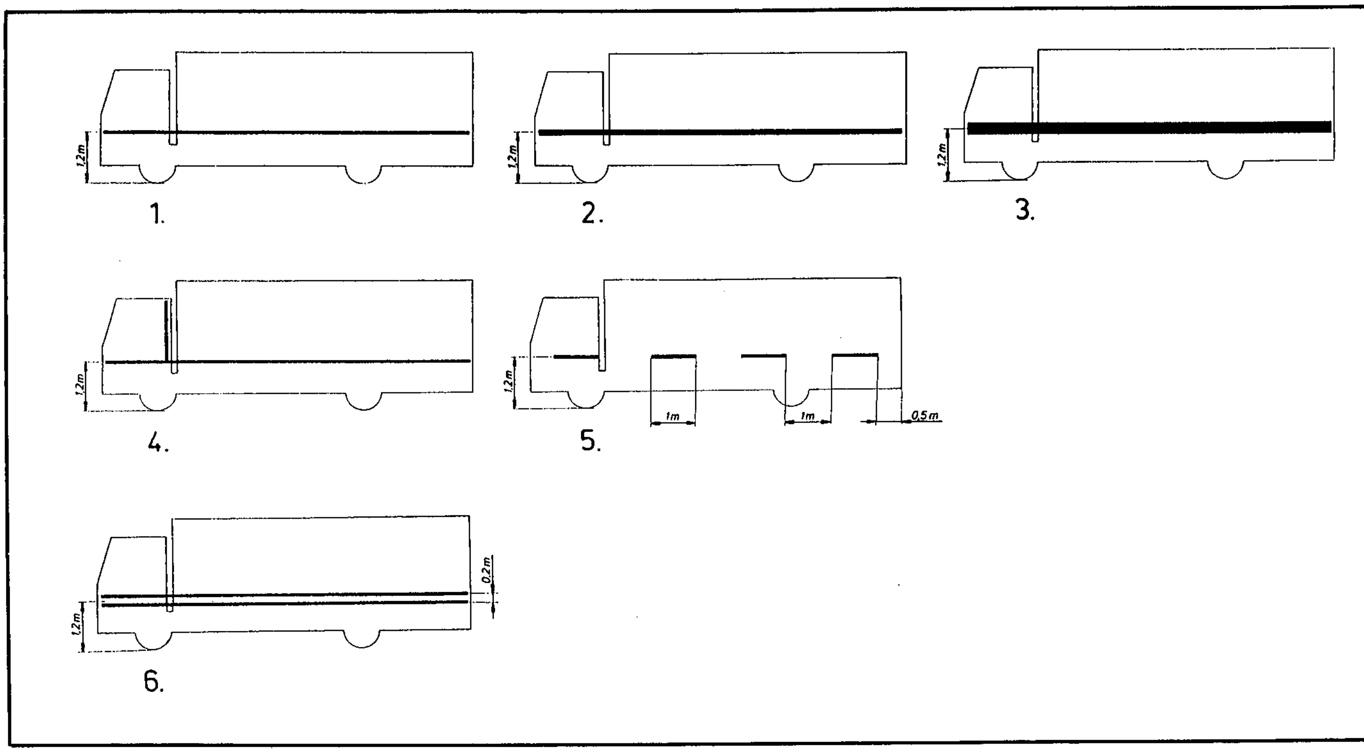

Figure 3. Side marking of trucks

In this Figure the optimal luminances L for the markings shown in Figure 3 and 4 are plotted. The optimal markings are

marking 3: (Figure 3)
marking 6: (Figure 3)
marking 7: (Figure 4)

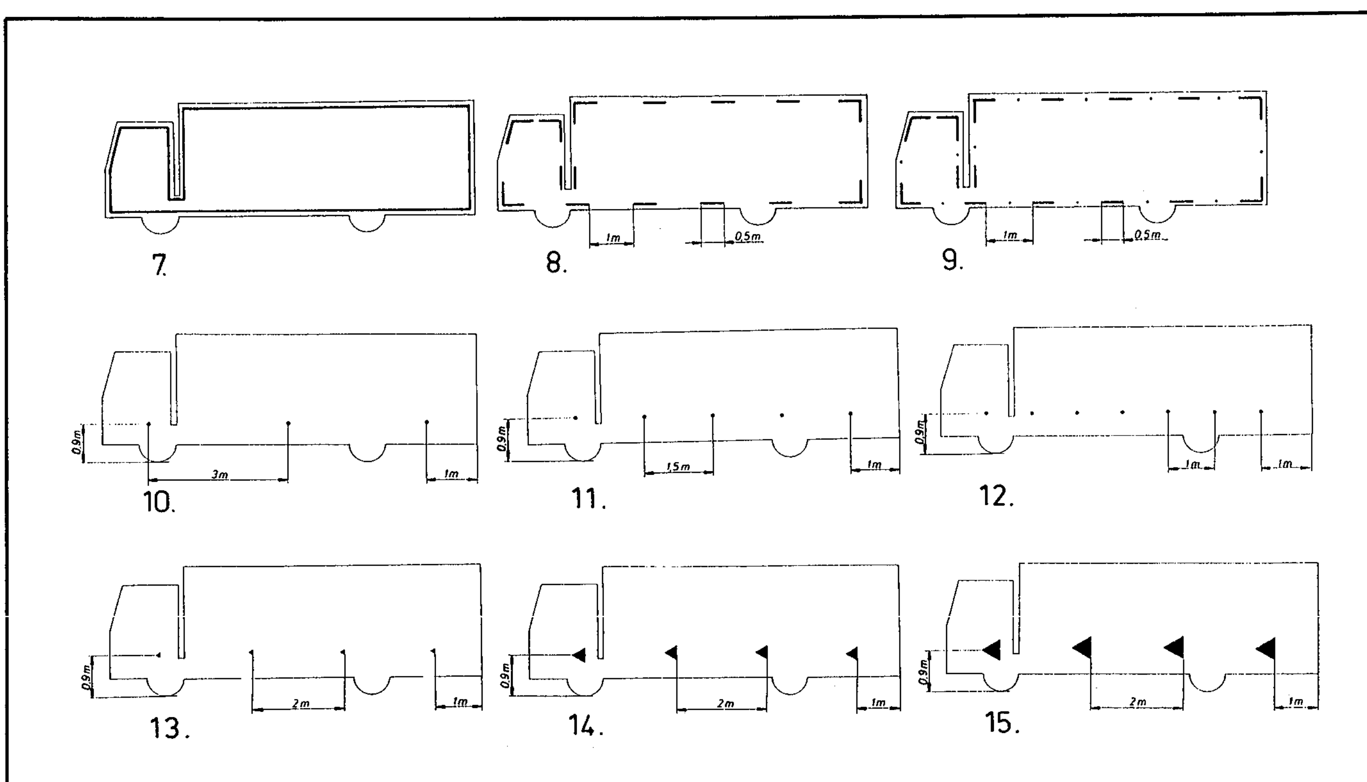

Figure 4. Side marking of trucks

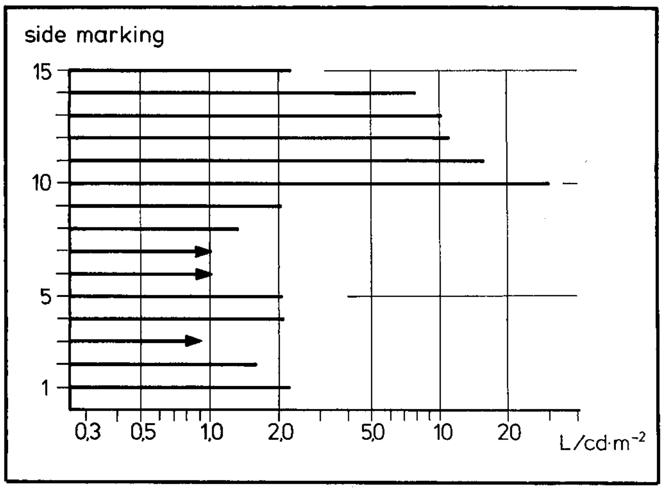

Figure 5. Luminances L for optimal side marking of trucks

The worst markings are 10, 11, 12, 13, 14, the markings with dots and triangles.

Rear Markings of Trucks

In the same test-setup the 15 different rear markings as shown in Figure 6 and 7 (annex) were tested.

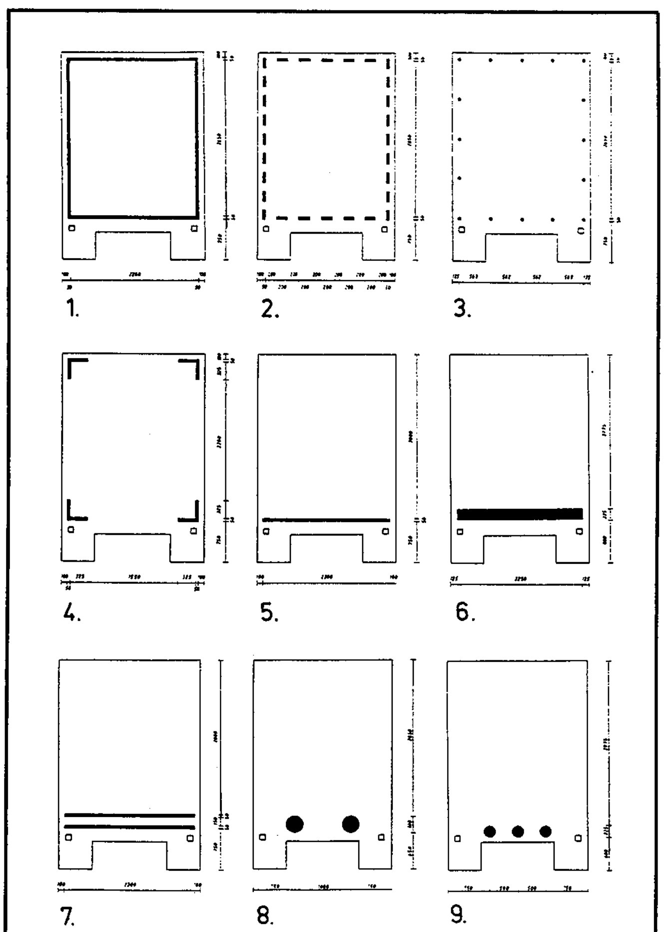

Figure 6. Rear marking of trucks

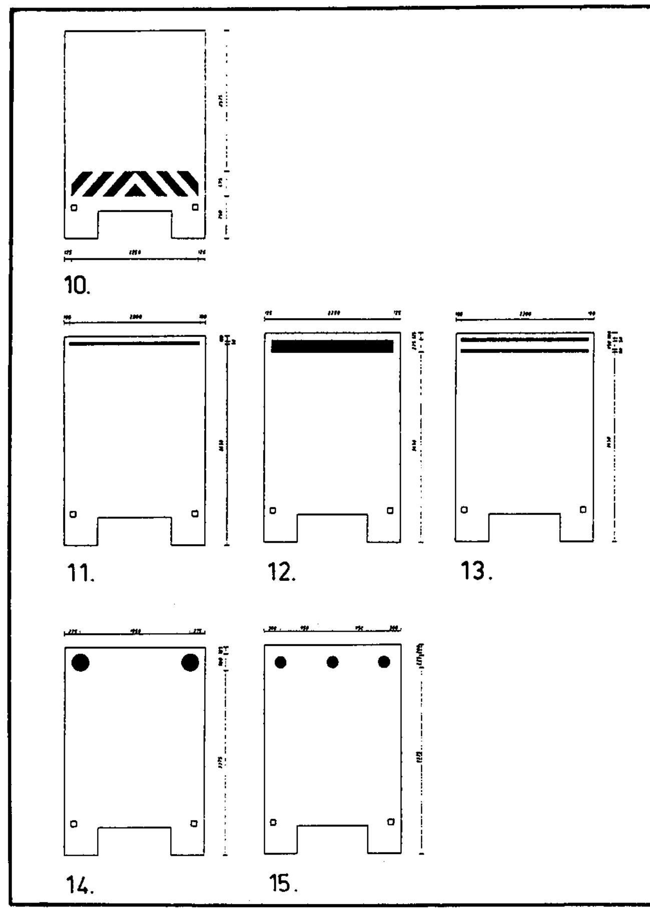

Figure 7. Rear marking of trucks

The results for the rating experiment are shown in Figure 8 were again the optimal luminances L for the 15 different rear markings are plotted. The optimal markings are

marking 1: (Figure 6)
marking 12: (Figure 7)
marking 14: (Figure 7)

The worst case is marking 3, the marking with dots.

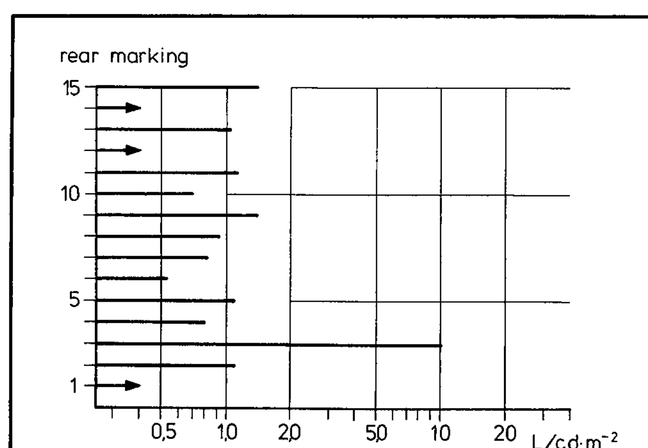

Figure 8. Luminances L for optimal rear marking of trucks

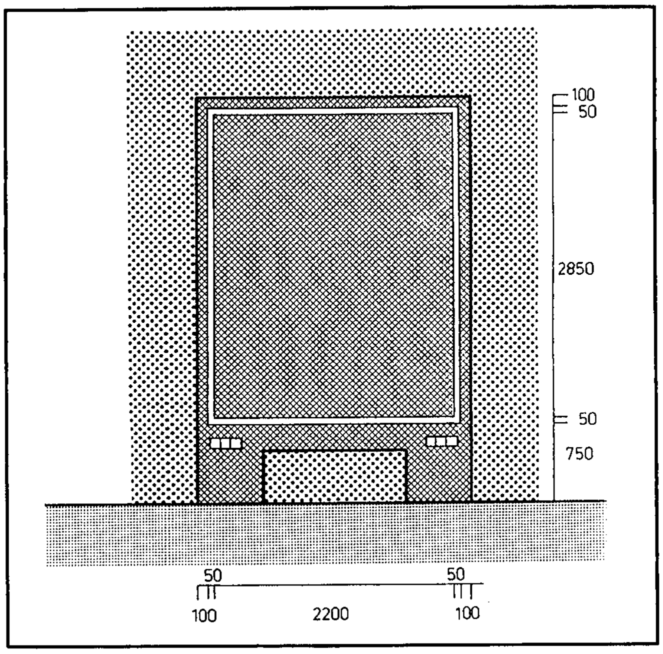

Figure 9. Rear side of a truck with a conture-marking with passive materials

Optimal Marking of Trucks

Out of the down scaled experiments one optimal marking is for example the conture-marking as shown in

Figure 4: marking 7
Figure 6: marking 1.

A rear side of a truck with this kind of marking is plotted in Figure 9.

Improvement of Side Visibility for Safety While Turning

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Abstract

In Japan, fatal accidents in which persons are sometimes caught in heavy-duty trucks when the trucks turn left became a big social problem starting around 1976. (In Japan, the driver's seat is usually on the right hand side.) The analysis result of the accidents showed that it will be effective in reducing the accidents to improve the visibility to the left and to prevent persons from being caught under truck rear wheel. To remedy the situation, the introduction of regulations were considered which require the improvement of "indirect" visibility (visibility through mirrors), betterment of the pedestrian protection side guard, addition of the direction indicator lamp at the middle of the vehicle side, etc.. On the other hand, each truck manufacturer newly installed an auxiliary side window in the left door and enlarged the front windshield, thus improving visibility, especially to the left. This report describes the introduction results of auxiliary side window and enlarged front windshield. Thanks to the effects of these revisions, combined with those of regulations and others, the number of the fatal accidents is now approximately half of that for 1976.

Introduction

In Japan, fatal accidents in which persons are caught in heavy-duty trucks when the trucks turn left became a big social problem starting around 1976.

In those days, the number of such accidents was not large. But the other party in the accident, such as pedestrians, bicycles and motor bicycles, was in a disadvantaged position. Also, in many cases, death accidents were tragic, such as persons being run over and killed. Therefore, they probably attracted public attention.

As remote causes of the accident, heavy-duty trucks run in the city, creating the traffic situation where they are mixed with pedestrians, bicycles, motor bicycles and other vehicles at intersections. In addition, many intersections are narrow, and as a result, the difference in turning radius between the front left wheel and the rear left wheel becomes large, that is, the rear left wheel turns sharper than the other party thinks, in left turn (Fig. 1).

In this paper, the authors report the history of the revision made to solve the problem, taking the case of the Isuzu vehicle as an example. The 1978 and 1986

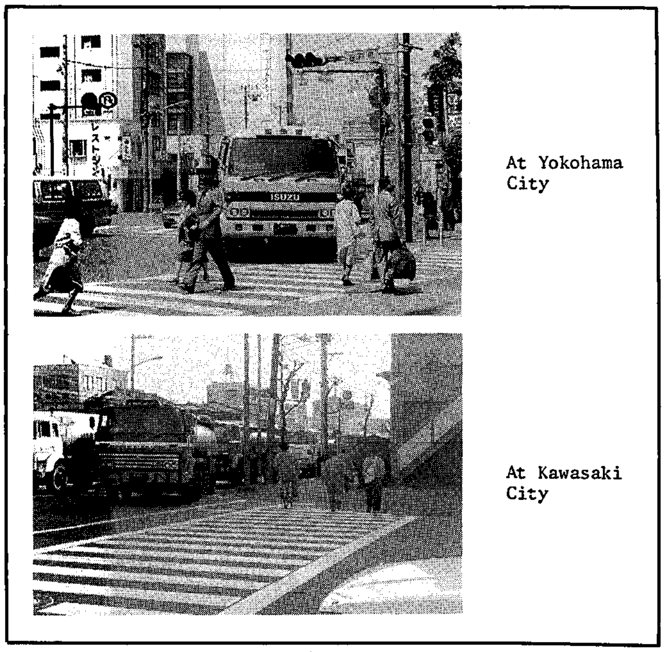

Figure 1. Typical city intersection in Japan, heavy-duty-truck mixed with pedestrians and bicycle on a pedestrian crossing

model year Isuzu vehicles are shown for reference in Fig. 2.

Additionally, in Japan, the driver's seat is usually on the right hand side, therefore, left turn means to turn to the opposite side of the driver. Most of the heavy-duty trucks used in Japan are the cab over the engine type.

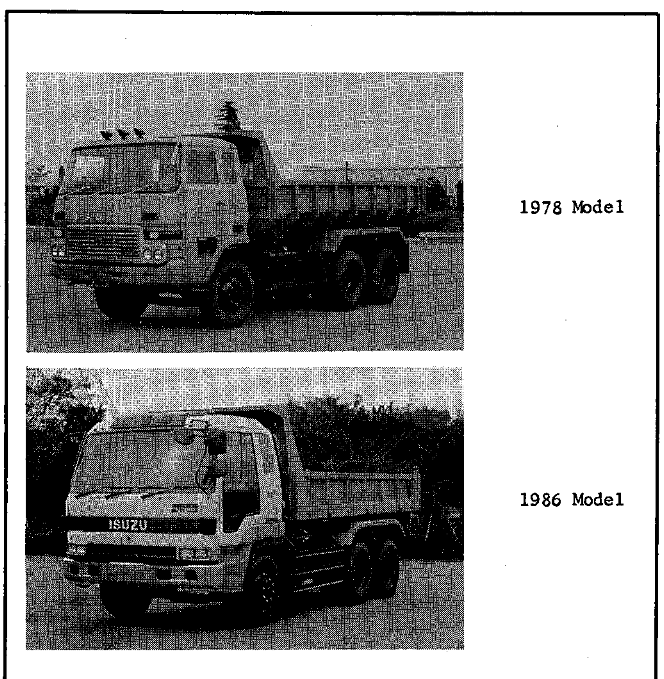

Figure 2. 1978 and 1986 model year Isuzu vehicles

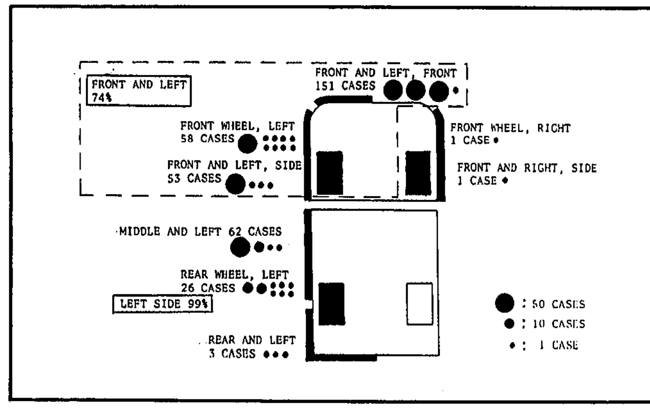

Figure 3. Collision portions of the heavy-duty trucks causing significant left-turn accidents

Analysis of Accidents

Typical significant accidents in 1978 by heavy-duty trucks (partially including buses) at left turn were analyzed in order to study counter-measures(1). Out of the total 2902 cases of accidents, 355 cases were analyzed. Shown below is the summary of the analysis results. (The significant accidents means both fatal and serious accidents.)

1) As shown in Fig. 3, collision was mostly with the left side of the truck. Especially, the front and left portion accounts for 74%.
2) As shown in Fig. 4, 83% of the truck drivers who caused significant accidents did not recognize the other party of the accident.
3) As shown in Fig. 5, collisions with the bicycle and the motor bicycle are many, accounting for 88%.

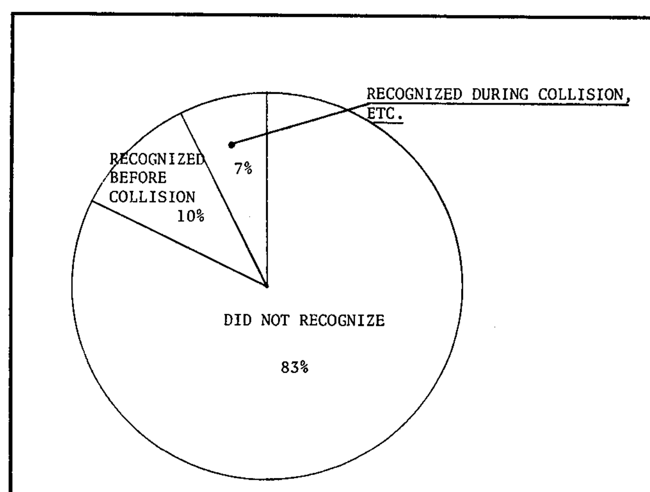

Figure 4. Recognition of the other party of the accident at the time of significant left-turn accidents by heavy-duty trucks

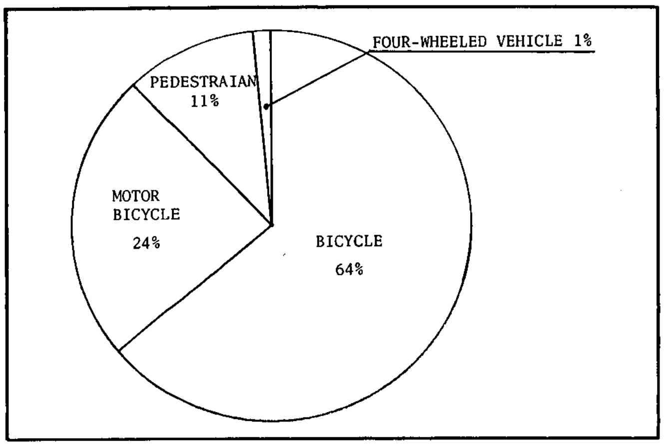

Figure 5. Classifications of the other party that collided with a left turn heavy-duty truck

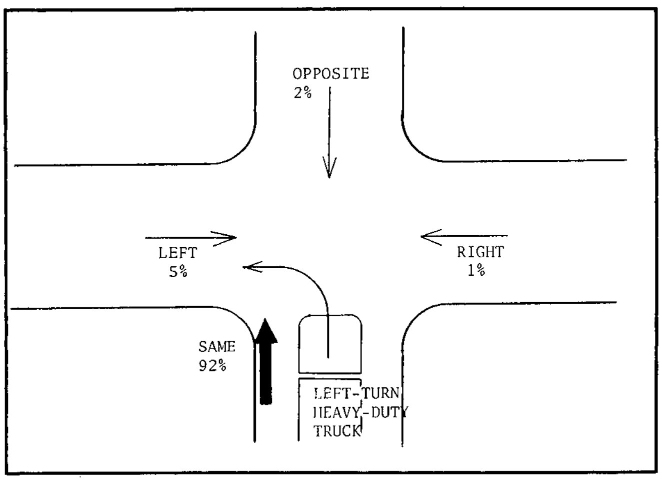

Figure 6. Moving direction immediately before the accident of the other party that collided with a left turn heavy-duty truck

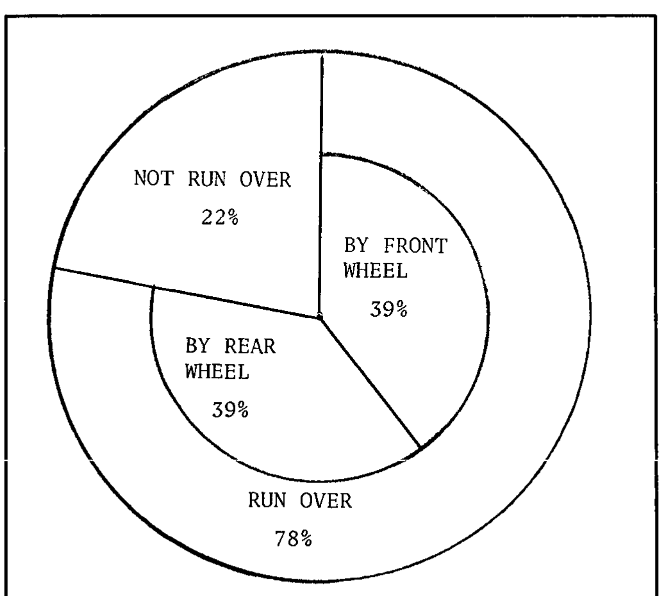

Figure 7. Rate of running over in significant left-turn accidents by heavy-duty trucks

4) As shown in Fig. 6, the direction of the other party immediately before collision is mostly the same as that of the heavy-duty truck, accounting for 92%.
5) As shown in Fig. 7, the rate of the other party run over by the truck wheel after collision is high, accounting for 78%. Half of the number is the case of running over by the rear wheel.

The above can be summarized as follows. In many cases, a heavy-duty truck unknowingly collides at the front and left portion with the two-wheeled vehicle running alongside the truck, leading to fatal accidents by running over.

Revisions to the Vehicle to Prevent Significant Accidents

- Descriptions of visibility improvement—

Improvement of the visibility to the left

To prevent significant accidents, it is a deciding factor to find accurately and quickly the other party that is on the left side, especially in the front and left of the truck. Therefore, measures were taken to improve both "direct" visibility (visibility through windows) and "indirect" visibility (visibility through mirrors)

Improvement of direct visibility

1) *Addition of an auxiliary window in the left door* As shown in Fig. 8, an auxiliary window was formed in the lower half of the left hand door so as to secure the visible area more than two times that before revision.
2) *Enlargement of the front windshield* As shown in Fig. 8, the windshield was increased vertically by 6.7 inch (170 mm) resulting in a 25% increase in windshield area and a 6° increase in lower visibility angle.

Improvement of indirect visibility. Japanese Safety Regulations for Road Vehicles were revised and tightened, and, as a result, visibility through the mirror was improved.

In Japan, the convex mirror has been used as a rear view mirror, different from Western countries where the flat mirror is used.

1) *Change of Japanese Safety Regulations for Road Vehicles, Article 44*
As shown in Fig. 9, the required visibility range was greatly increased.

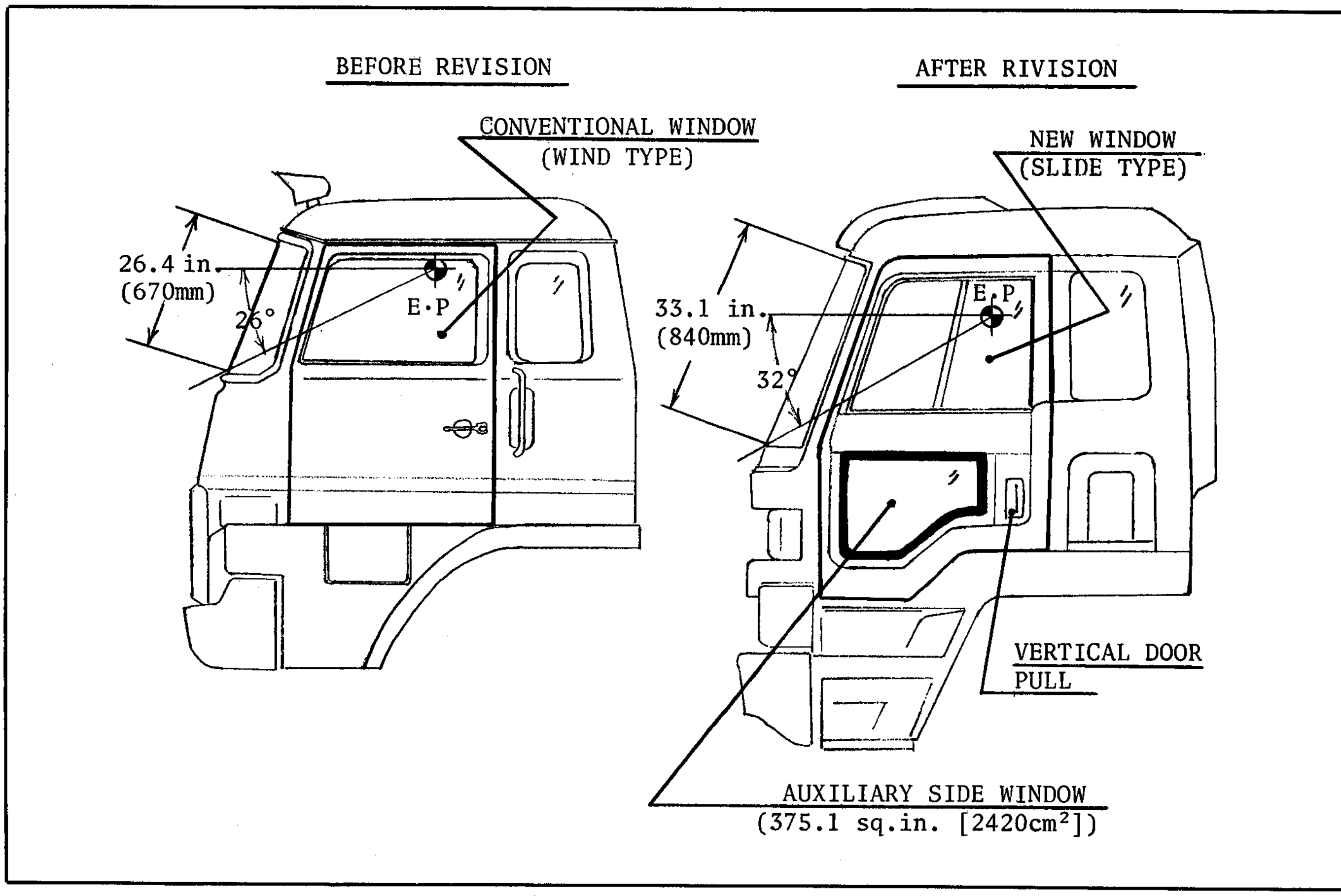

Figure 8. Comparison of the cab before and after revision

2) *Introduction of the three mirror system*
In order to meet the revised regulation, the side-under-view-mirror as shown in Figure 10 was installed. At the same time, the side-view-mirror and the under-view-mirror were enlarged.

Others

Besides the measures to improve the visibility to the left, the following two revisions were made to meet the requirement of the regulations, thus lessening the extent of injuries and reducing accidents. They are introduced here because they are closely connected with the main subject.

1) *Improvement of the pedestrian protection side guard (Article 18.2, Japanese Safety Regulations for Road Vehicles)*
The side guard was changed to a larger one with protective ability improved to reduce the extent of injuries due to running over by the rear wheel (Fig. 11).

2) *New installation of the mid direction indicator lamp. (Article 41, Japanese Safety Regulations for Road Vehicles)*

In order to securely transmit the intention of left turn to the two-wheeled vehicle running by the side of the truck, a direction indicator lamp was added at the middle of the vehicle body side (Fig. 12).

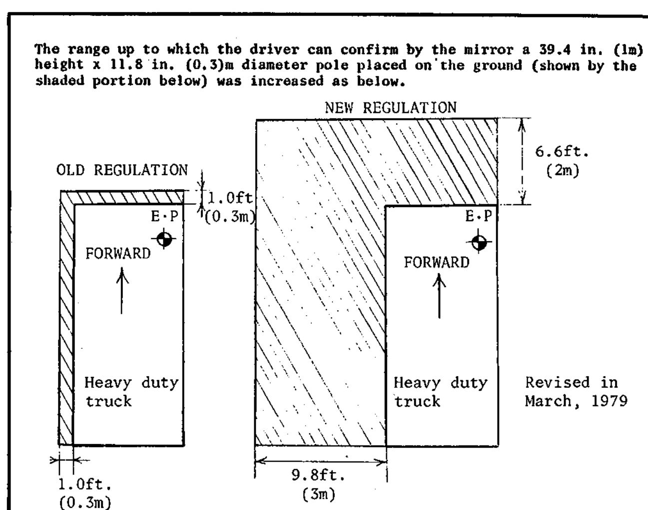

Figure 9. Contents of the amendment of Japanese Safety Regulations for Road Vehicles (Article 44) "Rear-View-Mirror, etc."

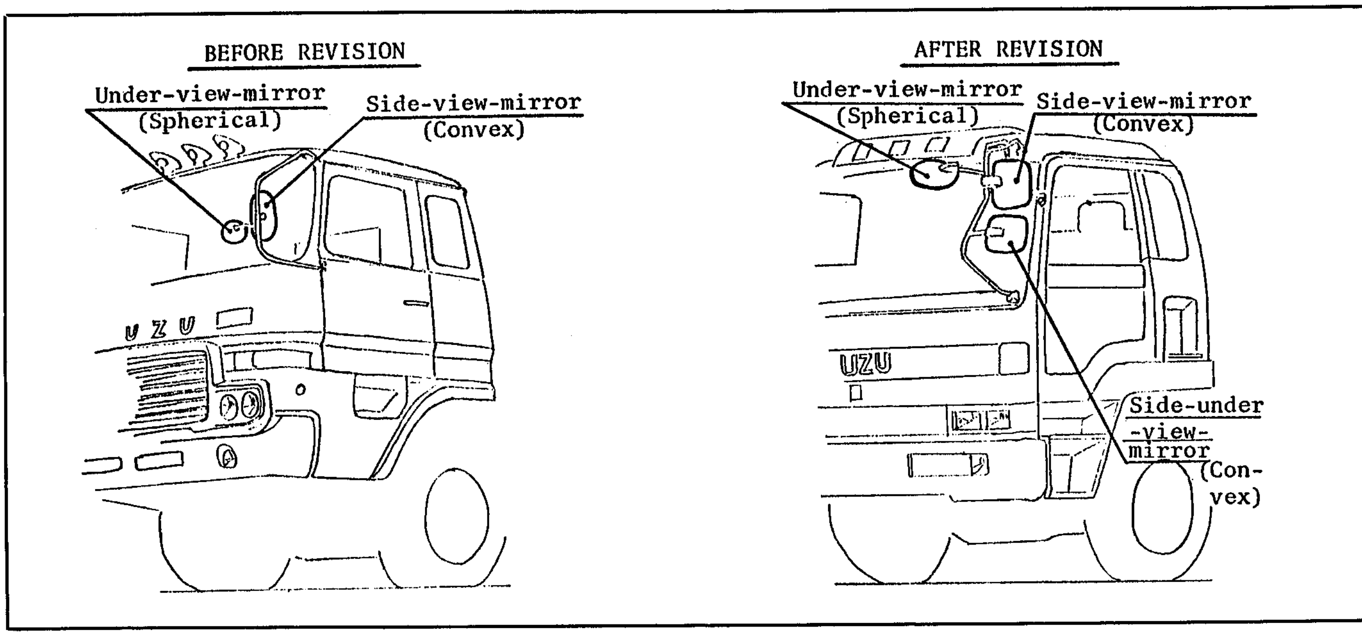

Figure 10. Comparison of the mirror before and after revision

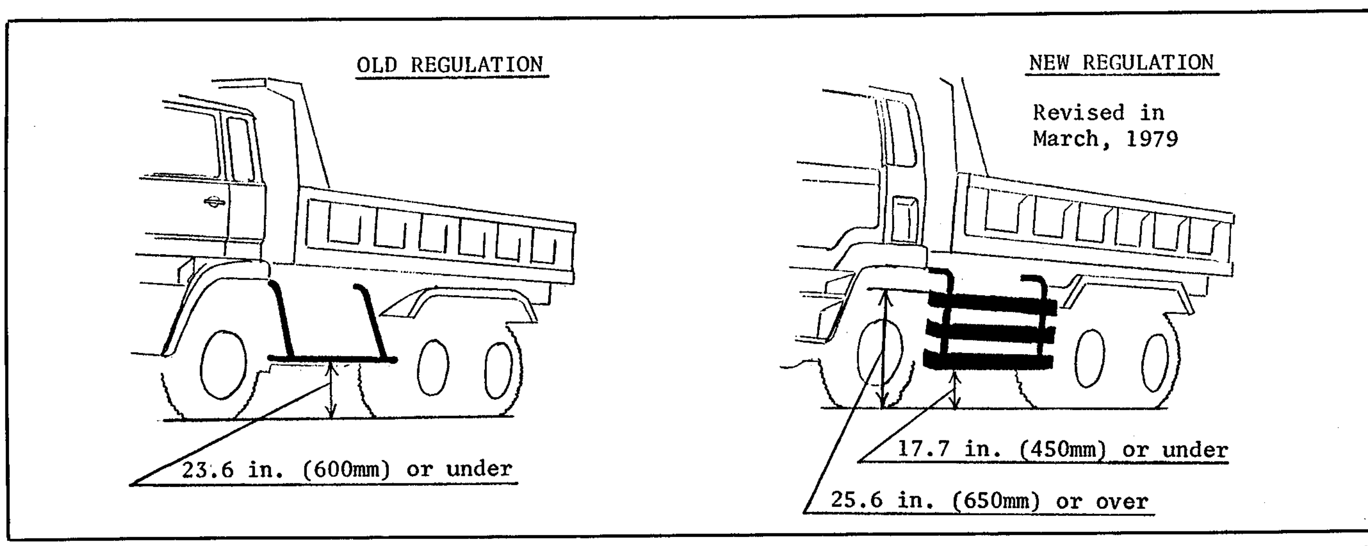

Figure 11. Contents of the amendment of Japanese Safety Regulations for Road Vehicles (Article 18.2) "Pedestrian Protection Side Guard, etc."

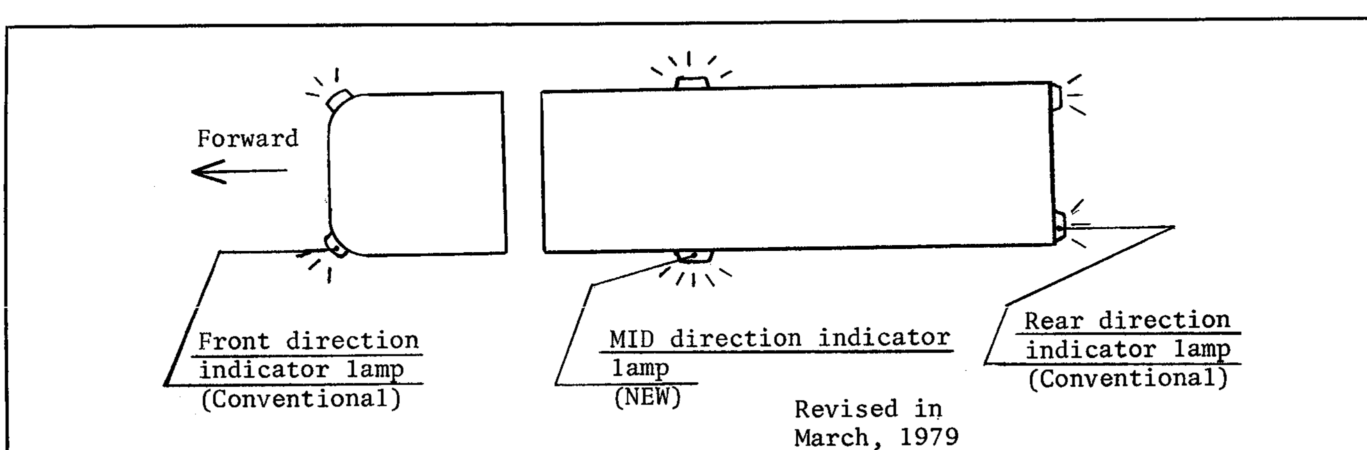

Figure 12. Contents of the amendment of Japanese Safety Regulations for Road Vehicles (Article 41) "Direction Indicator Lamp"

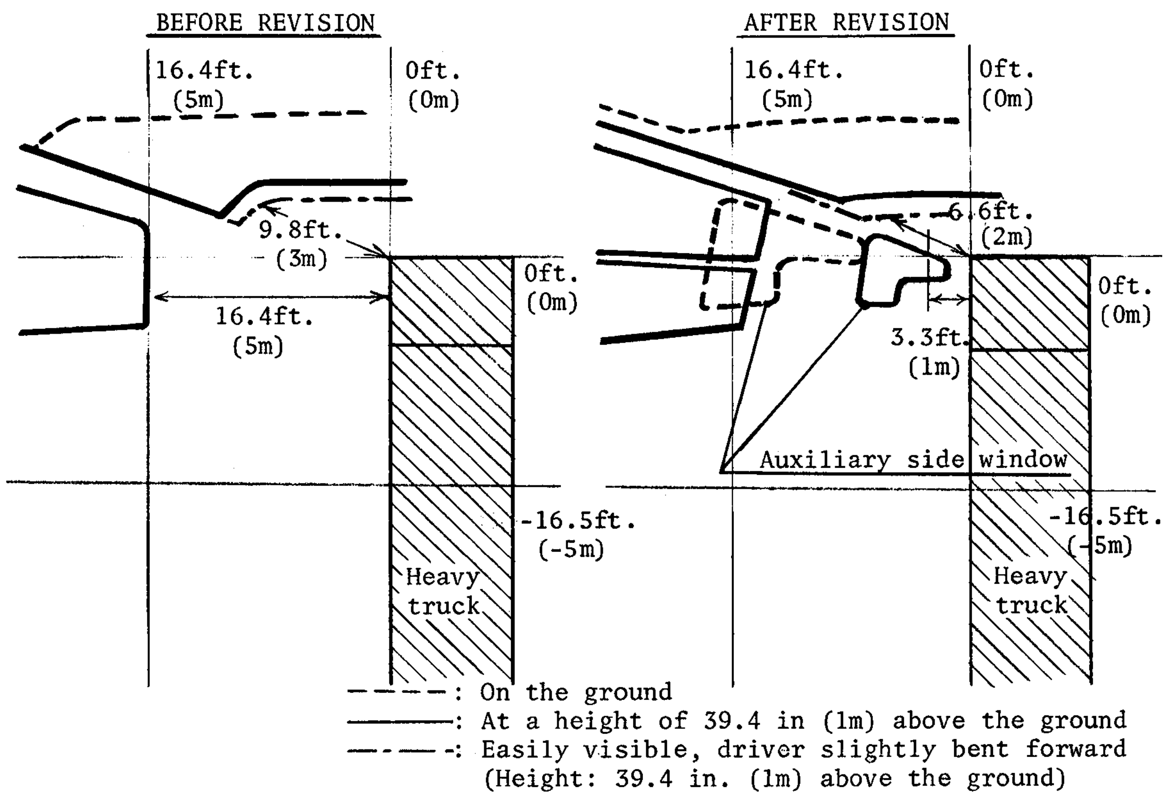

Figure 13. Direct Visibility diagram before and after revision

Effects of Visibility Improvement

Improvement of static visibility

The addition of an auxiliary side window showed a great effect of improving direct visibility (Fig. 13). Thanks to this window, when a human being came near the left side of the vehicle, it became possible to recognize him or her at a distance of approx. 3.3ft (1m) or more from the vehicle. Also, for the human being approaching diagonally in front, the recognizable distance was reduced to approx. 6.6ft (2m) or more. (Before improvement, this distance was approx. 6.4ft (5m) or more and approx. 9.8ft (3m) or more, respectively.)

Indirect visibility approximately doubled (Fig. 14). Especially, the visibility to the front and left of the vehicle was increased very much by the new addition of the side-under-view-mirror.

Effect of preventing accidents

Actual accidents were assumed and the effect of preventing accidents was confirmed. Fig. 15 shows example patterns for 4 seconds before collision with the pedestrian, bicycle and motor bicycle. Fig. 16 shows the relative location of the pedestrian, bicycle and motor bicycle rewritten on the visibility diagram with the truck placed in a fixed location.

From this, the following things are known, and it could be said that a series of countermeasures have large effects of preventing accidents (Fig. 17).

1) Chances of catching higher quality visual information by direct view increased.
2) The time of double visual confirmation by both direct and indirect view increased.
3) The time of visual confirmation only by the under-view-mirror (spherical) decreased.
4) The time of complete invisibility decreased almost to zero.

Effects of the Countermeasures Seen From the Statistics of Traffic Accidents

Fig. 18 shows the change in the number of fatal traffic accidents. Although total fatal accidents level off through the entire investigation period, the number of left-turn fatal accidents has been reduced in recent years to approx. half of that for 1976. Like this, the series of countermeasures have been found

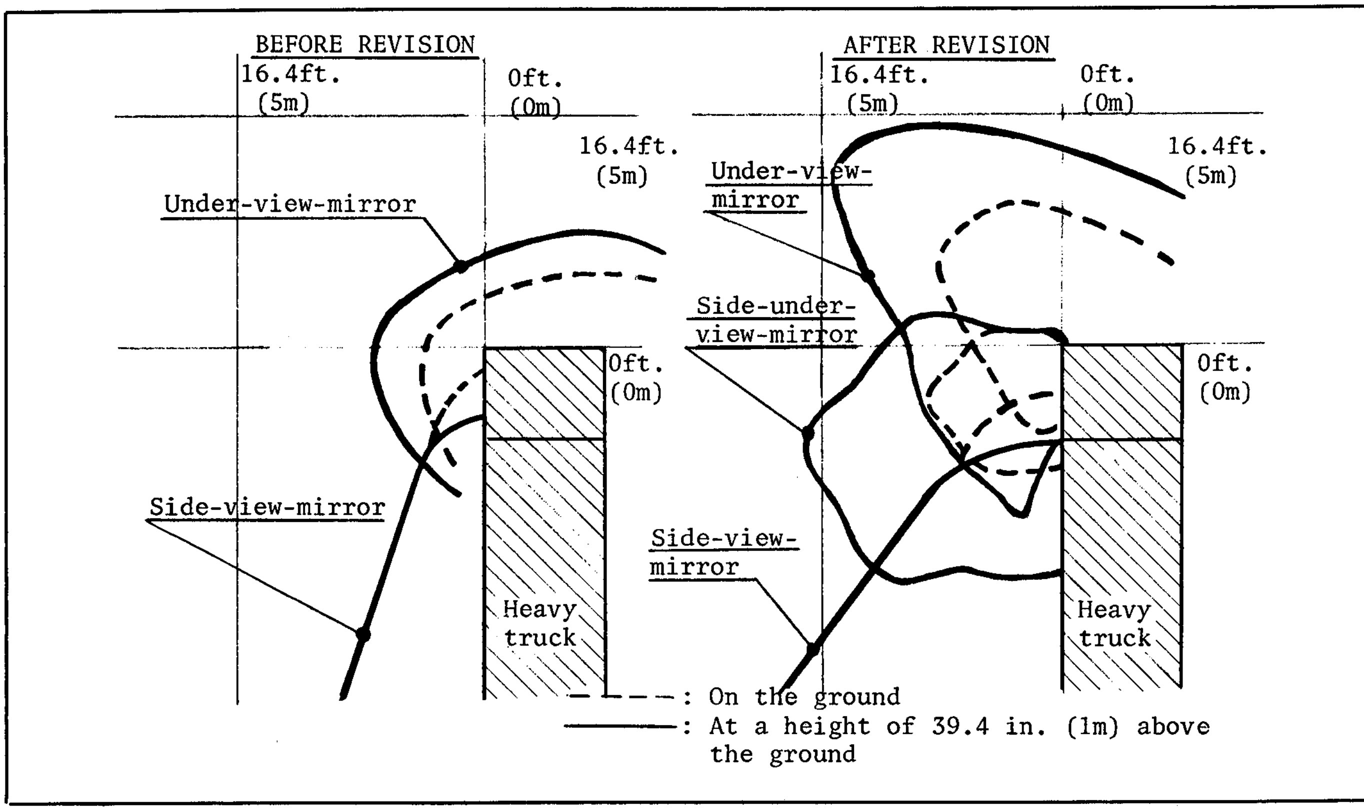

Figure 14. Indirect Visibility diagram before and after revision

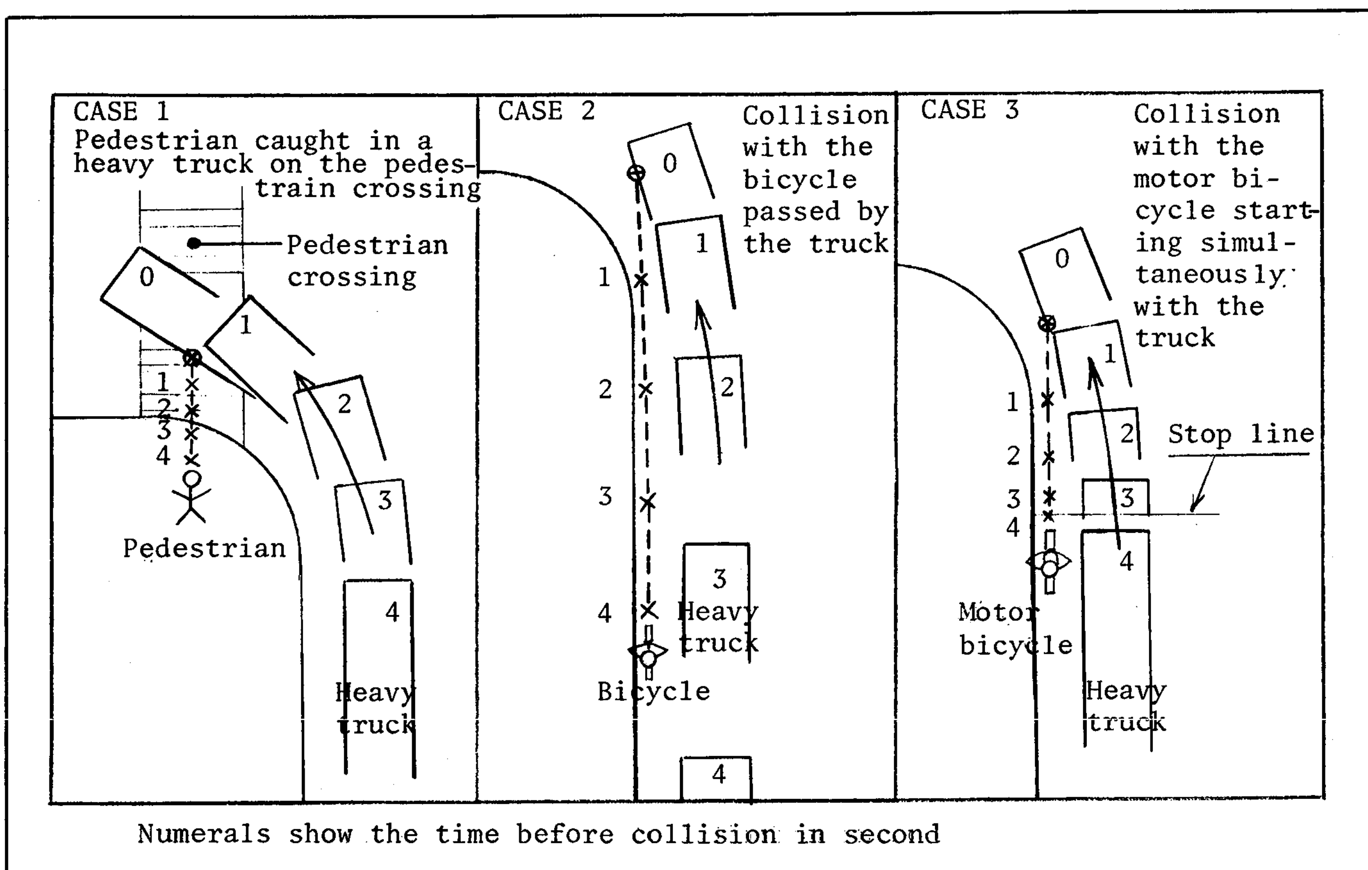

Figure 15. Typical patterns of collision with the left-turn heavy-duty truck

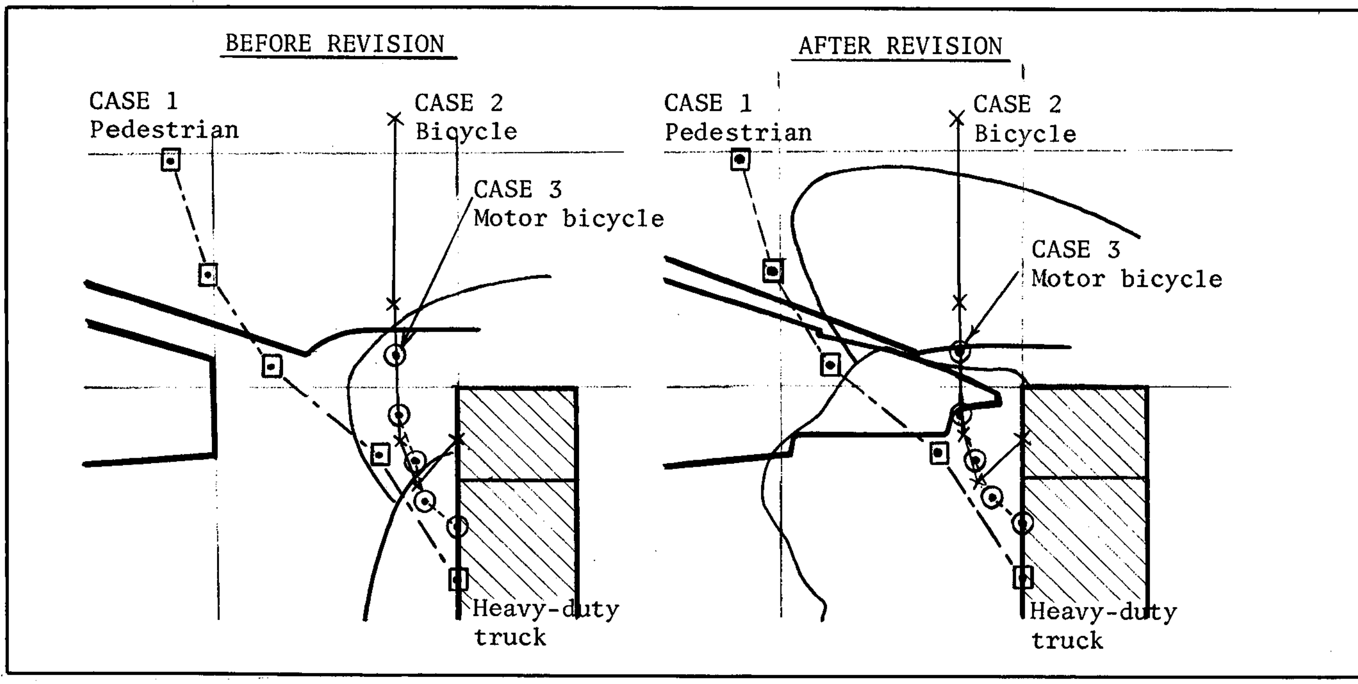

Figure 16. Movement of the other party of the left-turn accident when viewed from the heavy-duty truck

very effective in reducing left-turn fatal accidents although the effects are combined with those of the improved road environments, tightened regulations and other efforts by the government and with those of the education of the persons ranging from children to heavy-duty truck drivers by the police.

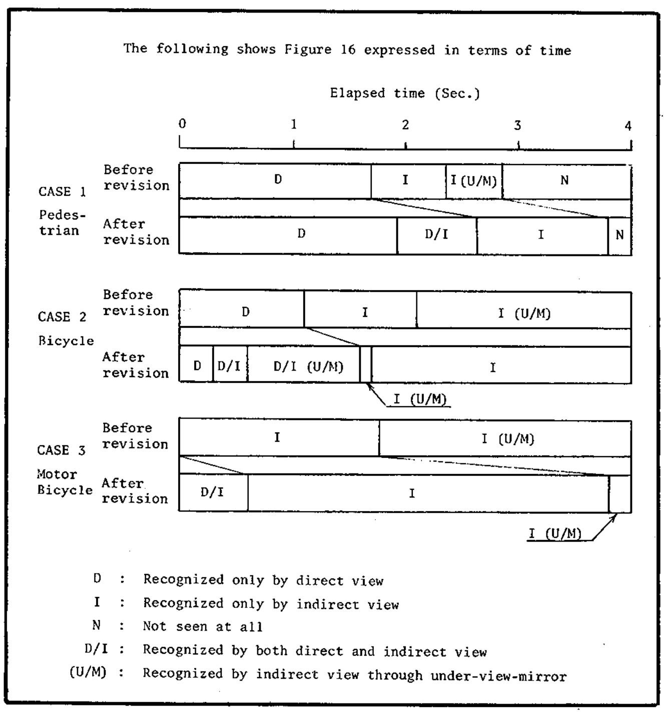

Figure 17. Comparison of ease of discovering the other party of the accident before and after revision

In this paper, report has been made of the improvement of the visibility to the left in the case of Isuzu Motors. But similar measures have been taken also by each of the other manufacturers of heavy-duty trucks.

Conclusion

Addition of an auxiliary window in the left hand door, enlargement of the front windshield and use of the three mirror system were reported as means to improve visibility. It is considered that these are the measures near to ideal at the current level of technology. However, no matter what improvement is made

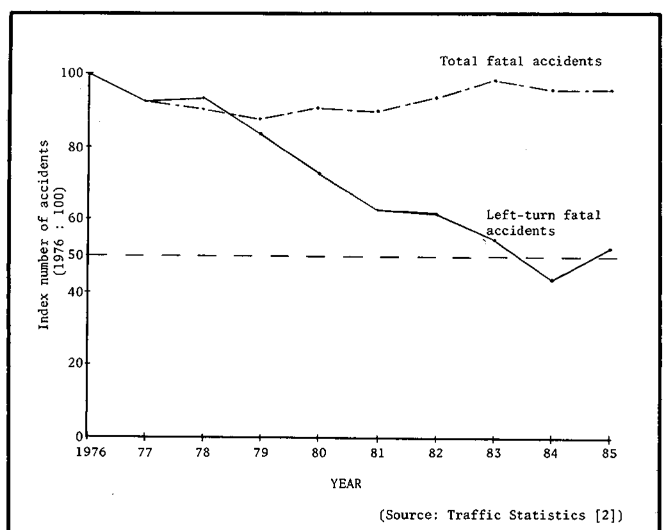

Figure 18. Change in total accidents and left-turn fatal accidents

in visibility, it will be meaningless unless drivers take care and confirm safety.

From this point, studies are under way on the installation of the system on the vehicle which will warn drivers when there are persons around the truck. However, as it is difficult to distinguish between fixed objects on the road and persons, this system has not yet been put into practical use. The authors hope that electronics will make great progress and that it will be applied to realize such a system in the future.

References

1. Traffic Safety Measures Committee of Japan Automobile Manufacturers Association, INC. "Report of Actual Conditions and Analysis of the Accidents by Heavy-Duty Vehicles in Turning Left and Right" (1980)
2. Traffic Bureau, National Police Agency, Japan "Traffic Statistics" (1976—1985) Japan Traffic Safety Association

Analysis of Heavy-Freight Vehicle and Tank-Truck Accidents

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Abstract

It is intended to analyse heavy-freight vehicle accidents paying special consideration to tank-truck accidents with the help of the DAIMLER-BENZ driving simulator in Berlin. After the interpretation of accident statistics the characteristics of freight- and tank-vehicle accidents were formulated by "characteristic accident patterns". On the basis of these patterns real accidents could be evaluated and some typical accidents (single-vehicle-accident, frontal- and front/rear-collision) were chosen for our intended profound simulator analysis.

A comparison between tank-truck and heavy-freight vehicle accidents shows a 50% higher participation of tank-trucks in front/rear collisions on motorways as well as in single-vehicle accidents and in frontal collisions on the highways ("Bundesstraße, Landstraße"). It is assumed that "liquid sloshing" could be one of the causal factors of such tank-truck accidents. The interaction between liquid sloshing and the accident characteristics of tank-trucks shall be investigated with the help of the driving simulator. A real-time analysis of liquid sloshing demands a simple, yet precise mathematical description. A mechanical model (pendulum-analogy and spring-mass system) for the simulation of combined transversal and longitudinal liquid sloshing of tank trucks is proposed; and the equations of motion are presented.

Introduction

In the analysis of accidents involving commercial vehicles, the effects produced by shifting cargo such as that encountered in partially loaded tank trucks have hitherto been scarcely considered as a factor possibly causing road accidents. The objective of this study was to investigate commercial-vehicle accidents with special attention directed to shifting loads, and with the aid of calculated and experimental real-time simulation in the Daimler-Benz driving simulator. The resulting analysis performed here is part of the project BASIS [German acronym for "Evaluation of Active Safety Devices in Simulator Work"] sponsored by the German Ministry for Research and Technology (BMFT).

Hypothesis: "Liquid Sloshing" as a Possible Cause of Accidents

The point of departure for this analysis was the question as to whether shifting cargo—particularly, sloshing liquid loads—in commercial vehicles could influence the dynamic road behaviour in such vehicles (specifically, tank trucks) to such an extent as to represent a cause of road accidents.

An analysis of accident statistics concerning tank trucks reveals that such vehicles are, in relative terms, 50% more frequently involved in road accidents than other commercial vehicles. Such increased participation in accidents has been observed for tail-end accidents on German freeways (Autobahn), as well as for single-vehicle accidents and head-on collisions on national highways and rural roads in Germany. This phenomenon is emphasized in *Figs. 1 and 2* as a "conspicuous difference" in the statistics. Assessment of such statistics will of course point out characteristics particular to tank-truck accidents; however, it does not suffice to allow conclusions to be drawn with respect to the influence exerted by the load. Consequently, statements of various authors will, to begin, be cited below in initial support of the hypothesis stated above.

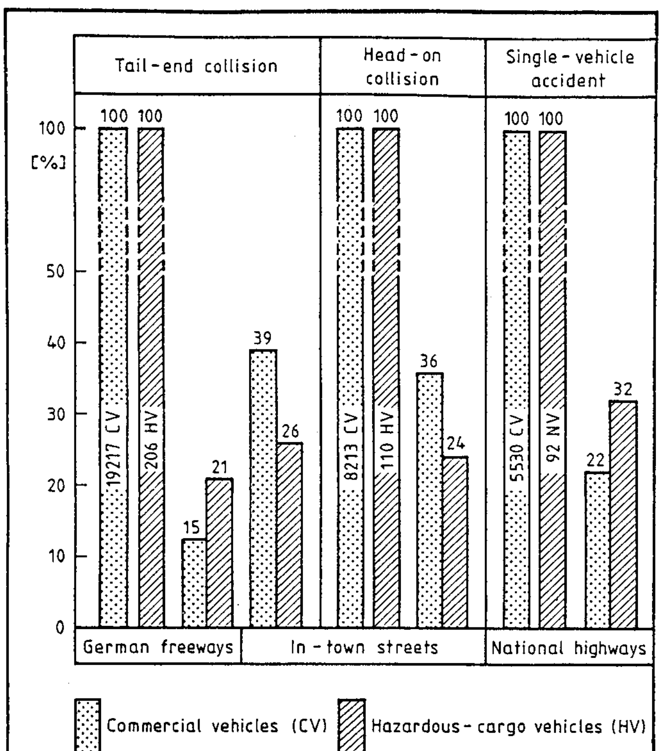
Bauer[2] investigated the effects of shifting fluids on the structural and flight stability of rockets, and later broadened his studies to cover tank trucks. With respect to semitrailer tank trucks, he reported as follows: "A liquid loading may lead to considerable handling difficulties if a free liquid surface exists, sloshing about its fundamental natural frequency, thus creating forces and moments acting on the vehicle, which may easily exhibit a multiple of the inertia force determined for the same mass if it were considered rigid and fixed to the container."[2, Part I, p. 45]. In their analyses of tank-truck accidents in Australia, Griffiths et al. reported the following: "Among the factors that lead to tanker rollover are their high center of gravity (c.g.), 'soft' roll stiffness, and sloshing of the liquid load"[3], p.21.

Erwin et al[.4] investigated the threshold value for the lateral acceleration beginning at which vehicle rollover can be expected. *Fig. 3* throws light on the association between rollover threshold and the percentage frequency of single-vehicle rollover accidents. Investigation here was made of the three-axle tractor/two-axle semitrailer in fully loaded condition: i.e., without possibility of the shifting of liquid loads. The graphical representation is based on evaluation of records from the USA Bureau of Motor Carrier Safety (BMCS), in combination with a dynamic vehicle-behavior model which was used to help determine the rollover threshold.

The percentage share of rollovers in single-vehicle accidents demonstrates a direct relationship to the rollover threshold, as the nonlinear plot in *Fig. 3* makes clear. According to the opinion of the authors involved here, the threshold value is primarily dependent on the height of the center of gravity.

Even though partially loaded vehicles were not considered in this investigation, it is nevertheless obvious that a progressive, percentual rise in the probability of a rollover in a single-vehicle accident occurs together with a decrease in the rollover threshold. A low rollover threshold, furthermore, is occasioned not only by a high center of gravity—but also as a result of dynamic fluid forces.

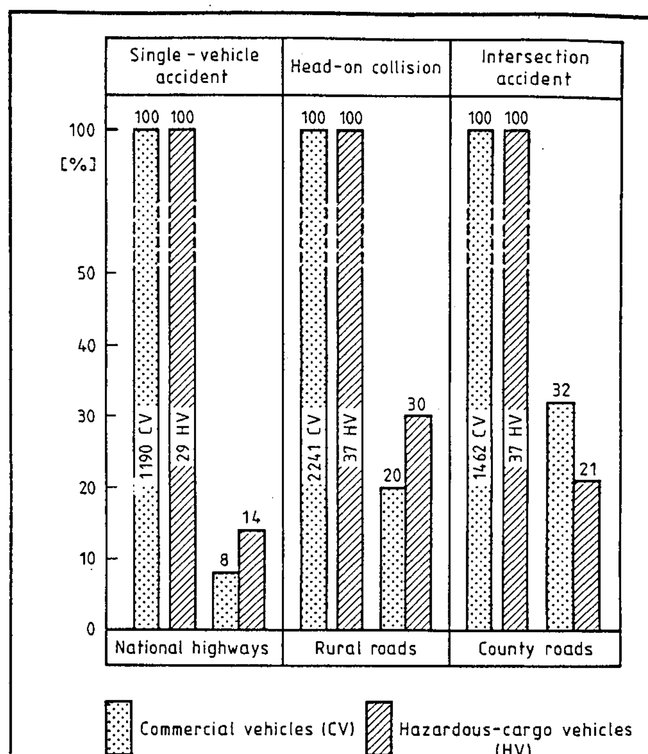
In investigations carried out by Strandberg[5], attention was called to experiments conducted by Abramson et al[6], in pointing out the distinct possibility that fluid forces in the case of resonance can rise to values seven times as great as those observed



Example for reading graphs: 21% of all accidents with hazardous-cargo vehicles in tail-end accident take place on German freeways (Autobahn)

Data evaluation by TÜV Hazardous-Cargo Accident Analysis /1/

INSTITUT FÜR FAHRZEUGTECHNIK Prof. Dr.-Ing. H. Appel	Conspicuous differences between participation of hazardous-cargo and commercial vehicles in traffic accidents, with characteristic accident types as reference factor	Fig. 1 Stedtnitz



Example for reading graphs: 14% of all hazardous-cargo accidents on national highways take place as single-vehicle accidents.

Data evaluation by TÜV Hazardous-Cargo Accident Analysis /1/

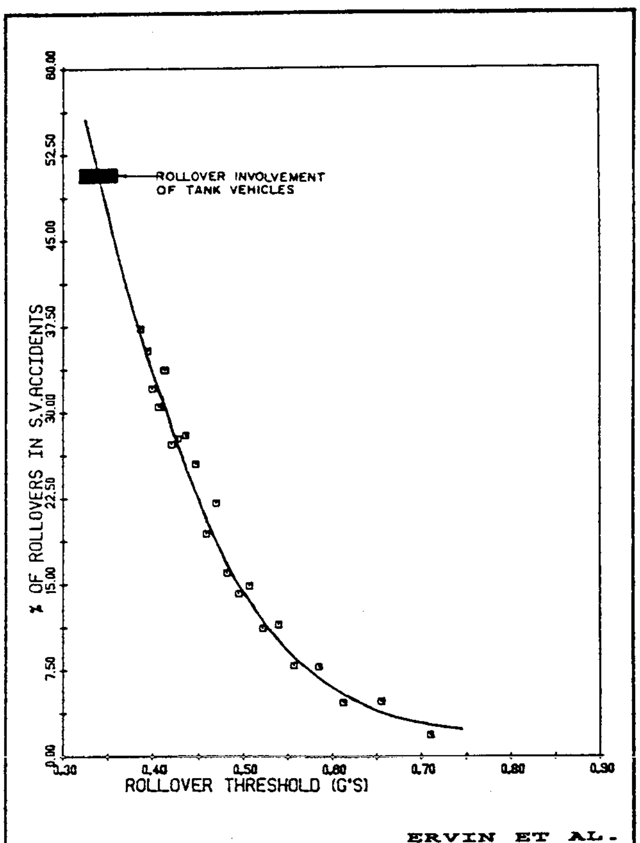
INSTITUT FÜR FAHRZEUGTECHNIK Prof. Dr.-Ing. H. Appel	Conspicuous differences between participation of hazardous-cargo and commercial vehicles in traffic accidents, with types of streets as reference factor	Fig. 2 Stedtnitz

when the liquid load cannot shift inside its container.

The following findings were established concerning the overturn risk of a tank truck from the model experiments conducted by Norström et al[7, p. 69]: "With 50% load volume it was found that the increase in overturning risk compared to rigid load could be up to somewhat more than two times, both in harmonic oscillations at frequencies low enough to occur in normal driving and in the double-lane change maneuver."

Until now, however, such influences on the dynamic vehicle behavior of tank trucks could not be proved in road tests. The difficulties inherent here were reported by TOPAS in the following[8, p. 168]: "Contrary to all expectations, sloshing liquid loads in half-filled containers exhibit only moderate effects; in any case, the measured angles of roll fell below those values observed for the fully loaded vehicle. In a subjective sense, even the slightest movements of the liquid load—for example, the sensation detected by humans when the vehicle is at rest—can readily be detected. The effective consequence for the behavior of the moving vehicle is, however, relatively slight."

The dynamic vehicle measurements serving as the basis for this conclusion do not, however, suffice to allow an evaluation of the influence of vehicle loading on the behavior of such cargo vehicles in accidents. This question can be elucidated only by sufficient analysis of the entire control system complex represented by the driver, his vehicle, and the accident-site circumstances.



INSTITUT FÜR FAHRZEUGTECHNIK Prof. Dr.-Ing. H. Appel

RELATIONSHIP BETWEEN ROLLOVER THRESHOLD AND ROLLOVER ACCIDENT INVOLVEMENT (BASED ON BMCS ACCIDENT DATA FOR THE YEARS 1976-78) /4/

Fig. 3

Accident analysis of heavy-freight vehicles and tank trucks in a closed control system is carried out with the aid of the Daimler-Benz driving simulator[9,10], in conjunction with the Institute of Automotive Engineering at the Technical University of Berlin.

In a parallel test procedure, typical and "characteristic" heavy-freight-vehicle and tank-truck accidents are—on the one hand—selected and recreated in the driving simulator by a large number of test drivers, with testing in this case performed serially. In the parallel procedure, on the other hand, the driving-simulation model is modified to recreate the movement of liquids inside tank trucks.

Characteristic Accidents

Only the analysis of representative accidents can lend hope for the reduction of future traffic accidents on a broad basis. For this reason, it is necessary to select truly "characteristic accidents" for purposes of accident research on a driving simulator.

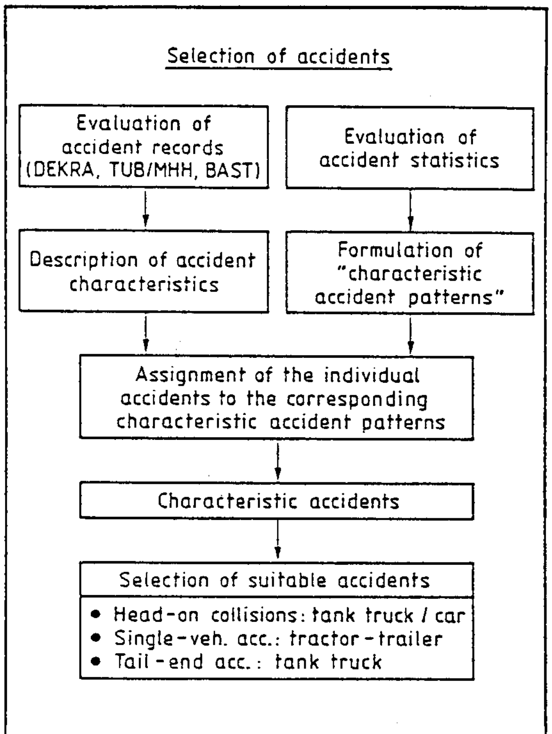
"Characteristic accidents" are considered to be those accidents for which the parameters describing the accident (e.g., type of collision, type of vehicle, collision velocity, etc.) represent, in their combination, percentually salient phenomena observed in accident statistics.

Fig. 4 depicts the parallel procedure for the determination of such characteristic accidents.

"Characteristic accident patterns" result from the study of road accident statistics in which salient, or predominating, patterns can be brought to emerge (see *Fig. 5*). With the aid of these patterns, actual accidents can be assessed with regard to their respective characteristics.

From a comparison of the accident patterns with actual accidents, finally, characteristic accidents were able to be ascertained—from which three typical examples were chosen for further treatment on the driving simulator. These examples were as follows: a head-on collision of a tank truck with a passenger car on a national highway, a single-vehicle accident in which a tractor-trailer unit loaded with hanging loads of meat turns over while negotiating a curve, and a tail-end collision of a tank tractor-trailer unit which runs into a construction-site vehicle on the freeway.

For purposes of simulating the surroundings, the accident sites were recorded by video means and were photographed. With the aid of road plans, the course of the roadway is presently being reconstructed on the driving simulator in the form of a digital video



Institut für Fahrzeugtechnik — Analysis of Commercial-Vehicle / Tank-Truck Accidents with the aid of the Daimler-Benz Driving Simulator — Fig. 4 — Stedtnitz

Type of accident:	Collision with another vehicle approaching headon
Type of vehicle:	Tank truck
Approved total weight:	12... 16 metric tons (also 16... 22 tons in North Rhine-Westphalia)
Type of road:	Rural road
Road conditions:	Dry
Age of drivers:	35... 44
Cause of accident:	Excessive speed under the circumstances Other faults of vehicle driver
Direction of impact:	11...12 o'clock
Point of impact:	Left front of truck
Vehicle hit:	Passenger car
Speed of collision:	70... 80 km/h (relative speed)

Institut für Fahrzeugtechnik — Example of a characteristic accident pattern — Fig. 5 — Stedtnitz

representation, to include the site slopes and lateral inclinations. See[11].

Simulation of the Movement of Liquid Inside Tank Trucks

Basically, two procedures can most effectively be used for the calculation of the movement of liquids inside tank containers. In the first possibility, the liquid is considered to be a continuum, and the forces exerted on the container walls by the liquid are determined with the aid of an approach taken from potential theory. In the second possibility, the liquid is represented as a purely mechanical spring-mass or pendulum system in which the forces arising from the movements of the liquid act at the point of spring attachment on the tank wall, or at the point of pendulum suspension.

Since a liquid simulation system must be prepared for implementation on the driving simulator, the possible approaches for model work must be evaluated with respect to their complexity and their feasibility of being run in real-time conditions. Accordingly, the mechanical approaches—owing to the clarity and the simplicity of the equations of movement which can be employed—must be given preference here over those involving potential theory, although the latter do in fact enable greater accuracy.

Extremely accurate liquid simulations are not, however, required for evaluations of the influence of the liquid on the dynamic behavior of commercial vehicles. It has, for example, been determined that consideration of only the fundamental mode of natural oscillation will suffice to satisfactorily describe the forces exerted by the liquids onto the tank, since the liquid masses and forces of higher modes are negligibly small with respect to the fundamental. According to Slibar and Troger[12], assumption can be made of a mass fraction amounting to only 3 . . . 12% of the mass participating in the fundamental mode, for sufficient consideration of the fundamental harmonic oscillation. The forces occurring on the basis of higher harmonic oscillations are, as a result, correspondingly slight for such cases.

Since the eigen frequencies of the harmonic oscillations lie above the excitation frequencies which arise from the driver's steering actions, the simplified consideration of only the fundamental oscillation is in fact justified.

The basic approach in the preparation of a mechanical simulation model for liquids is based on the assumption that only part of the entire liquid mass participates in the sloshing movement, whereby the remainder of the liquid mass can be considered to remain connected to the tank wall. This validity of this assumption has in the past been confirmed through numerous experimental investigations.

With the exception of one model based on potential theory (the vehicle simulation system TDVS, plus the computer program SLOSH devised for liquids[13]), there has been until now no other spatial liquid-simulation program conceived for freight vehicles.

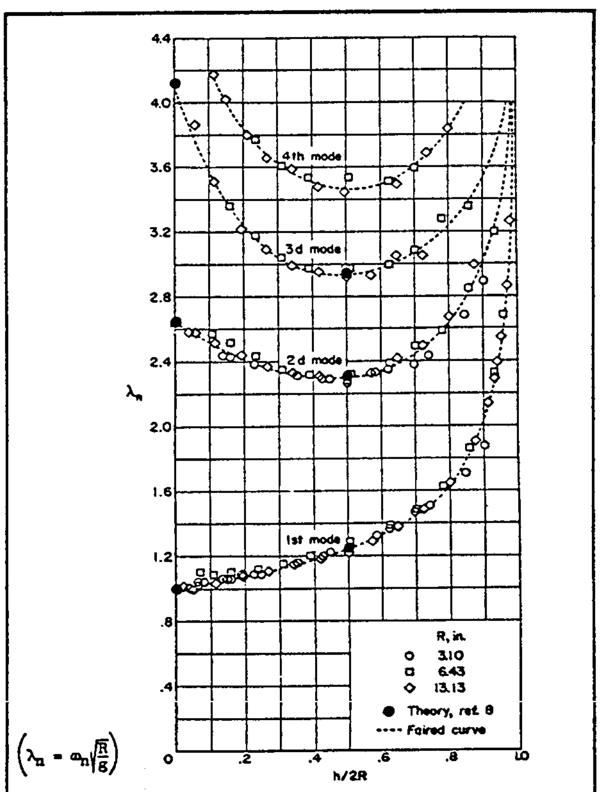
In the following, a spatial and mechanical model is described which utilizes insights gained from aeronautics and astronautics in arriving at the most suitable approach for arrival at a model for the respective truck-tank geometry in the direction of excitation. In this approach, movements of the liquid in longitudinal and lateral directions are initially considered separately. They are later superimposed.

The Pendulum Model for Transversal Oscillations

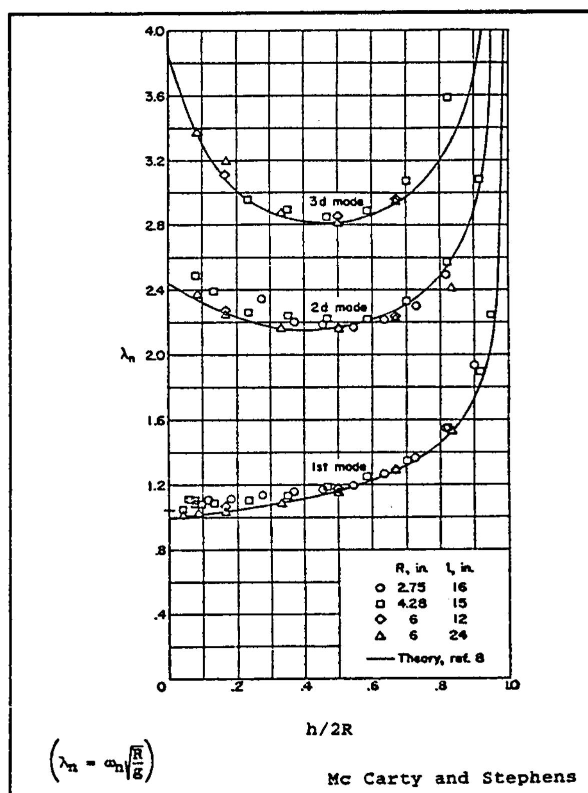
For transversal liquid oscillations in containers with rounded tank walls, the pendulum analogy has proved to be the most effective, as can be confirmed from the studies conducted by Sumner[14,15,and16], one of the pioneers in work with the "Pendulum Analogy." His work, in addition to elaborations subsequently made thereon by Sayar[17] with regard to nonlinear oscillations, form the basis for the model described here. Although Sayar and Sumner conducted their investigations on spherically shaped tanks, application of their findings to tank geometries featuring circular cross-sections is permissible, since the oscillatory behavior with typical tank-truck forms is sufficiently extensively similar to that of their containers. This fact has been additionally confirmed by analytical tests conducted by Budianski[18], as well as in model experiments performed by McCarty and Stephens [19]. See *Figs. 6 and 7.*

The natural frequencies of the liquid, which represent a characteristic value for the description of liquid motion, demonstrate a maximum deviation from each other of approximately 10% for a tank filling ratio of $h/d = 0.8$ (whereby h = the filling height and d = the tank diameter).

The model for transversal liquid motion consists of the fixed mass—designated by m_0, with a moment of inertia I_0—which does not take part in the liquid movement, the mathematical pendulum with the mass m_1 which represents the fundamental oscillation, a damping element with the damping factor K, and a spring with the nonlinear characteristic C for simulation of nonlinear factors which arise from the curvature effect (a force of restoration arising, in turn, from the curvature of the tank). See Sayar[17].



INSTITUT FÜR FAHRZEUGTECHNIK Prof. Dr.-Ing. H. Appel	FLUID FREQUENCY PARAMETER SPHERICAL TANK, LATERAL EXITATION, MC CARTY AND STEPHENS /19/	Fig. 6



INSTITUT FÜR FAHRZEUGTECHNIK Prof. Dr.-Ing. H. Appel	FLUID FREQUENCY PARAMETER HORIZONTAL CIRCULAR CYLINDER, LATERAL EXITATION /19/	Fig. 7

Fig. 8 defines the variables for the mathematical formulation of the pendulum analogy for the general case. The dimensions are explained as follows:

h_0	=	distance from center of tank to fixed mass
cg	=	center of gravity
C	=	geometric center of the tank
A	=	hinge point
l_{cg}	=	distance from the center of the tank to center of gravity
q	=	vehicle center of rotation
$l_{p,q}$	=	distance from hinge point of pendulum arm to vehicle center of rotation
l_p	=	distance from center of tank to hinge point of pendulum arm
L_p	=	length of pendulum arm
φ	=	angle from vertical through which pendulum oscillates
Θ	=	angular rotation about q
h	=	liquid depth
m_0	=	fixed mass (i.e., non-sloshing)
I_o	=	moment of inertia of fixed-mass (non-sloshing mass)

Fig. 9 depicts the external and effective forces as they act on the vehicle tank and the pendulum (cf. Sayar[17]).

The equation of motion for the pendulum results from the law of angular momentum for the accelerated reference point, as follows:

$$d\vec{L}_A/dt + m_1 [(\vec{r}_{AS} \times \ddot{\vec{x}}_A) + (\vec{r}_{AS} \times \ddot{\vec{z}}_A)] = \Sigma \vec{M}_A \qquad \text{(Eq. 1)}$$

where:

$\vec{L}_A$	=	the angular momentum about A
$\vec{r}_{AS}$	=	vector from the point of pendulum suspension to the center of gravity of the pendulum mass
$\Sigma \vec{M}_A$	=	sum of the moments about point A
$\vec{L}_A$	=	the angular momentum about A
$\ddot{\vec{x}}_A$, $\ddot{\vec{z}}_A$	=	Accelerations of the point of pendulum suspension

It follows that:

$$m_1 L_p^2\ddot{\varphi} + m_1 g L_p \sin\varphi + k L_p^2\dot{\varphi} + c\varphi^3 L_p = m_1 (L_p \ddot{x}_A \cos\varphi - L_p \ddot{z}_A \sin\varphi) \qquad \text{(Eq. 2)}$$

With $\sin\varphi \approx \varphi - \varphi^3/6$ and $\eta = 1/6 - c/m_1 g$ and the degree of damping $\vartheta = k/2m_1\,\omega_0$ it follows that the equation of motion of the pendulum is as follows:

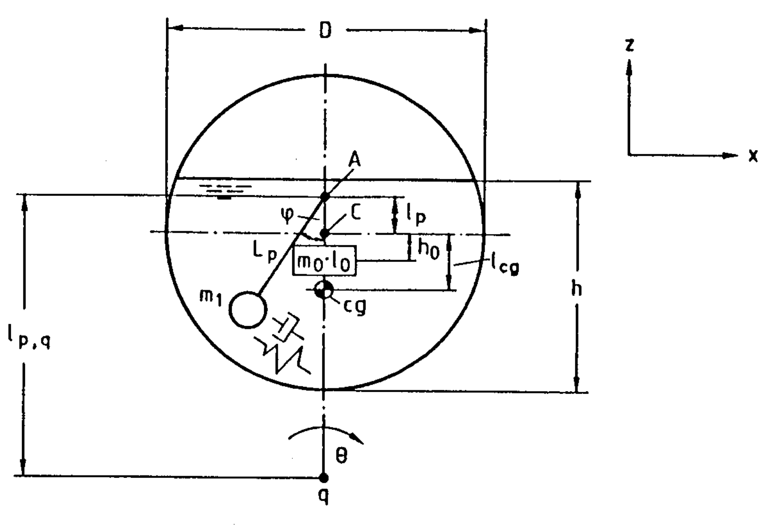
$$\ddot{\varphi} + 2\vartheta\omega_0\dot{\varphi} + \omega_0^2(\varphi - \eta\varphi^3) = 1/L_p\,(\ddot{x}_A \cos\varphi - \ddot{z}_A \sin\varphi) \qquad \text{(Eq. 3)}$$

In order to determine the forces from the liquid which act onto the tank, the sum of the external and the effective forces is found according to D'Alembert's Principle. See *Fig. 9*.

The following applies to the tank container:

Σ horizontal forces (external minus effective) = 0

$$- c\,\varphi^3 \cos\varphi - F_H - T \sin\varphi - k\,L_p\dot{\varphi} \cos\varphi + m_0\,\ddot{x}_0 = 0 \qquad \text{(Eq. 4)}$$

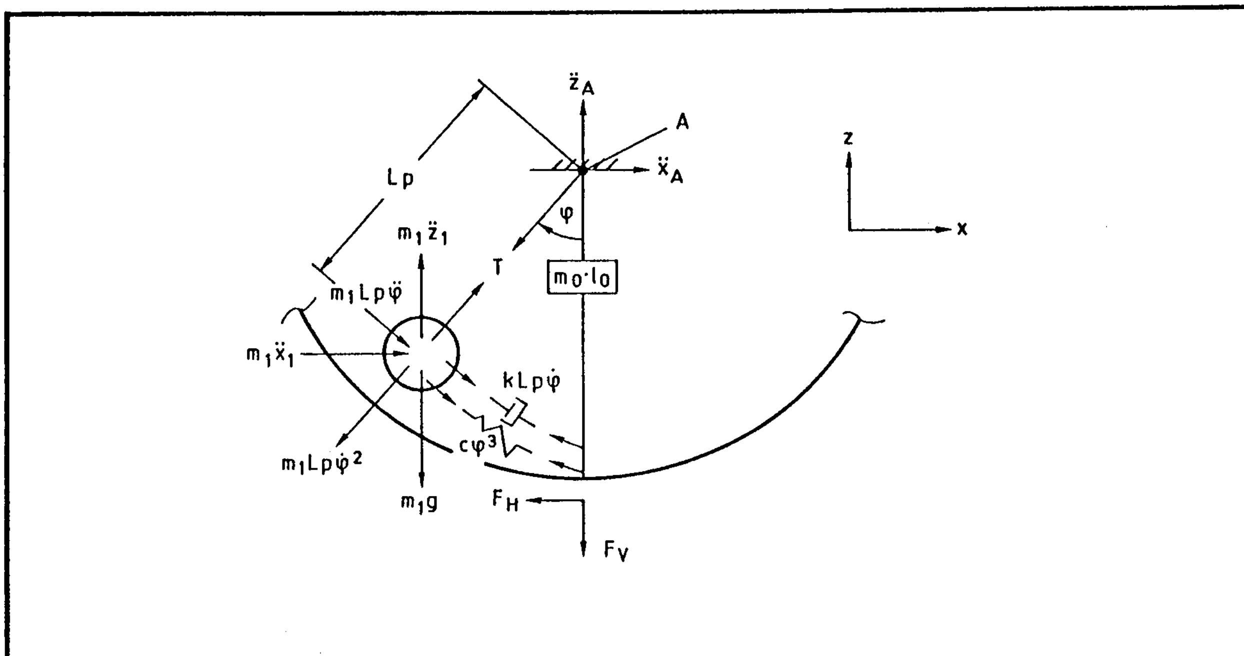


INSTITUT FÜR FAHRZEUGTECHNIK
Prof. Dr.-Ing. H. Appel

SCHEMATIC DIAGRAM OF A HORIZONTAL CIRCULAR CYLINDRICAL CONTAINER PARTIALLY FILLED WITH FLUID AND ITS ANALOGOUS PENDULUM ANALOGY FOR LATERAL LIQUID SLOSHING

Fig. 8

Stedtnitz



INSTITUT FÜR FAHRZEUGTECHNIK Prof. Dr.-Ing. H. Appel

EXTERNAL AND EFFECTIVE FORCES FOR THE PENDULUM AND THE TANK

Fig. 9

Stedtnitz

The following applies to the pendulum:

Σ horizontal forces (external minus effective) = 0

$$T \sin\varphi + k\,L_p\,\dot\varphi\cos\varphi + c\,\varphi^3\cos\varphi - m_1\,L_p\,\dot\varphi^2 \sin\varphi + m_1\,\ddot x_1 + + m_1\,L_p\,\ddot\varphi\cos\varphi = 0 \qquad \text{(Eq. 5)}$$

If equations 4 and 5 are added, the total force as exerted on the tank is as follows:

$$F_H = m_1\,\ddot x_1 + m_0\,\ddot x_0 + m_1\,L_P\,\ddot\varphi\cos\varphi - m_1\,L_p\,\dot\varphi^2\sin\varphi \qquad \text{(Eq. 6)}$$

It must be pointed out here that, in the event that the vehicle is rotated about point q (see *Fig. 8*), the accelerations $\ddot x_1$, $\ddot x_A$, and $\ddot x_0$ will depend on the angular acceleration Θ (see Sumner [14 and 16]).

The force F_H is the sum of the horizontal forces which are composed of the following: the forces of inertia of the masses, the horizontally acting bar force at the point of pendulum suspension ($T \sin\varphi$), and the damping and spring force acting at the level of the pendulum mass. The various points at which the forces act must be taken into consideration for formulation of the moments about the center of rotation q.

The approach for the vertical force is as follows for the forces acting onto the tank:

Σ vertical forces (external minus effective) = 0

$$c\,\varphi^3\sin\varphi - F_V + k\,L_p\,\dot\varphi\sin\varphi - T\cos\varphi + m_0\,(\ddot z_0 - g) = 0 \qquad \text{(Eq. 7)}$$

The following applies to the pendulum:

Σ vertical forces (external minus effective) = 0

$$T\cos\varphi - k\,L_p\,\dot\varphi\sin\varphi - c\,\varphi^3\sin\varphi - m_1\,(g - \ddot z_1) - m_1\,L_p\,\dot\varphi^2\cos\varphi - m_1\,L_p\ddot\varphi\sin\varphi = 0 \qquad \text{(Eq. 8)}$$

Now, when equations 7 and 8 are added, the following results:

$$F_V = -\,m_0\,(g - \ddot z_0) - m_1\,(g - \ddot z_1) - m_1\,L_p\,\dot\varphi^2\cos\varphi - m_1\,L_p\ddot\varphi\sin\varphi \qquad \text{(Eq. 9)}$$

For the event that roll and pitch motion can be neglected, then equations 6 and 9 can be simplified by taking $\ddot x_1 = \ddot x_0 = \ddot x_A = \ddot x$ and $\ddot z_1 = \ddot z_0 = \ddot z$.

Model for Longitudinal Oscillations

For the purpose of simulating longitudinal liquid motion in tank containers, application can be made of the comparison of this motion with motion in tanks displaying rectangular geometry. In both cases, the tank inner-wall surface at a right angle to the surface of the liquid is determining for the motion behavior of the liquid.

In accordance herewith, Bauer[2, p. 47] pointed out the extremely similar motions of liquid in longitudinally excited tanks displaying circular-cylindrical form and rectangular form, after studying frequency parameters in these two container configurations.

For the analysis of rectangular tank forms, studies, in professional literature have featured only spring-mass simulation models, including the associated parameters. *Fig. 10* shows such a spring-mass model designed in accordance with Dodge[20].

The model approach for the simulation of longitudinal sloshing in tank trucks can be maintained on a simpler basis, since the motion of liquids in tank containers with straight vertical walls is considerably more linear in nature than that in containers with curved tank walls (see Dodge[20, p. 205]).

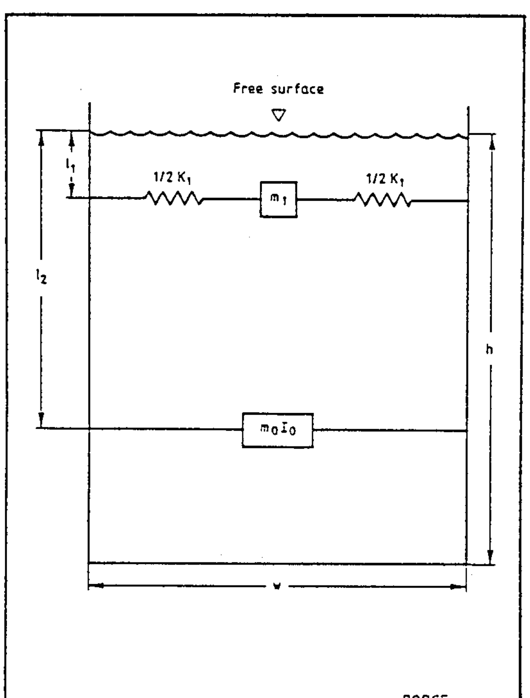
With this model, masses of higher order can be neglected, owing to their relatively slight share in the forces of the liquid involved.

In the model according to *Fig. 10*, a linear damping element must additionally be provided for simulation of viscosity or of baffles installed in the tank.

The equation of motion of the model represented in *Fig. 10* corresponds to the equation for a simple spring-mass-(damper) system.

Superposition of Longitudinal and Lateral Liquid Motion

The contents of the individual tank chambers of a tank truck are each represented here by one simulated liquid system. These systems consist of a pendulum model depicting lateral motion, as well as a spring-mass system to depict longitudinal sloshing, in accordance with the descriptions given in the two sections above.



SIMPLE MECHANICAL MODEL FOR A RECTANGULAR TANK /20/

Fig. 10

Now, if these two models are arranged perpendicularly to each other in a coordinate system fixed with respect to the tank, and if they are fixed in the geometric center of the tank, then they provide a spatial representation of the motion of the liquid. This representation, however, does not suffice to account for the mutual interaction of lateral and longitudinal sloshing, since the motion of the two models is independent of each other. In the same manner, assumption was made for the simulation systems in the two previous sections here, that the upper surface of the liquid is plane in the direction of movement not described in the respective model. This factor must be sufficiently taken into account when the two models are incorporated together.

For the model approach shown in *Fig. 11*, the attitude of the longitudinal and lateral simulation systems was for this reason maintained variable in the plane lying perpendicular to the direction of motion described by the respective model.

An example will make this clearer. When a tank truck begins to negotiate a curve, the liquid in the cargo container moves in a direction lateral to the outside of the curve, as shown at the top of *Fig. 11*. If the vehicle then decelerates, the liquid would move longitudinally toward for the direction of travel—whereby the initial situation provided for the longitudinal sloshing is represented by the laterally banked upper surface of the liquid. The longitudinal "sloshing force" would then act with lateral displacement, and no longer in the central geometrical plane of the tank. Consequently, the simulation model for the longitudinal movement must also be laterally displaced, as shown in *Fig. 11*.

Further experiments investigating the moments involved will be required to determine whether this lateral displacement of the longitudinal liquid model, as represented in *Fig. 11*, should be oriented to the lateral displacement of the center of gravity of the fluid.

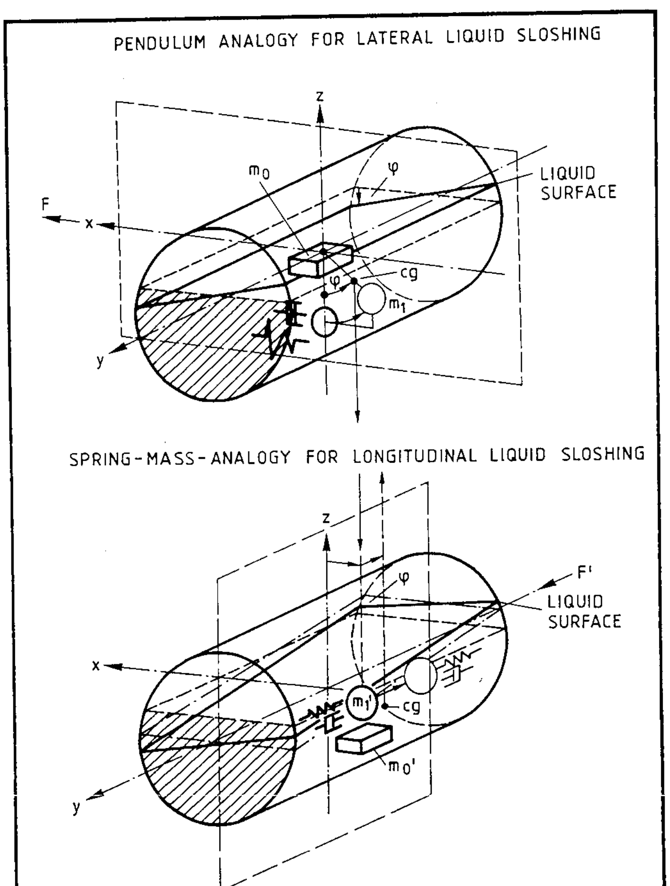
Now, the longitudinal displacement of the center of gravity determines the longitudinal shift of the lateral liquid model, since the lateral force of the liquid—owing to the longitudinal sloshing of the cargo—also no longer acts at the middle of the tank.

Mutual interaction between the two models, arranged perpendicular to each other as they are, is therefore involved in the manner described here. With displacement of the models as described above, forces of inertia should not be allowed to enter the contemplation of the systems in the direction of lateral displacement; rather, consideration may be taken here of only the constantly shifting points at which forces exerted by the liquid are applied.

In the case of the vertical force arising from the pendulum model (Eq. 9), it must be taken into account that this force was derived from the theoretical model approach for lateral liquid motion. Consequently, further experimental confirmation of the situation here is necessary. The point at which the vertical force is applied is not derived from superposition of the longitudinal and lateral model. For this reason, assumption should be made of the vertical force acting at the center of gravity of the liquid at rest.

Model Parameters

The parameters contained in the mechanical liquid model presented here can be described both analytically as well as experimentally, as Pfeiffer[21] and Sumner[14,15,and16] have elaborated. As far as the tank geometrical forms are involved here (rectangular and spherical), all of the parameters described in the model are quantitatively known and can be gleaned from professional literature[14,15,16,and20]. These parameters have been made available in non-dimensional form, with the result that procedures can be applied to geometrically similar tanks of any size.



INSTITUT FÜR FAHRZEUGTECHNIK Prof. Dr.-Ing. H. Appel | SUPERPOSITION OF LONGITUDINAL AND LATERAL LIQUID SLOSHING | Fig. 11 | Stedtnitz

Discussion

The liquid-cargo simulation model depicted here represents a further development of the mechanical liquid models taken from aeronautic and astronautic engineering applications—models which were originally conceived for calculation of the structural and flight stability of rockets. As a result, the question logically arises as to the validity of application of the model concept for tank vehicles.

In the answering of this question, the study presented here provides closer examination of the assumptions used as basis for the liquid simulation and the determination of model parameters involved in tank-truck efforts.

As has been demonstrated in the work performed here, assumption of the following is justified:

- absence of friction
- absence of cavitation
- vertical position of the upper surface of the liquid with respect to the pendulum.

In accordance with investigations performed by Strandberg[5, Vol. II, p. 86], friction forces in the liquid due to viscosity may be considered negligible in comparison to inertial effects.

Cavitation—above all, at such sharp edges as encountered at baffles, when longitudinal sloshing occurs—can arise and can lead to damage in the tank. Since cavitation acts for only very brief periods, however, its effects contributing to the influence of the liquid load on dynamic vehicle behavior can be neglected[5, Vol. II, p. 88].

Justification for assumption of the vertical position of the liquid surface with respect to the pendulum axis was confirmed in Sumner's model experiments[14]. In addition, confirmation has been provided in actual driving tests with a tank truck, in which the vertical position (referenced to the liquid surface of a physical pendulum mounted in a tank chamber) was indeed observed[22].

The effects of the following assumptions on the validity of the liquid model have not yet been able to be assessed:

- absence of rotation of the liquid
- small excitation amplitudes
- superposition of longitudinal and lateral sloshing movements.

The assumption of freedom of rotation is of necessity based on the determination of parameters in accordance with potential theory. In reality, however, the liquid does in fact rotate. This phenomenon can, possibly, be compensated for by the superposition principle.

The parameter values determined from the model experiments can vary according to the excitation amplitude of the tank. Since excitation amplitudes are small in aeronautic and astronautic study, and since great amplitudes of this nature are in fact encountered in motor-vehicle engineering, it will be necessary in the future to devote particular attention to this point of discussion.

The superposition principle proposed in this section must, in addition, be subjected to further study regarding the validity of its employment.

Answers still open to the questions raised here can most effectively be provided with the aid of model experiments and actual driving tests, the planning of which has already begun.

With regard to assessment of the hypothesis stated at the beginning of the paper—"Liquid Sloshing: a Possible Cause of Accidents?"—it would appear most advisable to await the results of testing performed on the Daimler-Benz driving simulator.

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Specialized Procedures for Preparing the Accident-Avoidance Potential of Heavy Trucks

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Abstract

The ultimate goal of the work described here is to improve the safety quality of the transportation of goods on highways. Even though the transportation missions of heavy freight vehicles have led to a variety of axle configurations, suspensions, dolly types, and articulation joints, this paper provides a condensed summary of fundamental mechanical properties that can be used to make first-order estimates of the braking and steering performances of these vehicles.

Specialized analysis procedures, based on basic properties of tires, suspensions, brakes, and steering systems, are discussed. These procedures provide simple analytical methods for predicting vehicle performance in maneuvers associated with operating truck combinations on highways. The performance characteristics considered are: (1) braking efficiencies, (2) transient low-speed offtracking, (3) high-speed offtracking in a steady turn, (4) directional and roll stability in steady turns, and (5) rearward amplification and roll stability in an avoidance maneuver.

The emphasis of this paper is on describing the features of simplified analytical procedures for estimating the accident-avoidance potential available to drivers for handling towing units and having trailers and semitrailers follow (track) without rolling over or exceeding pavement boundaries.

Introduction

Since 1970 the Motor Vehicle Manufacturers Association, the National Highway Traffic Safety Administration, and the Federal Highway Administration in the United States have each supported extensive research programs that have examined the braking capabilities, directional control and stability, tracking fidelity, and rollover limits of heavy trucks. These efforts have indicated that the heavy truck is a special class of vehicle requiring its own test procedures, analytical methods, and computerized models for evaluating braking and steering performance. As a result of these research efforts laboratory facilities are now available for measuring the mechanical properties of heavy trucks including those properties associated with mass distribution and the properties of truck tires, suspensions, and steering systems. Simulation models are now available for predicting the braking and steering responses of heavy trucks. These simulation models use mechanical properties measured in the laboratory to predict how vehicles will perform on the highway or in vehicle tests. Test procedures have been developed for demonstrating the performance capabilities of trucks in maneuvering situations in which some heavy vehicles are known to have very limited capabilities. The results of tests and simulations have been compared to verify that the model builders understand the observed phenomena. The current state of knowledge provides considerable insight with regard to maneuvering characteristics such as:

- low-speed offtracking (cornering in town)
- high-speed offtracking (turning at highway speeds)
- braking efficiency (constant deceleration braking)
- roll stability (rollover)
- steering sensitivity (handling)
- rearward amplification (obstacle evasion)

The maneuvering characteristics listed above have been selected for use in estimating how well vehicles will perform relative to the following practical goals that are, in essence, accident avoidance goals:

- the rear of the vehicle should follow (track) the front with adequate fidelity
- the vehicle should attain a desirable level of deceleration during braking
- the vehicle should remain upright (not roll over)
- the vehicle should be controllable and stable enough to follow a desired path.

With respect to these goals, the vehicle should be able to perform acceptably over appropriate ranges of operational factors including loading, speed, roadway friction, lateral acceleration, tire wear, brake maintenance, etc.

Based on the current status of knowledge concerning (a) the maneuvering performances of heavy trucks, (b) analytical procedures for predicting vehicle responses, and (c) experimental procedures for examining truck properties, researchers[1,2] have developed plans for guiding decisions on safety benefits. As illustrated in Figure 1, the item entitled "definitions of response phenomena" is the central factor in a research program that would provide information needed to weigh the tradeoffs between costs and benefits in determining the boundaries between acceptable and unacceptable performance. The definitions of pertinent response phenomena provide the foundation for (1) assessing the distributions of performance levels existing in the current truck fleet, (2) estimating the links between vehicle characteristics and certain types of truck accidents, (3) developing practical countermeasures for various types of trucks and truck accidents, and (4) developing test methods for demonstrating the performance capabilities of specific vehicles. These definitions are expected to portray response characteristics in selected maneuvering situations in terms of performance signatures (that is, graphs of performance capabilities), and performance measures (that is, distinguishing features of the signatures). (Sets of maneuvers, signatures, and measures for trucks have been discussed in previous publications[3,4].) The purpose of this paper is to further the definition of these response phenomena by describing basic mechanical considerations that have been incorporated into specialized procedures (simplified analysis-methods) developed for predicting performance levels using limited amounts of parametric data pertaining to the mechanical properties of heavy trucks[5].

Pertinent Mechanical Properties for Making First Order Estimates of Accident Avoidance Potential

The mechanical properties of the components of heavy trucks have been compiled into a "factbook"[6]. The influences of component properties on maneuvering performance are discussed, and

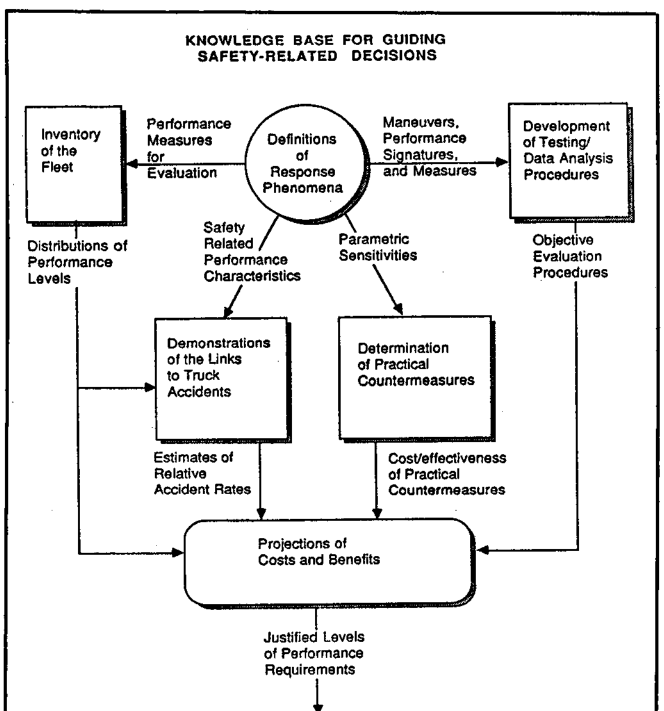

Figure 1. Knowledge base for evaluating safety requirements

tables indicating the relative importance of various mechanical properties to the dynamic performance of heavy trucks are presented there. The following discussions relate pertinent mechanical properties to the maneuvering situations listed in the Introduction.

Low-Speed Offtracking

The amount of low-speed offtracking for a particular vehicle depends upon its basic dimensional properties indicating where the axles are located (how far apart they are) and where articulation joints (hitches) are located. For simplified analyses, tandem and multiple-axle suspensions can be treated as single axles located at the centers of the suspension groups. These simplifications will not be accurate for vehicles with wide spread axles operating on slippery road surfaces, but otherwise they are sufficiently accurate for first order estimates of offtracking.

High-Speed Offtracking

The amount of high-speed offtracking depends upon those mechanical properties that influence low-speed offtracking plus pertinent mechanical properties describing the installed tires and the distribution of load. The needed information on load distribution can be supplied by specifying axle loads. For simplified analyses, the total load on tandem axle sets will suffice if the tandem's suspensions have load leveling mechanisms.

The information on axle loads is used in determining the angles that a vehicle's tires must assume to provide adequate side forces for the vehicle to follow highway curves at highway speeds. An additional piece of data needed for estimating these angles is the "cornering stiffnesses" of the tires. The angles involved here are "slip angles", and measured data on the lateral force capabilities of tires are usually presented as functions of slip angle and load. The cornering stiffness is the rate of change of lateral force with respect to slip angle evaluated at small slip angles and at loads specific to the vehicle's (the tire's) operating condition.

Braking Efficiency (Constant Deceleration Braking)

In addition to the axle and hitch locations and "static" loads as needed for the previous maneuvering situations, the heights of the centers of gravity of the units comprising a vehicle are required for analyzing braking performance. Also, the heights of the hitches are needed to determine the loads existing on the vehicle's axles when the vehicle is braking at a given level of deceleration.

To determine the friction level required to prevent wheels from locking up and vehicles from becoming directionally uncontrollable, the distribution of braking effort from axle to axle is important. Ideally, the instantaneous ratio of braking force divided by vertical load would be the same for all wheels—that way, no wheel would require more roadway friction than another wheel—this corresponds to the maximum efficiency for utilizing a prescribed level of roadway friction. However, the distribution of braking forces from wheel to wheel may not be capable of achieving high efficiency over the in-use range of road surface and vehicle loading conditions. Special equipment, such as antilock systems or load sensing proportioning systems, are needed to prevent wheel lock and to achieve good braking efficiencies. The proportioning arrangements provided by conventional braking systems have an important influence on braking efficiency, and in order to calculate proportioning, brake effectiveness (gains as functions of pressure) are needed for each brake.

Some of the load leveling mechanisms used in heavy trucks react to brake torques in a manner that causes interaxle load transfer between axles in tandem sets. This load transfer can be large enough to have a significant influence on braking efficiency.

Although brake effectiveness or gain seems like a straightforward matter, it is not. Brake effectiveness is strongly influenced by random variations in lining friction, brake wear, work history (temperature history), and adjustment and maintenance practices. Nevertheless, simplified analyses can provide first order estimates of braking efficiencies that are suitable for judging the relative performances of vehicles that are expected to have comparable levels of maintenance and braking experience.

Rollover

To make first-order estimates of a truck's rollover threshold (that is, the level of lateral acceleration at which a truck will rollover), it is necessary to treat the vehicle as an assembly of sprung and unsprung masses. Even so, the most important determinant of roll stability is magnitude of the ratio formed by dividing half of the track width of the vehicle by the height of the center of gravity of the sprung mass of the vehicle. The amount of lateral translation of the sprung mass is much less important than the ratio of track width to c.g. height, but it is still critical during rolling. Suspension roll stiffnesses and roll center heights are needed to predict this lateral translation, and also, to predict the roll restoring moments tending to keep the vehicle upright.

The vertical stiffnesses of the tires serve to react the suspension roll moments against the ground. The maximum amount of this restoring moment at any axle is limited to the product of half of the track width times the axle load. The distribution of roll stiffnesses and vertical loads from axle to axle determines those axles whose restoring moments will be limited as the vehicle approaches its rollover threshold.

Finally, the hitches have a significant part to play in the rolling process. The typical pintle hitch does not provide a roll restraint between the units that it connects, while fifth wheels and turntables do provide roll constraints. The analysis of combination vehicles employing pintle hitches requires individual roll analyses for each independently rolling section of the vehicle.

Steering Sensitivity (Handling)

The term "steering sensitivity" is associated with the amount of steering wheel angle (or front wheel angle) required to maintain a steady turn at constant velocity. It is a performance measure associated with "handling." Specifically, it refers to the amount of change in steering angle accompanying a unit change in lateral acceleration (that is, the inverse of the lateral acceleration gain with respect to steering inputs). When the steering sensitivity approaches zero for a particular vehicle, that vehicle is approaching a situation in which a small change in steering angle causes a large change in lateral acceleration. In this case, the vehicle is approaching divergent instability, and it will be more difficult to steer properly than other vehicles having larger steering sensitivities (that is, vehicles having less gain per unit change in steer angle).

Steering sensitivity depends primarily upon how the tires on the steered "towing" unit (either a truck or a tractor) are loaded and upon the cornering stiffnesses of those tires.

To the extent that the properties of towed units influence the vertical and lateral loads on the tires of the towing unit, the towed units influence steering sensitivity. With regard to steady turning, information on the properties of typical full trailers is not needed because their dollies are equipped with pintle hitches that do not apply significant vertical loads, lateral loads, or roll moments to the unit ahead of the dolly.

In addition to the static loading of the towing unit's tires, the amount of side to side load transfer at the various axles of the towing unit is important. This is because a large amount of side to side load transfer can cause a small but important reduction in the total cornering stiffness of the tires installed on an axle.

The factors mentioned above imply that in order to evaluate steering (handling) performance it is necessary to know all of the properties mentioned previously for the other maneuvering situations plus information concerning details of the influences of vertical loads on tire cornering stiffnesses.

Rearward Amplification

In combination vehicles, primarily those employing full trailers, the motion of the last unit can be a greatly amplified version of the motion of the leading tractor or straight truck. This amplified motion is analogous to "cracking the whip", and it is referred to as "rearward amplification." Rearward amplification occurs in rapid (emergency) obstacle avoidance maneuvers performed at highway speeds[7].

Examinations of (a) results from experimental studies[8,9] and (b) the equations of motion for articulated vehicles[10,11] indicate that the properties of both towing units and towed units have significant influences on rearward amplification.

The important mechanical properties of trailers as towed units are their wheelbases and their ratios of (a) the sum of all of the cornering stiffnesses of the tires installed on the trailer divided by (b) the mass of the trailer.

The important properties of towing units (be they trailers, semitrailers, tractors, or trucks) are (a) the distance from their center of gravity to the rear hitch, (b) the locations of their wheels, and (c) the cornering stiffnesses of their tires.

Forward velocity has a large influence on the amount of rearward amplification. Rearward amplification becomes important at speeds of approximately 45 mph (72 kph) for vehicles that are susceptible to it, and it becomes increasingly larger as speed increases.

Simplified Analytical Methods for Predicting Truck Performance

The following analysis procedures might be viewed as numerical equivalents of rules of thumb. They have been programmed for use on personal computers[5]. These calculation methods are based on specialized models pertaining to specific maneuvering situations. Detailed descriptions of various versions of these models will be published in forthcoming reports[5,12]. The following discussion is aimed at providing an understanding of the basic ideas used in these simplified analyses.

Low-Speed Offtracking

In this case the vehicle may be envisioned as an assembly of axles and hitches with specified locations. The basic principle involved here is that, at low speed near zero velocity, the wheel planes of the tire-sets will remain tangent to the paths of motion of the tire-sets. (That is, the slip angles will be zero.) To illustrate this idea consider a simplified unit of a combination vehicle in which a generic tire-set is responding to the motion of its leading hitch point (see Figures 2 and 3). A discrete approximation to the path of the tire-set can be obtained by the construction illustrated in Figure 3. The tire always remains a fixed distance behind the hitch point and it proceeds along a straight line segment that is determined by the discrete points describing the hitch motion and the last position of the tire. (In somewhat mathematical terms, the hitch point follows a general curve and the path of the tire is the "tractrix" of that general curve.)

Given the path of the leading hitch point and having determined the path of the tire, one can determine the path of a trailing hitch point at the rear of the unit.

In practise, the general curve of the center of the front axle of the vehicle is given. (That is, it is assumed that the driver steers to attain a desired path for the front axle.) In addition, the discrete points describing the path of the front axle are at small intervals—say 1 foot (0.3 m) apart. Furthermore, the path of each hitch point is first calculated using a numerical equivalent of the graphical construction illustrated in Figure 3, and then it is used as the general curve for determining the motion of the preceding unit and its rear hitch point.

High-Speed Offtracking

During low-speed offtracking, if the front axle follows a constant radius turn long enough (usually more than through 90 degrees of turn), the vehicle will eventually reach a steady condition with a fixed level of offtracking. This level of offtracking can be determined from the properties of right triangles since the tires operate at nearly zero lateral force, that is, at nearly zero slip angles.

When a vehicle negotiates a steady turn at highway speeds, its tires generate the lateral forces needed to make the turn. The tires must operate at non-zero slip angles (see Figure 4) to produce these lateral forces. The geometry of the steady turning situation at highway speeds differs from the low speed situation due to the non-zero slip angles developed at highway speeds. Figure 5 shows the influence of slip angle, α, on the geometry of a generic trailer. The equation given in the figure provides an approximation to the

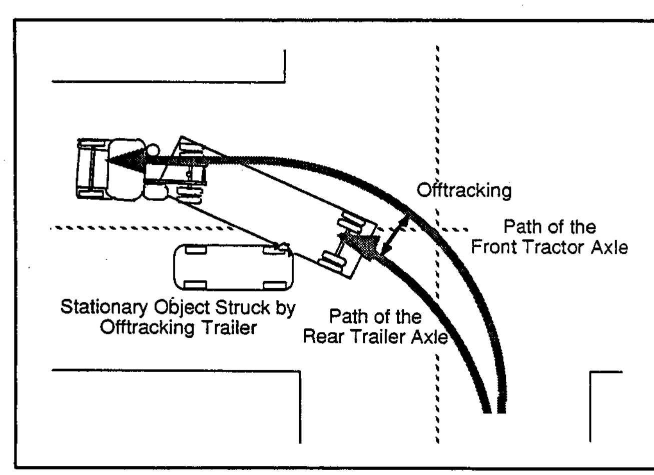

Figure 2. In low-speed offtracking, each axle tracks inboard of the preceding axle

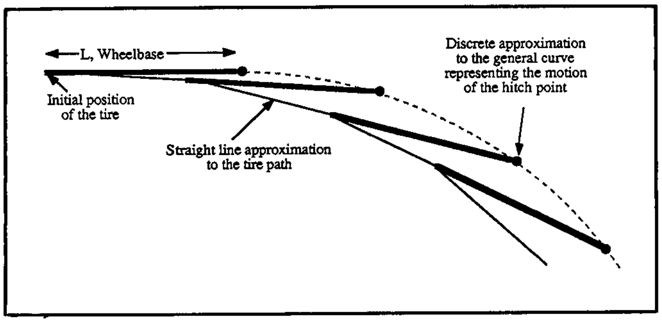

Figure 3. Offtracking behavior of a semitrailer following the motion of the fifth wheel kingpin

side force required from the tire set. Using this approximation, slip angle is found to be a function of cornering stiffness of the tires, vertical load on the axle, and the lateral acceleration of the turn. Once the slip angle is determined, the analysis is reduced to trigonometry; specifically, two applications of the law of cosines provides the radii of the wheel set and the rear hitch point if the center of the front axle is assumed to follow a curve of a given radius.

As illustrated in Figure 5, there is a speed at which the offtracking is zero. Above this speed the wheel tracks to the outside of the turn, and below this speed the wheel tracks to the inside of the turn reaching maximum inboard offtracking at zero speed. Typical heavy vehicles operating at highway speeds on highway curves tend to offtrack towards the outside of the turn by a small amount—on the order of 1 or 2 feet (0.3 to 0.6 m)—but enough to cause tripping on curbs if the driver is not aware of the position of the rear axles.

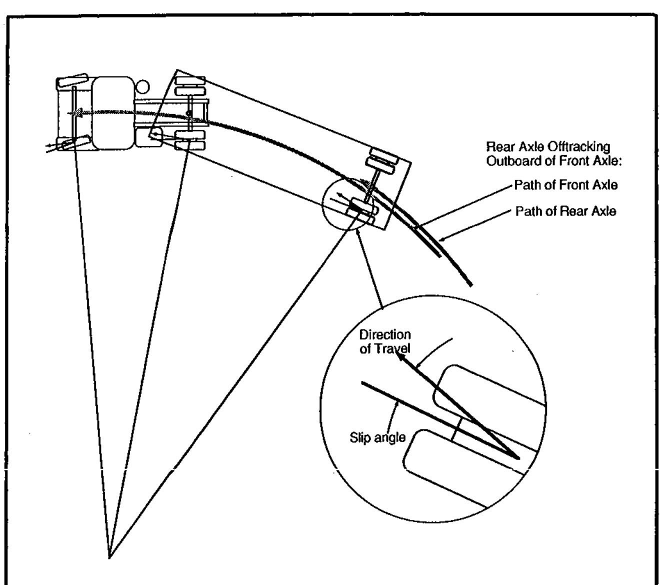

Figure 4. When cornering at speed, trailers may offtrack outboard of the tractor if the tire slip angles are large enough

Braking Efficiency

The braking capability of a vehicle can be estimated by analyzing its performance in a series of constant deceleration maneuvers. By assuming straight line motion, the complexities of vehicle roll and yaw can be neglected.

If the brake forces at the wheels are known, the vehicle's mass distribution helps determine its resulting deceleration. The pitching motion of the vehicle characterized by loads being transferred from the rear wheels onto the front wheels, can be represented in a straightforward sequential calculation starting from the rear of the vehicle. The pitch moment and force balances (see Figure 6) for each unit yield hitch and axle loads, which when carried forward to the power unit, would define the wheel load distribution for the vehicle. The amount of load that is transferred depends upon the deceleration level, the hitch and axle locations, c.g. heights, static load distribution and brake proportioning.

Special consideration is given to tandem sets, which redistribute their axle loads based upon a percentage of the braking effort generated at their wheels (see Figure 7). This interaxle load transfer is incorporated into the force and moment balance for each unit. The

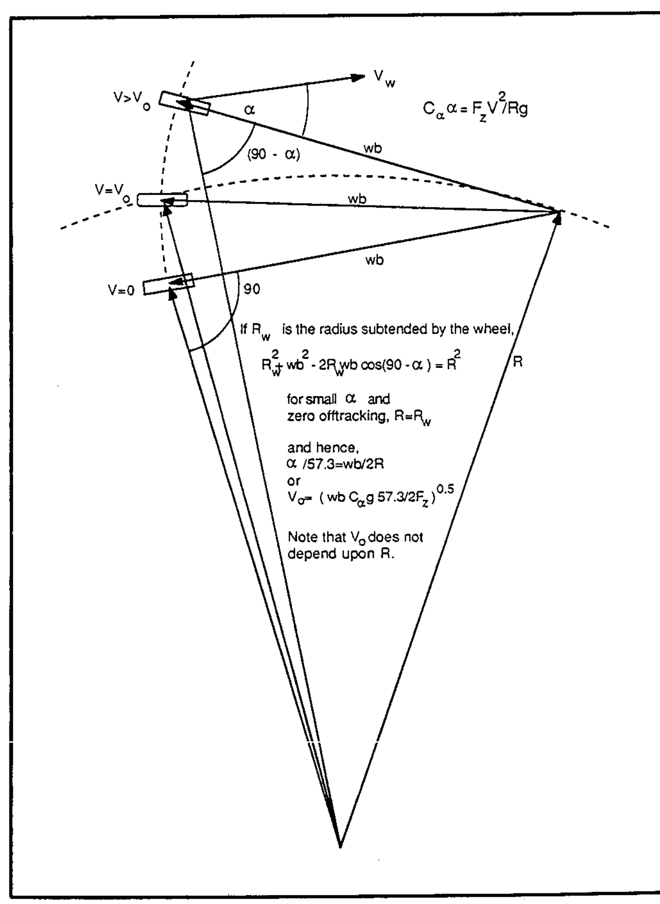

Figure 5. Illustration of V_0, the speed for zero offtracking

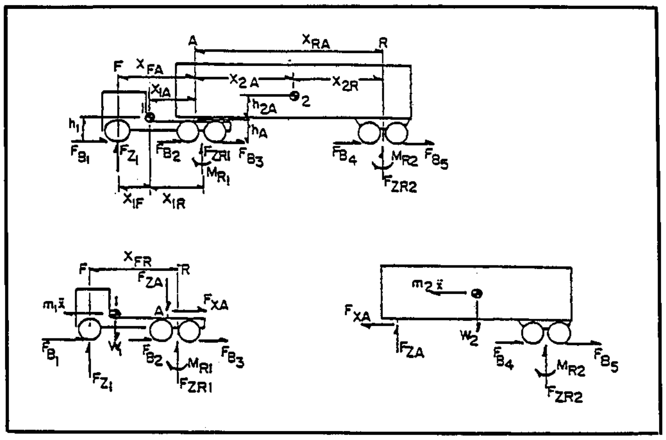

Figure 6. Dimensions and freebody diagram for a tractor semitrailer

analysis also accounts for full trailers with fixed dollies. Instead of transferring some load onto the forward unit, the turntable of the fixed dolly causes much of the pitching motion to be reacted out at the wheels of the full trailer. Besides altering the equations for the force and moment balances, the full trailer does not complicate the analysis.

The wheel loads determine the amount of road friction required to achieve the computed level of deceleration. The friction is given by, μ = Fx/Fz.

The axle whose wheels require the maximum amount of friction is most prone to wheel lock-up and is therefore the critical axle. The braking efficiency is defined as the ratio of the vehicle's deceleration to the friction required by the critical axle.

At higher levels of deceleration, rear-to-front load transfer increases. Wheel unloading caused by the load transfer increases the frictional demands of the vehicle resulting in a reduction in the braking efficiency. Braking efficiency can therefore be portrayed as a function of deceleration. In addition to deceleration, braking efficiency is very sensitive to the vehicle's static loading condition. Empty vehicles tend to have low efficiencies implying the dangers of locking wheels and losing directional control.

Roll Stability

The conditions for roll equilibrium are determined in this analysis. The vehicle is envisioned as proceeding through a sequence of roll angles of its sprung

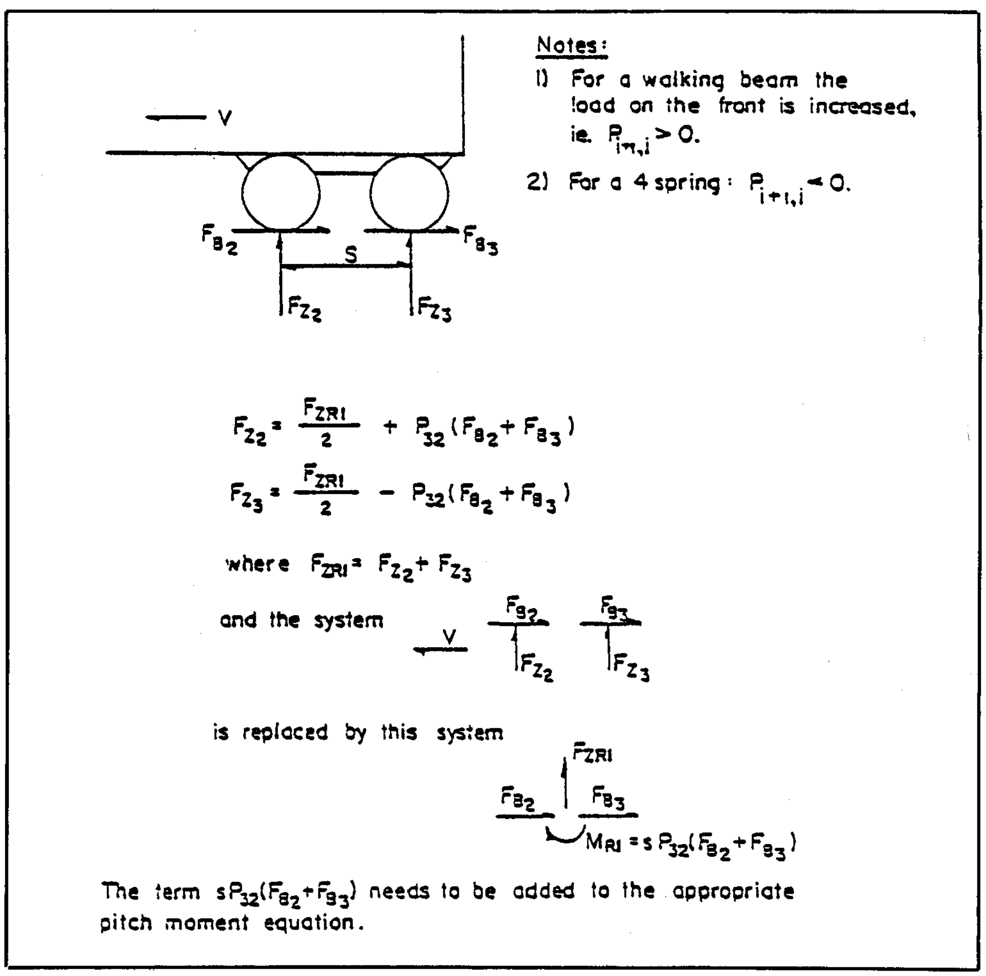

Figure 7. Approximate representation of interaxle load transfer

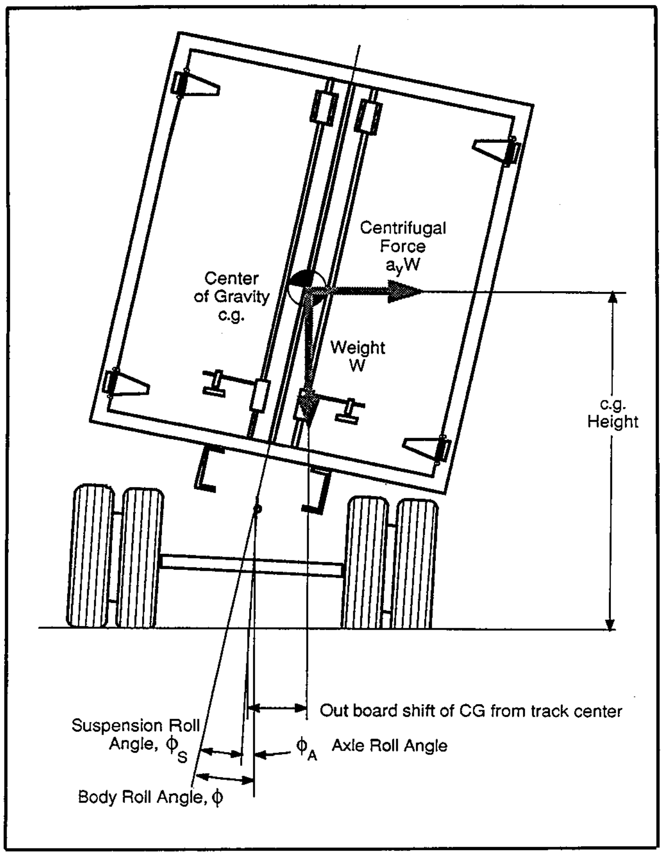

Figure 8. A heavy truck in a left turn

mass. For each of these roll angles, the equilibrium equations of motion are solved for lateral acceleration, the roll angles of each axle, and the relative roll angles between the sprung mass and each of the axles. (See Figure 8.) These angles determine the restoring moments produced by the suspensions and the tires. The conditions for wheel lift-offs are determined and the equations are adjusted to account for the "saturation" of restoring moment accompanying lift-off. As the suspensions and tires deflect the mass centers translate to the outside of the turn, thereby increasing the moment tending to produce rollover. The results of this analysis are examined for the maximum level of lateral acceleration that can be sustained. This is the rollover threshold.

Steering Sensitivity (Handling)

This analysis is based on the equilibrium equations for steady turns at constant velocities. The required front wheel steering angle is calculated as a function of lateral acceleration. In addition, if the vehicle can become unstable at high speeds and lateral accelerations, the boundary between stable and unstable operation is determined. This boundary is displayed on a graph with lateral acceleration as the horizontal axis and forward velocity as the vertical axis, thereby indicating those combinations of lateral acceleration and forward velocity for which the vehicle will be stable.

Examinations of the equations of motion for articulated vehicles show that during a steady turn there is a single value of yaw rate that applies to all units in the combination vehicle—otherwise, the articulation angles would be changing and the motion would not be a steady turn. This observation leads to a procedure in which lateral acceleration is specified at each step of a series of computations made at a constant velocity. (In a steady turn, lateral acceleration is the product of velocity multiplied by yaw rate.)

The general form of a step of this procedure is illustrated in Figure 9. Applicable notation is defined by the figure. The items shown in this figure can be used to explain the basic ideas involved in the calculation. First, the force of constraint acting at the front of the last unit can be calculated from properties of the last unit and the selected values of yaw rate, r, and forward velocity, u. Next, since the force of constraint at the front of one unit has an equal but opposite reaction on the next unit, the force at the rear of the next unit is known. As indicated in the diagram, the force of constraint at the front of any unit can be determined if (a) the force at its rear hitch point, (b) yaw rate, and (c) forward velocity are known. The analysis proceeds in this manner going from unit to unit towards the front of the vehicle. Once the analysis reaches the forward unit (the tractor) the amount of steer angle needed to obtain the required force at the front of the first unit is determined, thereby providing the information desired for predicting the steering sensitivity aspects of handling performance.

The forces of constraint contain information that is useful in understanding the handling performance of

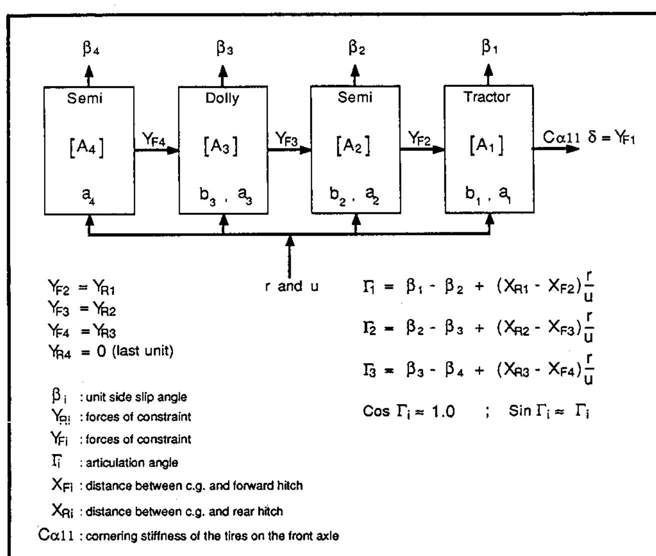

Figure 9. The elements involved in calculating steering angle, δ

certain vehicles. For vehicles with full trailers the constraint force at the front of the dolly is usually very small so that changes in the rearward units have little influence on the steering response of the tractor (or truck). For tractor semitrailers, the lateral force of constraint at the fifth wheel is approximately equal to the lateral acceleration in g's times the static load on the fifth wheel. This piece of knowledge can be used to simplify the analysis of the tractor semitrailer, or at least, to check results from more complicated calculations. (See Appendix A for further discussion of Figure 9.)

Rearward Amplification

In a full trailer, the conventional dolly provides a wagon tongue type of steering that causes the center of gravity of the trailer to follow the path of the hitch point with excellent fidelity in normal lane changing maneuvers. However, for sudden obstacle avoidance maneuvers with steering inputs having periods on the order of 2 to 3 seconds, the motion of the trailer's c.g. can be an exaggerated version of the motion of the hitch point. As explained earlier, the severity of this tendency depends upon the mechanical properties of the trailer and its towing unit.

In a surprise avoidance maneuver, the full trailer is like a mechanical servomechanism in which the motion of the pintle hitch is the input and the motion of the trailer's c.g. is the output. The magnitude of the input in the avoidance maneuver depends upon the properties of the unit towing the full trailer. The magnitude of this input depends also upon the period of the maneuver. To find the maneuver producing the largest amplification, it is necessary to compute results using a set of periods covering the range in which maximum amplification is expected. Figure 10 illustrates the type of result that can occur at a high level of amplification.

In order to avoid performing a number of simulations to evaluate rearward amplification, another approach has been used to obtain an estimate of the tendency towards rearward amplification. This approach consists of evaluating the linear range transfer function between the lateral acceleration of the leading unit and the lateral acceleration of the trailing unit. Although not as accurate as the simulation approach, the approach using the transfer function allows one computation to be used to find the maximum amplification.

Furthermore, the overall transfer function can be approximated by products of intermediate transfer functions relating the motions of c.g.'s to the motions of hitch points and then the motions of hitch points to the motions of c.g.'s in proceeding from the front to the rear of the vehicle. This approximate "factoring" of the transfer function is possible because the forces at the pintle hitches are small enough to be neglected. This helps to identify which portions of the overall transfer functions are the important contributors to rearward amplification, and thereby aids in identifying where improvements can be made by changing vehicle design.

Summary and Concluding Remarks

Clearly, a vehicle's tires are the connections to the roadway that are used to achieve the driver's transportation objectives. The tires provide (a) forces that cause the vehicle to move longitudinally, laterally, and directionally plus (b) the forces that support the load and hold it upright. The qualities required for maneuvering heavy trucks and avoiding accidents in braking and steering situations are dependent upon the following tire-related matters:

- where the tires are located on the vehicle
- how the tires are loaded
- how the tires are braked
- how the tires are oriented with respect to their direction of motion

Specifically, this paper indicates that values for the following mechanical properties are needed to make first order estimates of the braking and steering performance qualities of heavy trucks:

—locations of the axles
—locations of the hitches
—cornering stiffnesses of the installed tires
—static axle loads that the vehicle exerts on the pavement
—effectiveness of the brakes installed on each axle
—interaxle load transfer in tandem suspensions

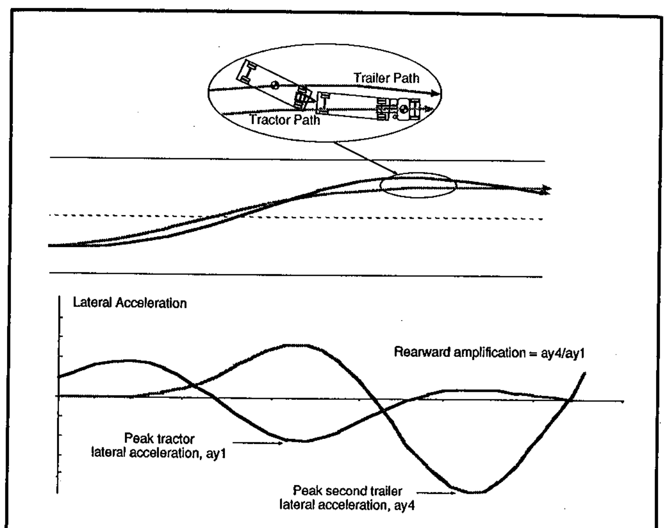

Figure 10. In a rapid lane change maneuver, rearward amplification results in "crack-the-whip" action of the rear trailer, sometimes resulting in rear trailer rollover

—weights and center of gravity heights of the sprung and unsprung masses of the units comprising the vehicle
—suspension roll stiffnesses (or spring stiffness and spread plus auxiliary roll stiffness)
—roll center heights of each suspension
—vertical stiffnesses of the tires
—track width between the centers of the tire sets installed on each axle
—types of constraints provided by the hitches used to couple units together
—influences of vertical load on the cornering stiffnesses of the installed tires
—compliance in the steering system and the steering ratio

Each of these mechanical descriptors has a bearing on how the tires are operated and how heavy trucks will perform in braking and steering maneuvers.

The analysis procedures described in this paper are aimed at evaluating the following performance measures:

- offtracking at in-town intersections
- offtracking on highway ramps
- friction utilization and braking efficiency during deceleration
- the lateral acceleration threshold at which trucks rollover
- the steering sensitivity and stability of tractor semitrailers, single unit trucks, and the leading units of multi-articulated vehicles
- the rearward amplification of the lateral acceleration from the first to the last unit of articulated vehicles.

This list of maneuvers could be expanded to include matters such as:

- response times in sudden turning situations
- braking during turning
- responses to external disturbances (wind gusts, road bumps, etc.)
- brake fade during mountain descents
- frictional requirements for negotiating turns on slippery road surfaces

Nevertheless, the set of performance measures discussed herein provides a starting point for safety improvement programs of the type illustrated in Figure 1.

It is intended that the understanding provided by describing and discussing simplified models will assist vehicle designers, accident investigators, and highway designers in making informed judgements concerning the accident avoidance potential of various types of heavy trucks.

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Appendix A

The quantities symbolized in square brackets (in the form $[A_i]$ in Figure 9) contain combinations of tire cornering stiffnesses and axle locations having relevance to handling considerations. (They also contain terms dependent upon the mass of the unit and its forward velocity.) The important terms related to the tires and their locations are (1) the "damping in sideslip", that is, the sum of the cornering stiffnesses of all of the tires located on the unit divided by the forward velocity, (2) the "damping in yaw", that is, the sum (over all tires) of the cornering stiffness of each tire times the square of its longitudinal distance from the unit's center of gravity—all of this divided by velocity, and (3) the "coupling between sideslip and yaw", that is, the sum (over all tires in front of the center of gravity) of the cornering stiffness of each tire times its distance from the center of gravity minus the same sum for all tires behind the center of gravity of the unit—again all of this divided by velocity. Without enough damping, vehicles may either sideslip or yaw excessively to attain the forces required for equilibrium. For tractors and trucks, the coupling between sideslip and yaw is usually small (that is, it is the difference of large, nearly equal numbers) but nevertheless it is important. If it is greater than zero for the towing unit, there exists the possibility that the vehicle may be directionally unstable at highway speeds.

The discussion above is independent of the roll motion of the vehicle. The vertical loads on the tires depend upon the lateral acceleration of the turn and the roll motions of the units. The cornering stiffnesses of the tires depend upon their vertical loads, and hence upon the roll properties of the vehicle. Through this mechanism, the amount of change in cornering stiffnesses is large enough to make some heavy trucks directionally unstable at lateral acceleration levels that are less than the rollover thresholds of these trucks.

Another factor of importance is the stiffness of the steering system. This stiffness is effectively in series with the cornering stiffness of the front tires, thereby increasing the amount of steering wheel angle needed for a specified level of front wheel angle. This is equivalent to reducing the cornering stiffnesses of the front tires.

Returning to Figure 9, the a's and b's represent the influences of the hitch locations and the influences of the forces of constraint on the force and moment balances pertaining to each unit. The distance from the hitch to the center of gravity of a unit indicates the leverage that the associated force of constraint has on the yaw motion of the unit. Short lever arms to the rear hitch points of dollies generally mean that the force of constraint at the front hitch point will be small. In addition, for semitrailers, if the rear hitch point is located near the "neutral force point", the force of constraint at the rear hitch has little influence on the force of constraint at the front hitch. The location of the neutral force point with respect to the center of gravity is given by the ratio of the "coupling between sideslip and yaw" divided by the "damping in sideslip." Although it may not be obvious in all cases, there are situations in which the steering sensitivities of the leading units of multi-articulated vehicles are not strongly influenced by the properties of the trailing units.

Finally, note that the sideslip angles (β's) of the motions at the centers of gravity of each unit are shown in Figure 9. As indicated by the equations given in the figure, the sideslip angles and the yaw rate can be used to calculate the articulation angles (Γ's). In addition to solving for the forces of constraint at the front of each unit, the equations of turning equilibrium can be solved simultaneously for the sideslip angles of each unit. Hence, this approach could be used for calculating high speed offtracking in a manner that includes the influences of spread as well as tandem axle sets.

Antilock Braking Equipment for Heavy Duty Vehicles and Its Evolutions Within European Regulation

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Abstract

Emergency braking of the heavy duty vehicles, especially the combinations, can be dangerous, particularly on wet or icy roads having a low coefficient of adhesion. Locking of the wheels having important consequences, loss of the trajectory control, jackknifing, slipping of the trailer, the equipment and car manufacturers have developed the antilock devices.

Some requirements exist since 1972 into the regulation 13 (ONU). The series fitting of the vehicles effectively started in 1980 and the experience permitted to complete these requirements in 1983 (EEC Directive 85/647).

The requirements ask to keep a good braking efficiency, a good stability, a low compressed air consumption and no wheel locking.

So the manufacturers have adopted different techniques to give the priority either to the braking efficiency or to the stability, select high, select low, independent regulation.

Furthermore, the compatibility between tractor and trailer shall be studied.

The wheel speed information available to the on-board electronics from the antilock system provides the basis for the future introduction of other functions such as antislip, retarder control, and speed limitation.

Today, these functions are separated but, for the future, it can be supposed that they all will be integrated either on the braking system or on the fuel injection.

Introduction

The ability of a vehicle to come to a stop over a short distance depends on the following three factors:

1) The time it takes the driver to react to an external stimulus,
2) The potential braking power of the vehicle,
3) The friction coefficient between the tires and the road surface.

Reaction time varies widely from driver to driver and thus cannot be considered to fall within the framework of regulation.

Braking power can be verified by a number of tests under load. Braking power cannot be put to effective use, however, unless it is carefully distributed to each of the axles and the friction between the tires and the road surface is sufficient to make use of it. If this is not the case, the stability of the vehicle can be severely impaired by wheel blockage on individual axles. The results can be a loss of steerability, spinning or, for trailer vehicles, jackknifing.

In order, to solve these problems, vehicle and equipment manufacturers have developed equipment that can detect the onset of wheel blockage and eliminate it by easing braking pressure.

Principles of European Regulation

These efforts at eliminating wheel blockage required that special stipulations be written into existing regulations to ensure that reasonably effective operation was maintained. One approach to the blockage problem, for example, was to considerably reduce braking pressure, which would not have been a satisfactory solution.

It was for this reason that in 1970, the United Nations Economic Commission for Europe established a set of regulations in this area. This text became Appendix 13 of Regulation 13.

The stipulations that were adopted cover the following points.

The addition of antilock equipment must not significantly diminish braking effectiveness either on high adhesion or low adhesion road surfaces. This must be verified for loaded and unloaded vehicles.

The equipment must be effective enough to prevent the wheels from blocking even during hard braking or when the vehicle passes from a high adhesion to a low adhesion road surface.

Antilock equipment that is used with braking-energy reserve systems must not itself consume significant amounts of power. Braking effectiveness must remain sufficient even after braking continuously for 15 to 20 seconds and applying the brake pedal four times in succession.

Finally, neither the antilock equipment nor the braking system itself should be adversely affected by electrical or electromagnetic fields.

Evolutions

These regulations enabled European manufacturers to market vehicles equipped with antilock equipment, which they did beginning in 1980. Four to five years of experience with these systems demonstrated the need for modifications and additions to the original regulations, and in 1985 a working group was formed to do this. This group published EEC Directive 85/647, which included a new appendix on testing antilock equipment.

The most important points of this new directive are as follows.

- Classification of antilock systems into three categories based on performance
  - Category 1: must ensure steerability, stability and effective braking on split surfaces
  - Category 2: must ensure steerability and stability on split surfaces
  - Category 3: must ensure stability on uniform surfaces with low as well as high adhesion coefficients.
- Introduction of a split surface test that enables antilock devices to be categorized as defined above.

These additions are such that the European countries are now prepared to require antilock equipment on the types of vehicles included in the following table.

Table 1.

Buses, GVW > 12 T	Category 1
Semi-trailer tractors GVW > 16 T	Category 1
Carrier capable of having a trailer and GVW > 16 T	Category 1
Trailers and semi-trailers GVW > 10 T	ABS on at least two wheels on each side

GVW : Gross Vehicle Weight

ABS Circuit Arrangements

Vehicle and equipment manufacturers have developed different techniques in response to different directives in the regulations. The solutions applied to motor vehicles have generally been the following (see Figure 1).

- One sensor and one brake pressure control mechanism per wheel. This system, called independent wheel control or individual regulating system (I.R.), offers the advantage of making the maximum use of the adhesion potential of uniform road surfaces. It enables the maximum braking force to be applied to each wheel. The disadvantage of this system is that it creates torque that tends to spin the vehicle on its vertical axis. When applied to steered wheels, it can generate parasitic steering-wheel torque that interferes with steering.

- One sensor for each wheel and one pressure controller for both wheels on the same axle. The pressure threshold for both wheels may set to the higher of the two friction coefficients read from the sensors: that is called select high regulating system (S.H.), but is more usually set to the lower of the two friction coefficients read from the sensors. This is called select low regulating system (S.L.).

The advantage of the select low regulating system is that brake drag is equalized, which ensures effective steering. The disadvantage is that it does not make use of the highest available friction coefficient on split road surfaces, with the result that it is less effective than the first system (I.R.).

- One sensor and one pressure controller per wheel, but with the additional ability to diminish the differences between braking forces, and thus drag, applied to each wheel when the vehicle is braking on a surface that offers various friction coefficients. This solution represents a compromise between the most effective braking and the best stability and this is called modified individual system (M.I.R.).

Heavy vehicle manufacturers have used these solutions in concert, applying different solutions to different axles. Some examples of what has been done (Figures 2 to 4).

The principles used to outfit motor vehicles with antilock have also been applied to trailers and semi-trailers. The customary configurations are as in figures 5 to 7.

Whichever solution is adopted for a motor vehicle, it always represents a compromise between braking effectiveness on one hand and steerability and stability on the other.

Further, when connecting a trailer or semi-trailer to a motor vehicle, differences in equipment performance can give rise to compatibility problems. A notable reflection of this can be seen in the efforts put into improving vehicle coupling. The compatibility problem, which has been clearly recognized by manufacturers, requires that special care be taken in choosing equipment for each of the two coupled elements.

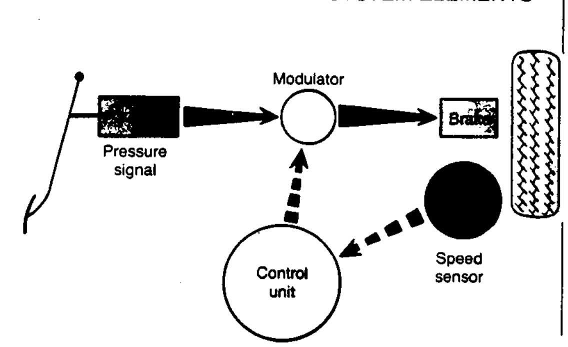
Beginning with the principle of antilock and the parameters used in its operation, equipment manufacturers have also been thinking of other vehicle functions that could be introduced.

The primary piece of information is wheel speed. What vehicle functions other than antilock device make use of this parameter?

1 - Indication of vehicle speed by the chronotachograph
2 - Speed limitation governor
3 - Drive slip control

The central nervous system of antilock and the functions mentioned above is the onboard processor. It follows that there is a possibility of making important progress in the areas mentioned, as the speed information needed for all these functions now usually comes from independent sources.

ELECTRONIC BRAKE PILOT SYSTEM ELEMENTS



Chronotachographs now operate through a mechanical or electrical connection to the gear box. The speed limitation governor, which acts on the fuel injection pump through a control linkage, gets its information on vehicle speed from the chronotachograph.

This information on speed, now used for the speed limitation function, could also be used for the antilock function.

This same information on wheel speed could also be used in anti slip systems to control the power being delivered to the drive wheels when skidding occurs. Control would be through a device similar to that now used with the fuel injection pump to limit speed. (See figures 8 to 10).

The speed information could be used to prompt a brief braking action on any found to be racing.

Finally, the retarder considered to be an indispensible extension to the braking system should not impede antilock operation. French manufacturers

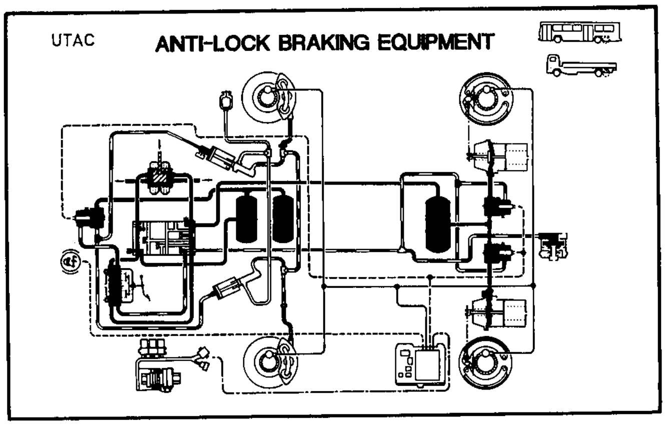


Figure 1

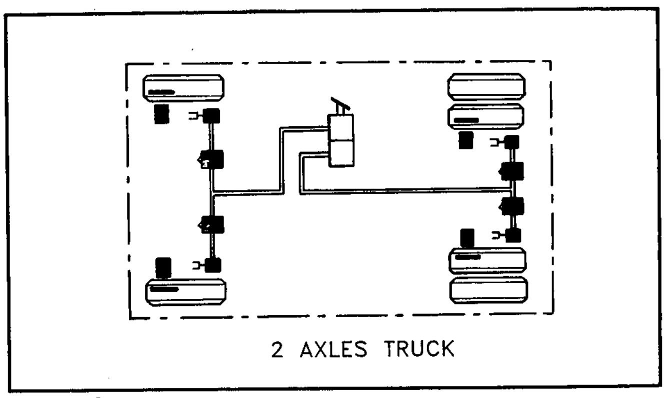


Figure 2

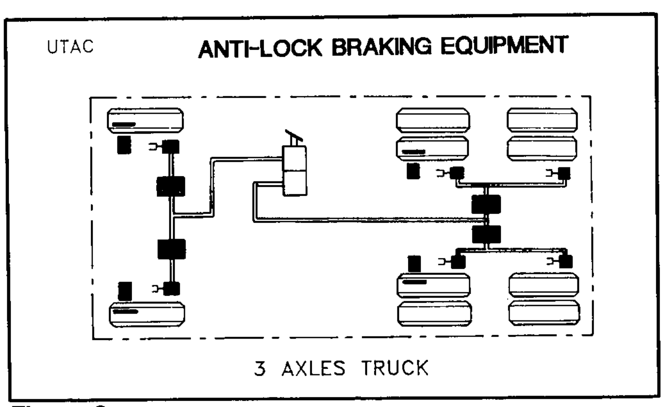


Figure 3

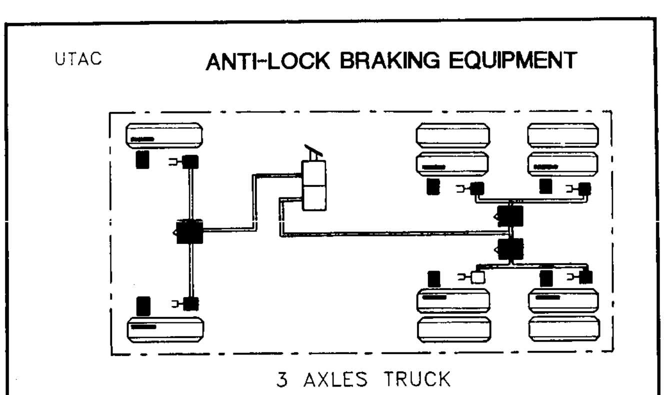


Figure 4

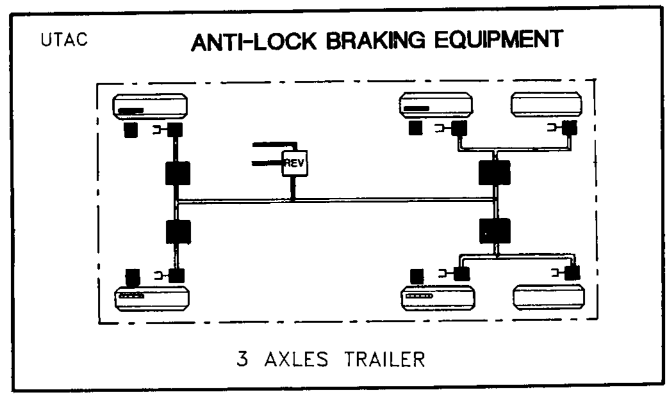


Figure 5

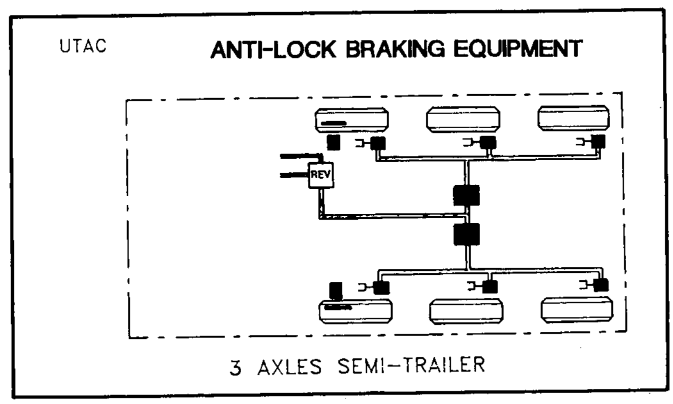


Figure 6

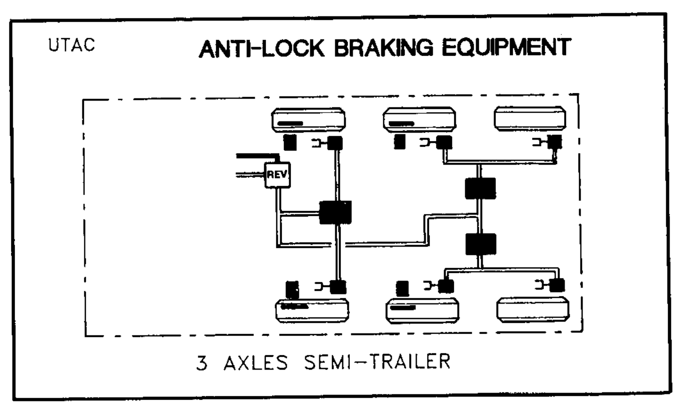


Figure 7

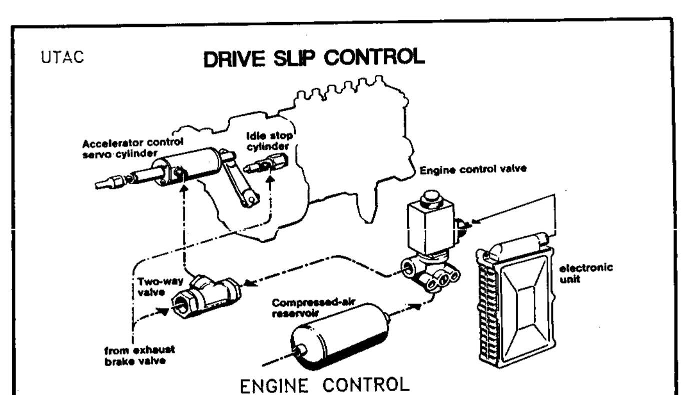


Figure 8

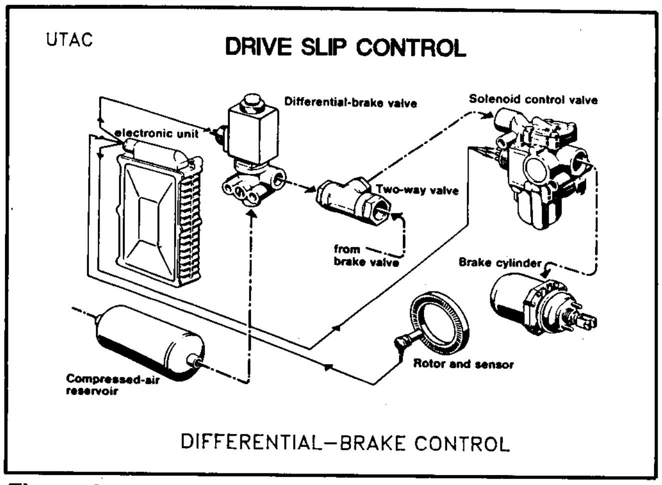


Figure 9

have done important research in this area, and the TELMA Corporation has developed an interface that links the retarder and antilock operations. This interface functions as described below.

When the antilock control processor is actively regulating braking through the antilock function, retarder operation is temporarily inhibited.

Once the processor ceases regulating, retarder braking power is gradually brought back to full capacity unless the processor intervenes with another command to regulate the braking functions.

This linkage makes it possible to prevent the retarder from degrading stability on slippery surfaces as well as to couple control of the retarder and brake operations.

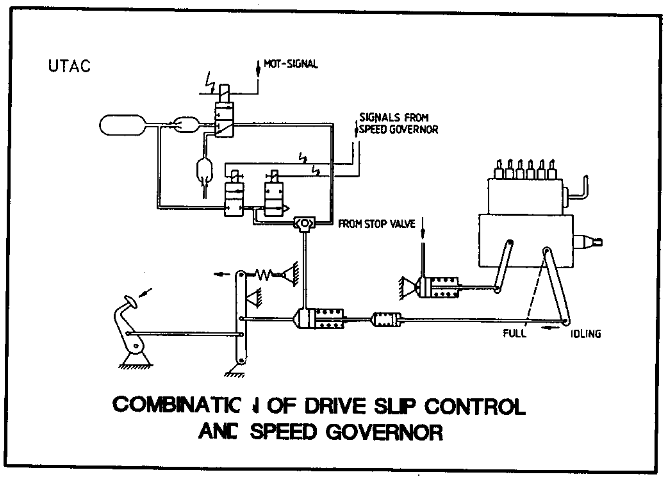


Figure 10

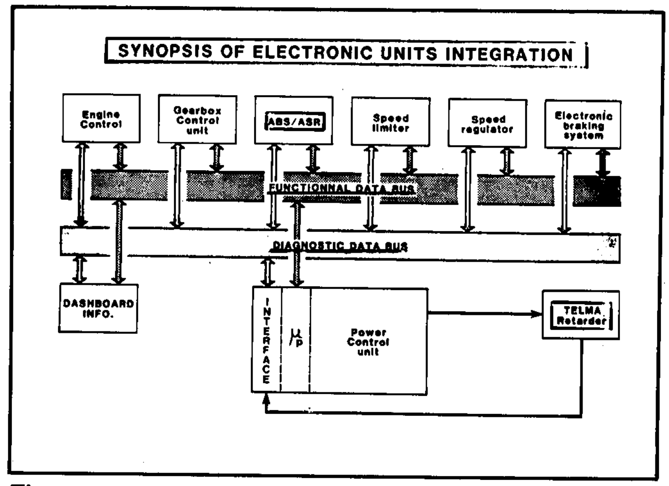


Figure 11

Conclusion

In the near term, it is also possible to imagine individual wheel speed sensors linked to a single electronic controller that would enable the following as they are needed:

- Speed information feedback for the driver
- Antilock braking and retarder equipment
- Speed limitation governor
- Drive slip control.

See figure 11.

The grouping of all these functions will allow a diagnostic on the functioning state and a detection of pannes.

See figure 12.

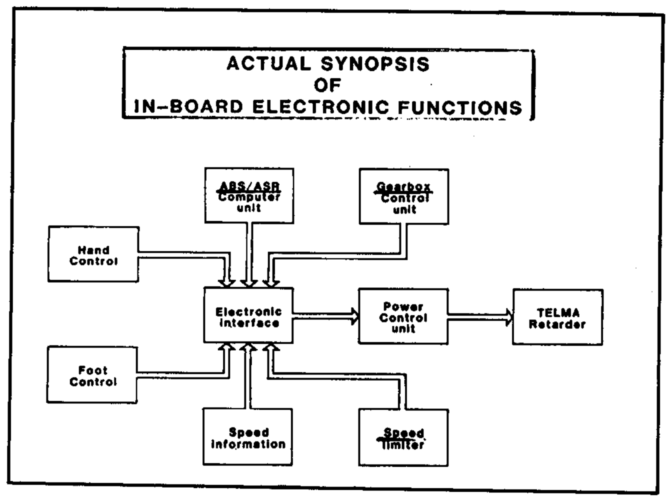


Figure 12

NHTSA'S Heavy Vehicle Brake Research Program—An Overview

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United States

Abstract

The National Highway Traffic Safety Administration (NHTSA) has been conducting experiments to assess the braking performance of heavy duty vehicles for a number of years. Many different types of vehicles have been tested and various aspects of braking performance have been evaluated. Experiments have included full scale vehicle tests on the test track, laboratory measurements on systems and components and dynamometer tests of brake assemblies and linings. Tests have included those which measure the status quo, as well as those which evaluate modifications and hardware designed to improve braking performance.

This paper provides a brief history of the Heavy Vehicle Brake Program, discusses major program areas and presents some of the significant findings and conclusions.

Program History

The National Highway Traffic Safety Administration (NHTSA) has been involved in research, testing and evaluation of heavy vehicle braking performance since the late 60's. NHTSA's first major effort was a large scale truck braking study conducted by the University of Michigan's Highway Safety Research Institute (HSRI). The objectives of this study, initiated in 1969 and completed in 1971, were to determine the braking performance levels exhibited by current design buses, trucks and tractor trailers and to establish the maximum braking performance capabilities of these vehicles when equipped with different types of advanced brake system hardware.

Much of the information developed in this study was utilized by the Agency in the formulation of Federal Motor Vehicle Safety Standard (FMVSS) No. 121, *Air Brake Systems*, which became effective for trailers on January 1, 1975, and trucks and buses on March 1, 1975.

In 1975 as production vehicles built to comply with FMVSS 121 became available, the Agency began to evaluate their performance and to compare it to that exhibited by vehicles built prior to FMVSS 121. A vehicle test program was established at the NHTSA's Safety Research Lab (SRL) in Riverdale, Maryland. SRL utilized the track facilities at the U.S. Army Proving Ground in Aberdeen, Maryland, for the necessary road tests. Also, as part of this program, SRL evaluated stability augmentation devices (including "anti-jackknife" devices) as possible alternatives to antilock brake systems. This initial program at SRL was the genesis of the current program at NHTSA's Vehicle Research and Test Center (VRTC) in East Liberty, Ohio; the SRL is now a division of the VRTC.

SRL's initial program was expanded to address various issues being raised as controversy surrounding FMVSS 121 began to grow. One of the major concerns within the trucking industry was the compatibility between the braking systems of pre and post FMVSS 121 vehicles. Many users reported that mixing pre and post FMVSS 121 vehicles in combinations (tractor-semitrailers, truck-full trailers, doubles, etc) resulted in degraded brake performance. In 1977, SRL tested a number of combination vehicles in various mixed configurations under a range of operating conditions on the test track. Many laboratory tests were also run.

In 1978, two significant events occurred: 1) the NHTSA moved SRL from its Riverdale, Maryland, laboratory to the Transportation Research Center (TRC) in East Liberty, Ohio, to become one of the two major divisions within the newly created Vehicle Research and Test Center (VRTC) and 2) the Ninth Circuit Court of Appeals issued a decision invalidating the stopping distance requirements specified in FMVSS 121. This eliminated the regulatory need to install antilock on heavy vehicles.

The move of SRL to Ohio slowed the progress of the program, but resulted in SRL having convenient access to extensive facilities ideally suited for heavy vehicle testing.

The court decision had a significant impact on the scope and direction of the program. The Agency wanted to study the performance of trucks, buses and trailers built to conform to FMVSS 121 when antilock systems were removed. Also, many different issues were raised during the time period that the standard was fully in effect that needed to be studied. With improved facilities available and many problems to address, NHTSA established the Heavy Duty Vehicle Brake Research Program (as it exists today) in 1979. This program, over the years, has addressed many different subjects relative to heavy vehicle braking. Vehicle road tests as well as laboratory and inertia dynamometer tests have been run. In addition to the in-house research at VRTC from 1979 to the present, the Agency has conducted research on heavy vehicle braking systems through contracts with private firms. This work includes a three-year in-fleet study of automatic brake adjuster performance and reliability, and a study of the benefits of retarders for heavy vehicles.

The purpose of the discussion which follows is to identify the major subject areas that have been addressed in the NHTSA Heavy Vehicle Brake Program over the years, briefly describe what has been done in each area and report some of the more significant findings. The references at the end of the paper cover the program in more detail.

Stopping Distance and Stability During Braking

Stopping distance tests have been run on over 70 different heavy vehicles. This group of vehicles consisted of buses, single unit trucks and combinations including tractor-semitrailers, truck-full trailers, doubles and triples. Vehicles were tested empty and fully loaded in straight line stops as well as braking and turning maneuvers. Various surfaces from dry pavement to ice were utilized. Although some of the tests were run with the driver fully applying the brake control (i.e., panic application) without limitations on wheel lockup and skidding, most testing was done to evaluate how quickly vehicles could stop under full directional control. This required that the driver modulate the brake control to minimize the amount of wheel lockup that occurred during the stop. Vehicle testing has been performed with fully operational brake systems as well as with simulated failures in the brake systems. Detailed results of all of these tests are given in References 1-7.

Approximately one fourth of the vehicles tested utilized hydraulic brake systems; the rest had air brakes, the most common system for heavy vehicles.

The test results indicate that the stable stopping capability of the various types of vehicles can be ranked as follows:

Stopping Capability Ranking	*Vehicle Type*
1 (best)	Buses (empty and loaded)
2	Loaded Tractor Trailers
3	Loaded Trucks
4	Empty Trucks and Tractor Trailers
5 (worst)	Bobtail Truck Tractors

This ranking is essentially independent of road surface coefficient of friction and vehicle speed. The ranking applies to "typical" configuration vehicles in these categories in either straight line braking or braking while turning maneuvers.

Looking first at air braked vehicles (as currently manufactured), Figure 1 shows the range of stable stopping distances that might be expected from 60 mph on a straight dry road for the different types of air braked vehicles, assuming the brake systems are in good condition, burnished and fully adjusted. Performance of a typical passenger car is also shown in Figure 1 for reference.

Buses perform best, primarily because under most conditions their braking force distribution is close to the normal force distribution on their axles. This allows buses to achieve maximum utilization of the tire/road friction force available at both axles before wheel lockup occurs. In effect, the buses have close to "ideal" braking distribution under most conditions. The front to rear weight distribution in a bus generally does not change substantially in going from the empty condition to the fully loaded condition due to the uniform nature of the loading. In addition, dynamic weight transfer in a bus is low due to a relatively low center of gravity height/wheelbase ratio.

Loaded tractor trailers also perform relatively well during braking due to the fact that their braking distributions and axle normal force distributions are similar. They do not perform quite as well as buses, however, due to the fact that the percentage of braking on their front (steering) axles is less than ideal. Loaded trucks do not perform as well as loaded tractor trailers. They experience more weight transfer onto their front axles than loaded tractor trailers which causes the percentage of braking available at the front axles of the loaded trucks to be even further below ideal.

The stable stopping capability of empty trucks, tractor/trailers and, in particular, bobtail tractors is relatively poor. This is due to the fact that their braking systems, which are sized for the loaded condition and have fixed braking force distributions, produce too much braking at the rear (or trailer) axles. These axles decrease in weight by a much greater percentage than the front steering axles when the loads are removed. This results in premature lockup and a corresponding loss of lateral (side) force capability at the tires on the "light" axle(s) permitting vehicle instability at relatively low deceleration levels.

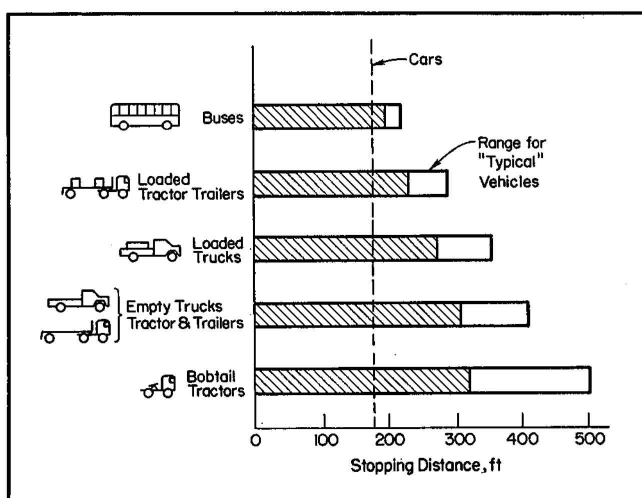

Figure 1. Stable stopping distance of heavy air braked vehicles from 60 mph on dry straight road

The problem is exaggerated if a vehicle has a short wheelbase and very lightly loaded rear axle which is why bobtail tractors exhibit the worst performance.

In general, most of the air braked trucks and truck tractors tested were found to be "under braked" on their front axles in that they would not lock up their front wheels before their rear wheels at any load level on any of the test surfaces including ice. In addition, several of the vehicles were equipped with front axle automatic limiting valves (ALV's) which reduce front braking substantially when control line pressures are low. Since low control line pressures are utilized when vehicles are empty, these valves further upset or degrade braking distribution in a situation where it is already considerably less than ideal. Complete removal or deactivation of the front brakes, a practice which is common among some truck users, obviously degrades the situation even further. The use of ALV's or the removal of front brakes increases the chance of drive wheel or trailer wheel lockup which can lead to spin-out, jackknife, or trailer swing.

Modification to test vehicles to increase the percentage of braking on the front axle such as removal of ALV's, increasing the size of brake chambers or installing variable brake proportioning systems (making braking distribution closer to ideal for straight line stops) resulted in optimum performance in the braking and turning case. Much shorter and more stable stops resulted in both cases. Increasing the front brake torque, however, did increase steering wheel pull when a vehicle was braked on an uneven coefficient of friction surface (difference in slipperiness left to right). This increase was insignificant if the vehicle was equipped with power steering and the steering axle had a low kingpin offset (scrub radius) but was quite significant when the vehicle had manual steering and a high kingpin offset. This indicates that steering system design must be taken into account if consideration is given to increasing front brake torque levels substantially above those which now exist. It may be necessary that vehicles have power steering and/or better steering geometry if greater levels of front brake torque are utilized.

Current model heavy hydraulically braked trucks were found to perform somewhat better than air braked trucks.* Figure 2 shows the relative performance of typical trucks with air and hydraulic brakes. Performance of the hydraulically braked vehicles is better primarily because they are typically designed with higher torque front brakes and achieve better braking force distribution particularly when empty.

*Class 6 and 7 single unit trucks and school buses are available from some manufacturers with either air or hydraulic brakes. Hydraulic brakes are standard on these vehicles and air brakes are offered as an option. Because of the additional complexity of the air brake system the cost of such a system is higher.

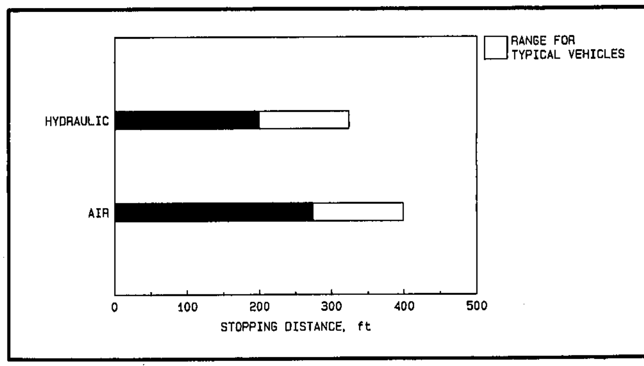

Figure 2. Relative performance of trucks equipped with air and hydraulic brakes—60 mph, dry road

One point that should be made about the above discussion of stopping capability is that it is based on the premise that brakes are in good working order, burnished and fully adjusted. If this is not the case, total brake force output may not be sufficient to produce a very high deceleration when the vehicle is loaded, even if the brakes are fully applied. With degraded output brakes, higher loads result in poorer performance, just the opposite of the situation as discussed above where the fully loaded vehicles out performed the empty vehicles.

Antilock

NHTSA tested a number of vehicles (five power units, four trailers and one converter dolly) with production antilock systems between 1975 and 1977. These tests described in References 2 and 3 were primarily designed to evaluate the braking performance gains provided by the antilock systems and to evaluate compatibility of tractors and trailers with and without antilock in various "mixed" vehicle combinations. Straight line stops as well as braking and turning maneuvers were run with both empty and loaded vehicles on surfaces ranging from dry asphalt to wet Jennite (pavement sealer). Although these tests did not specifically include an evaluation of reliability and vehicle test mileage was generally low compared to typical truck user mileage, the operation of a group of antilock equipped test vehicles did provide some insight into the problems being reported by the truck users. Component failures, electrical connector and wiring problems, intermittent failure warning light operation, etc were experienced on some test vehicles. When the systems were operating properly, however, braking performance gains were significant. Vehicles stopped shorter and were much more controllable with the antilock in operation. Antilock on the front axle prevented loss of steering, on the drive axle(s) prevented jackknifing and on the trailer axles prevented trailer swing. In terms of compatibility between vehicles with and without antilock there were no cases found where having antilock on one vehicle or one axle in a combination degraded performance. In fact, just the opposite was found. The application of antilock to any axle provided a braking performance improvement; the more axles that had antilock, the greater the improvement. This was found to be true with single unit vehicles, tractor semi trailers and doubles combinations.

After the Court struck down the stopping distance requirements in FMVSS 121 in 1978, very few trucks, trailers, and buses were built with antilock systems and very few users attempted to keep existing systems operational. Use and production of antilock essentially dropped to the negligible level in the U.S. In the meantime, however, interest in antilock began to grow outside the U.S., particualrly in Europe. Several European component manufacturers began producing second generation antilock systems and European vehicle manufacturers began offering them as optional equipment. Thousands of these systems are now in use in Europe and although no countries currently require antilock, several are considering issuing regulations mandating the systems on heavy vehicles.

Two fleets in the U.S. are currently operating a small number of vehicles with one particular brand of European antilock. In early 1986, one of these fleets made a two-axle straight truck available to NHTSA for testing at VRTC. Reference 8 provides a detailed description of the NHTSA test program on this vehicle. Figure 3 shows a summary of the test results and indicates the percent improvement in stable stopping distance that resulted when the antilock was operational in various types of braking maneuvers. Figure 3 is based on comparing the performance of a skilled test driver modulating the brakes on the vehicle without the antilock to the performance of the antilock system. Performance improvement with the antilock for more typical drivers would be expected to be even greater than that shown in Figure 3. In addition, Figure 3 assumes the driver has the presence of mind

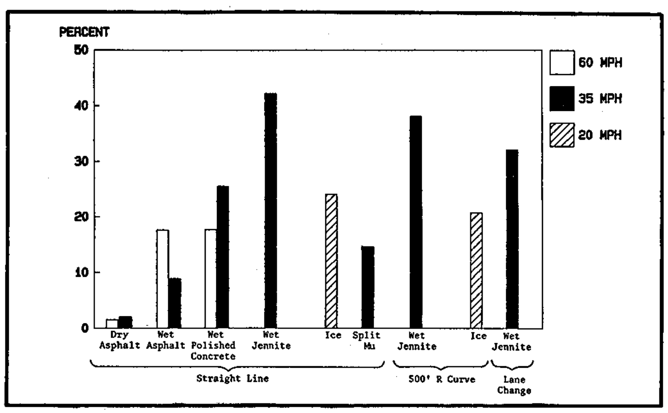

Figure 3. Percent improvement in controlled/stable stopping distance with antilock—fully loaded vehicle

to modulate the brakes if the wheels begin to lock. In an actual emergency on-road situation, this may not be the case: the driver may panic and lock the wheels skidding out of control.

In addition to this two-axle truck, VRTC has recently completed tests on a European tractor-trailer combination equipped with antilock and is currently in the process of testing a U.S. tractor-trailer combination with antilock. In the current tests, VRTC is evaluating the performance of a "standard" European system as well as modified versions of the standard system with different (simplified) control strategies. The purpose of these tests is to quantify the performance gains provided by the different control strategies. Individual wheel control, axle-by-axle control and tandem control strategies are all being evaluated.

In order to evaluate reliability of current antilock systems, NHTSA is planning to conduct (under contract) a two year fleet study of approximately 200 vehicles with various available antilock systems. This program will start in about a year. Action to find a suitable on-board data acquisition system has already been initiated. This data acquisition system will be a relatively simple device not unlike commonly used electronic tachograph and will monitor antilock function while the vehicle is in operation in the fleet. The NHTSA will also follow (via a contractor) experience with antilock in Europe as well as Australia where there are also many systems in-use. In addition, the NHTSA will attempt to obtain as much information as possible on antilock equipped vehicles operating in the U.S. that are not specifically included in the 200 vehicle fleet study.

Tractor and Trailer Compatibility

As mentioned earlier, compatibility between vehicles with and without antilock in combinations was studied by NHTSA in the late 70's. The subject of tractor and trailer brake system compatibility, however, is much broader than the issue of mixing vehicles with and without antilock. In fact, compatibility is a major area of concern today even though vehicles do not typically utilize antilock. In general, compatibility refers to the braking system on the tractor and the braking system on the trailer working together in harmony to provide desirable combination vehicle brake system durability and braking performance. It is primarily determined by the transient and steady state brake force distribution existing at the various axles of the combination vehicle.

Both of these aspects of compatibility have been addressed in depth in the NHTSA research program over the years. The transient brake force distribution is strongly influenced by the flow characteristics or timing of the pneumatic system. Most of the NHTSA research on pneumatic timing is described in Reference 9. The time that it takes for the pressure at the brakes to reach a particular level after the pedal is applied is known as the apply timing; the time that it takes the pressure to be reduced to a specified low level is known as the release timing. Overall apply time is important because it determines how quickly full braking force is achieved; this has an influence on stopping distance. Relative apply time for tractors with respect to trailers is also important because it affects the coupling or "kingpin" force between the two units during braking. High coupling force is undesirable because it aggravates the jackknife situation. If the trailer brakes apply slow with respect to the tractor brakes, the trailer will tend to "bump" the tractor harder. Figure 4 shows results of actual measurements made on an 80,000 lb combination during a panic stop on dry pavement using a semitrailer with an instrumented kingpin. It can be seen in Figure 4 that when the tractor brakes apply fast with respect to the trailer brakes, overshoot in kingpin force occurs. This overshoot reaches the same level as the case where the trailer brakes are not working (i.e., infinite trailer brake apply time). One point that should be made relative to Figure 4 is that even with the ideal case (trailer applies before the tractor) there is a substantial force at the kingpin.

Recent tests of typical late model vehicles(Reference 10) indicate that trailer brakes do not usually apply before tractor brakes. Figure 5 shows apply times (0 to 60 psi) for nine tractor-semitrailers and six doubles combinations.

Release timing is also important to vehicle stability although it has no effect on stopping distance. If a driver is applying his brakes in an emergency situation and locks the wheels, it is important that he be able to release the brakes as quickly as possible; otherwise he may skid out of control. Slow release times also affect brake temperature and wear.

Steady state brake force distribution is very important to compatibility. NHTSA research in this area is covered in References 4 and 10. In sublimit braking

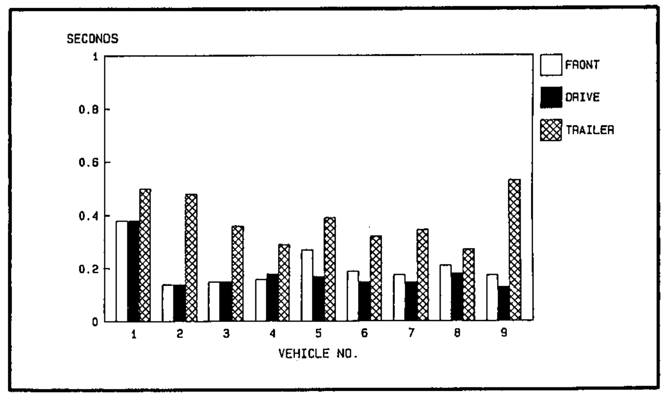

Figure 5a. Apply time at each axle set—nine tractor semitrailers

situations (i.e., well below the point of wheel lockup) brake forces must be balanced, otherwise excessive wear and temperature build-up will occur at the "overbraked" axle. In limit braking situations, wheels on overbraked axles will lock up and skid prematurely.

The input level at which braking force starts to occur at each brake, known as the brake force threshold pressure, is a very critical parameter to compatibility. If brake force at the tractor starts to occur at an input level below that needed for the trailer braking to start or visa versa, brake temperature imbalance is probable in repeated or continuous low pressure braking situations (such as mountain grade descents). If the vehicles are on low coefficient of friction surfaces, wheel lockup can occur prematurely.

Figure 6 shows the final brake temperature on a tractor and trailer at the end of a 5 mile long 4% grade descent at 45 mph as a function of threshold pressure difference between the tractor and trailer. Only a 2 psi difference in threshold pressure can make a difference of over 200°F in final brake temperature.

Figure 7 shows the effect of threshold pressure difference on braking efficiency by comparing a combination with equal thresholds to one when the

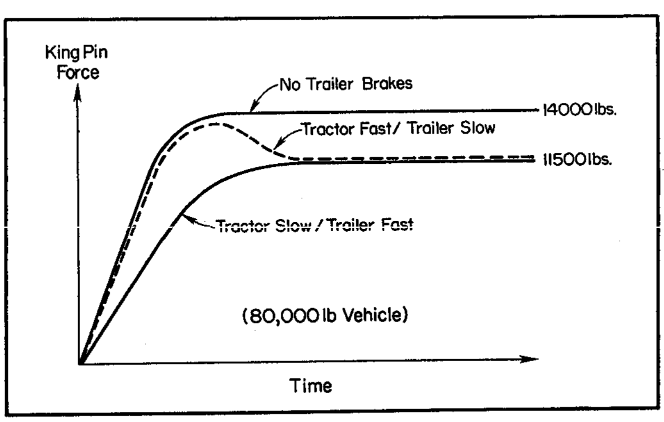

Figure 4. Kingpin force versus time for a panic stop

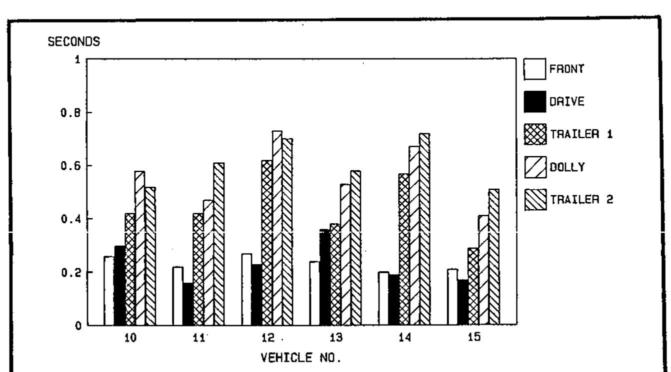

Figure 5b. Apply time at each axle set—six doubles combinations

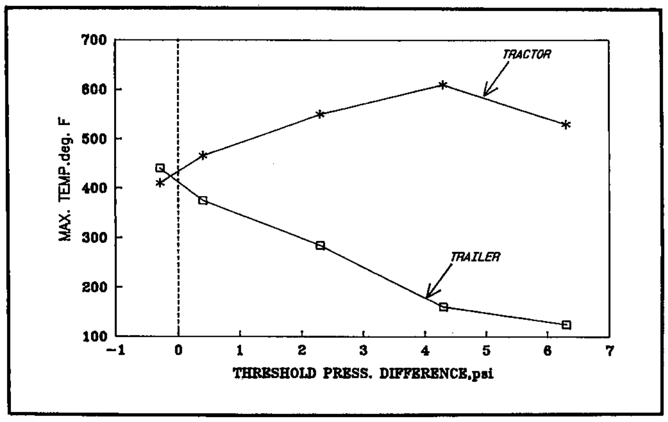

Figure 6. Brake temperatures for 4% grade simulation—five miles at 45 mph

trailer threshold is 4 psi lower than that on the tractor. On low mu surfaces particularly with the empty vehicle, the drop in efficiency that occurs when threshold pressure is different is quite significant.

Brake Adjustment

Both vehicle and inertia dynamometer tests have been run to determine the effect of adjustment on

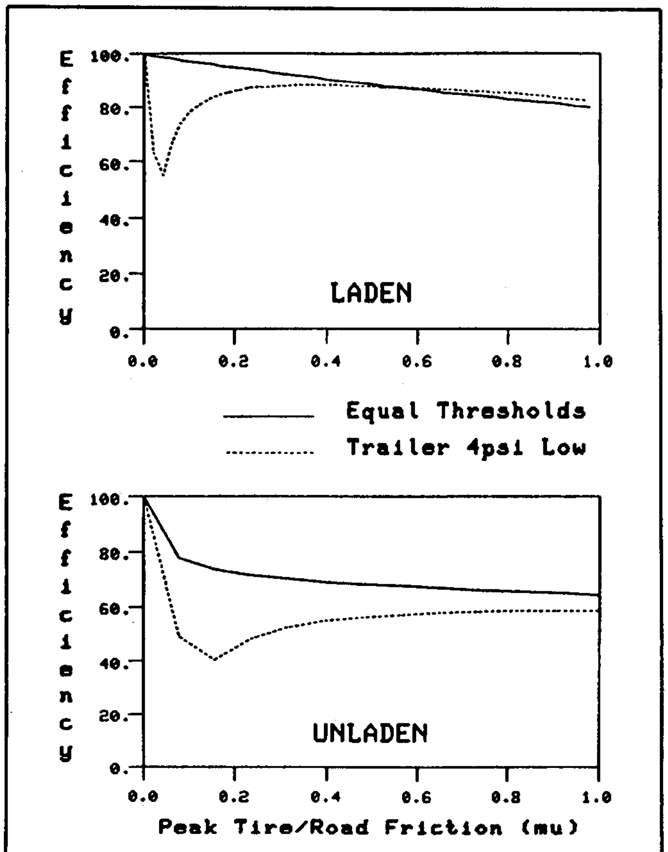

Figure 7. Braking efficiency for tractor-semitrailer combination with equal threshold pressures and 4 psi "low" trailer threshold (or 4 psi "high" tracter threshold)

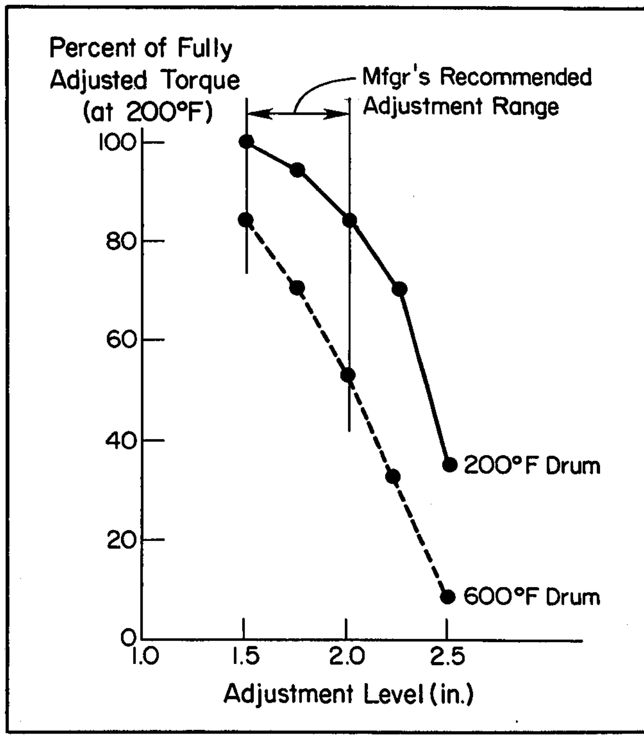

Figure 8. S-cam drum brake performance as a function of adjustment level and drum temperature

braking system performance. This work is described in References 11 and 12. It has been determined that the torque output of air braked heavy trucks is very sensitive to brake adjustment level. This is not the case for hydraulic brakes used on heavy trucks and most hydraulic brakes on cars and trucks are of the automatic adjusting type. The majority of truck air brake systems must be manually adjusted.

Figure 8 shows the effect of brake adjustment on the output of a typical heavy duty air brake at two different temperature levels, 200°F and 600°F (temperature in the brake drum).

The lower temperature represents a relatively "cool" brake that has not been exposed to a great deal of repeated or continuous braking. The higher temperature represents a relatively "hot" brake, and is typical for a mountain descent although it is by no means the maximum temperature that a brake might experience in service. Figure 8 is for an S-cam drum type brake, used on the majority (over 90 percent) of heavy duty air braked vehicles.

Adjustment level in Figure 8 represents the stroke of the air brake actuator (chamber) when the pressure in the actuator is 100 psi and the brake is at ambient temperature. Normally for the brake shown, the stroke of the actuator at 100 psi with the brake fully adjusted is approximately 1.5 inches; this stroke is required to take up the slack and deflection in the system. As the brake shoe wears, the stroke increases due to the greater actuator travel necessary to move the brake shoes out against the brake drum. For this particular brake, the manufacturer recommends that the brake be readjusted when the stroke reaches 2.0 inches although the actuator actually has a full travel of approximately 2.5 inches.

It can be seen from Figure 8 that at 200°F brake temperature, brake torque continually drops as adjustment level degrades from the fully adjusted level. This is true even over the manufacturer's recommended adjustment range; at the recommended readjustment point (2.0 inches) the torque has dropped to 85 percent of its fully adjusted level. When the brake is hot (600°F), there is a drop to 85 percent even when the brake is fully adjusted. This drop is due to two factors: 1) brake lining fade at the elevated temperature and, 2) brake drum expansion which results in an actuator stroke increase. Brake torque is reduced to 50 percent compared to a fully adjusted cool brake, when adjustment reaches the manufacturer's recommended readjustment point. This is a significant drop even though brake adjustment is considered to be acceptable in terms of the manufacturer's recommendations. Under this condition, the brake can only develop on half of the torque it could if it was fully adjusted and cool. Beyond the manufacturer's recommended adjustment range brake torque drop is even more dramatic, particularly if the brake is hot.

Reduced brake torque due to brakes being out-of-adjustment affects the brake force balance and overall braking capacity of the vehicle. As a result, not only is limit performance stopping ability affected, but downhill operations also become more prone to brake fade and runaway.

Figure 9 shows the results of limit performance stopping distance tests conducted on a fully loaded 6x4 truck at two different adjustment levels: 1) fully adjusted, and 2) at the manufacturer's recommended readjustment point. Both cool brakes (200°F) and hot brakes (600°F) are shown. Beyond the manufacturer's recommended adjustment range the stopping distance of the vehicle would be even longer than that shown in Figure 9.

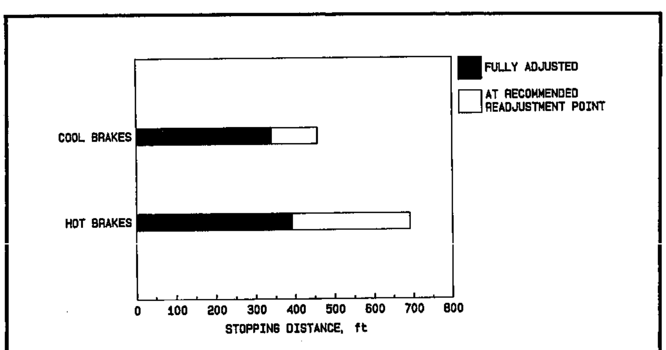

Figure 9. Stopping distance of fully loaded truck at two brake adjustment levels (60 mph—dry road)

Brake adjustment primarily affects the stopping capability of trucks when they are loaded; this is where maximum brake torque is needed to decelerate vehicle mass. With an empty vehicle, more than enough brake torque is usually available to lock the rear wheels despite the level of adjustment, unless adjustment is so poor that practically no torque is generated.

Retarders

Research performed under contract by the University of Michigan to evaluate the benefits of retarders for heavy vehicles(References 13,14,15) indicates that these devices can extend brake life and reduce the possibility of runaways on downgrades. VRTC, working in cooperation with the University of Michigan, conducted full scale vehicle tests to determine what effect these devices have on vehicle stability and stopping performance in limit braking maneuvers. Reference 16 describes this effort. The results of tests on two different combination vehicles indicate that in limit braking maneuvers, retarders can increase the stable stopping distances. Since most U.S. vehicles are "overbraked" on their drive axles, retarders (most commonly used retarders act through the drive axle) tend to upset brake force distribution even further. With retarders in operation, drivers must modulate the service brake control to an even greater degree to avoid wheel lockup and jackknife. Table 1 shows test results for braking tests on wet Jennite curves. Both of the vehicles use "engine brake" type retarders.

Use of retarders without applying the service brakes can also affect vehicle stability. Since retarders, in effect, utilize longitudinal friction at the tire/road interface, they reduce the lateral friction available for cornering. The maximum safe speed for entering a curve is reduced when the retarder is "on". In addition, loss of control at the limit speed can change from a stable "plow out" mode with the retarder "off" to an unstable jackknife mode if the retarder is "on".

In order to warn retarder users of the potential for stability problems, NHTSA has prepared an informa-

Table 1. Effect of retarders on stable stopping distance in slippery curves.

Vehicle	Loading	Curve Radius (ft)	Initial Speed (mph)	Best In-Lane Stopping Distance (ft) W/O Retarder	Best In-Lane Stopping Distance (ft) W/Retarder
6x4-S2	Empty Trailer	200	25	88	96
	Empty Trailer	500	40	311	323
	Bobtail	200	25	98	123
4x2-S1	Empty Trailer	200	25	88	98
	Empty Trailer	500	35	183	224
	Loaded Trailer	200	25	98	103

tional booklet(Reference 17) and given it widespread distribution in the trucking community. This booklet encourages the installation of retarders on vehicles since they do offer safety benefits and can extend brake system life significantly. The booklet, however, cautions against use of retarders (all types are usually controlled by an in-cab switch) in situations where the vehicle is empty and/or the road slippery.

Anti-Jackknife Devices

In 1975, NHTSA tested several different anti-jackknife devices on several different combination vehicles. Brake-in-a-curve, brake-during-a-lane-change and straight line braking tests were run on several different surfaces. Although these devices restrict articulation and prevent the tractor from hitting the trailer, they do not prevent wheel lock, and thus do not cure the basic instability problem. Figure 10 shows a tractor trailer both with and without an anti-jackknife device. Without the anti-jackknife device, the vehicle could jackknife if the tractor wheels lock. With such a device the vehicle will not jackknife but could spin as an entire unit (possibly across several lanes of traffic). In effect, the combination becomes a "long" straight truck.

Anti-jackknife devices will improve vehicle stability if the trailer wheels are kept rolling.* In this case, the trailer acts as a "rudder" for the combination vehicle. Most U.S. vehicles when empty, however, have their brakes balanced such that the first axle to lock on increasing brake input are those on the trailer. In a panic situation, full application of the brakes almost always locks the trailer brakes. It, therefore, cannot be assumed that the trailer wheels will be rolling in limit or emergency braking maneuvers.

Trailer Emergency Systems

A number of tests have been run to determine if the pneumatic systems on trailers could be simplified. This work, described in Reference 18, was performed in light of comments on a notice of proposed rulemaking to modify the requirements of FMVSS No. 121. Full scale vehicle tests, inertia dynamometer tests and laboratory tests of trailer pneumatic system "mockups" were performed. The results of these tests indicated that simplification would be possible but that care must be taken to avoid systems that permit spring (parking) brake drag without warning the driver. It was found that drivers could not feel spring brake drag even though it was occurring to the point of overheating and reducing the effectiveness of the brake system. It was also found in these tests that typical tractor plumbing is so restrictive in its delivery of supply air to the trailer that pneumatic failures on the trailer cannot be sensed by most tractors. Even if the trailer has a large leak, the tractor would still be able to maintain full reservoir pressure and would thus not give the driver any indication of the problem on the trailer. Unfortunately, in this case, the tractor low pressure warning light and buzzer which senses tractor reservoir pressure is no help in the event of trailer failures.

*Some promoters of these devices demonstrate them in tests with the trailer brakes turned off, an unrealistic operating condition

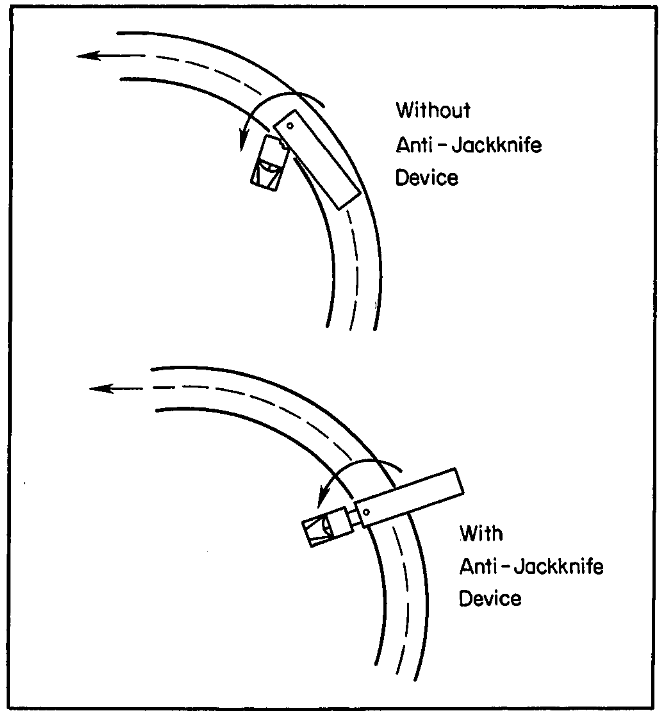

Figure 10. Loss of stability with and without anti-jackknife device

Performance of U.S. Versus European Vehicles

Comparative testing of a European tractor semi-trailer built to meet European brake standards (and not equipped with antilock) and a U.S. combination of equivalent size and weight has been performed. Many different types of braking maneuvers (straight line, curves, and lane changes), various surfaces and different vehicle loads have been included in these tests. Since this work has only recently been completed the report describing this effort in detail has not yet been published. The basic conclusion reached from these tests, however, is that the braking performance of the European combination is generally superior to that of the U.S. combination although the difference in performance was much smaller than expected. The European vehicle had the same brake on the front axle as on each rear axle and load sensing proportioning systems on the drive and trailer axles and this provided a more optimum brake force distribution over the range of operating conditions.

The only tests where performance was greatly different, however, were those with the bobtail tractors—the European tractor stopped much shorter and was easier to control during braking. When a simple bobtail proportioning system* was added to the U.S. vehicle, however, the performance of the two tractors was essentially the same.

Brake Linings

During the last year, dynamometer tests have been run to investigate the performance of heavy vehicle brake linings. This research which is expected to continue for at least another year is addressing two issues: 1) lining performance variability, 2) differences between asbestos and non-asbestos linings. Lining performance variability is important because it determines how closely brake force balance and braking efficiency can be controlled. This impacts tractor and trailer compatibility as well.

Understanding the performance characteristics of non-asbestos linings is important because they may be the only type of linings available in the future. Many vehicle manufacturers are now using them and EPA has proposed the complete elimination of asbestos in brake linings.

Data from this research effort will be made available to the SAE subcommittee that is currently developing new test procedures and rating schemes for brake linings. The SAE subcommittee is attempting to replace current SAE Recommended Practices for brake linings that are known to be inadequate. The data will also be provided to the SAE subcommittee that is developing Recommended Practices for tractor and trailer brake system compatibility.

Summary and Conclusions

NHTSA has been conducting research on heavy vehicle braking since 1969. Over the years many vehicles have been tested and many issues have been addressed. As a result of this effort, the braking performance characteristics of heavy vehicles are well understood and performance deficiencies have been identified. There is a large gap between the performance of passenger cars and heavy trucks and although this gap may never be completely bridged, significant improvements are possible.

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*This system is available as an option on some U.S. tractors. It reduces pressure to the drive axle brakes when a trailer is not connected to the tractor.

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Technical Session Seven

Pedestrian Protection

Chairman: Dr. Franco Rossi, Italy

NHTSA's Advanced Pedestrian Protection Program

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Abstract

The National Highway Traffic Safety Administration is conducting a research program addressing pedestrian upper body injuries inflicted by vehicle front ends. The objective of the program is to determine and demonstrate the feasibility of significantly reducing injury severity through structural modifications. Analysis of U.S. pedestrian accident data indicated that head and thorax impacts against vehicle faces, hoods, and fenders are of major importance, and that children are over-represented as accident victims. Accordingly, the research program consists of two main activities, which are briefly outlined in this paper. The first addresses adult and child head injury resulting from contact with the above-mentioned vehicle areas; the second, child thorax injury caused by the leading edges of hoods and fenders. Finally, an approximate estimate was made of the target population of head injuries in the U.S.—numbers of injuries now being sustained by those pedestrian accident victims who might benefit from vehicle modifications. Estimates are based on accident data projections and test results on production vehicles.

Background

Approximately 106,000 pedestrians are struck and injured by motor vehicles each year in the United States. Included are 7200 who are killed, accounting for 16 percent of the nation's motor vehicle traffic fatalities.

The National Highway Traffic Safety Administration (NHTSA) is addressing the problem through both crash prevention and crashworthiness research activities. The crashworthiness effort consists of two programs. One program has the objective of reducing bumper-induced leg injuries. The Advanced Pedestrian Protection Program, described in this paper, addresses upper body injury caused by contacts with other regions of the vehicle.

Accident Data Analysis

Accident data from the Pedestrian Injury Causation Study (PICS) were analyzed to identify pedestrian body areas that sustain the most frequent and severe injuries and to identify the sources of those injuries. PICS and National Accident Severity Study (NASS) data were used to determine other characteristics of the accident environment which should be known to define an effective research program.

We used the concept of "harm" developed by Malliaris et al.(1)* to provide a means of further quantifying the accident data. The total harm is the sum of all injuries suffered by crash victims, with each injury weighted according to severity. Injury severity is based on the AIS scale(2). The relative economic losses attributed to injuries at each severity level were estimated in Reference 3. The components of the societal losses are medical costs, productivity losses, and "other" expenses which include insurance and legal costs. Weighting factors for each injury level were suggested in References 1 and 4, based on the relative cost of injuries as reported in Reference 3. Table 1 shows the harm weighting factors suggested in Reference 4. These weighting factors give a basis for combining the harm caused by injuries at various severities.

The accident data indicate that most pedestrians are struck at relatively low impact speeds. This is understandable, since about 80 percent of all accidents (and more than half of the fatal accidents) occur on urban and local roads, and pre-impact braking occurs 70 percent of the time. Figure 1 shows the impact speed distribution for all pedestrian accidents. Note that most collisions occur at 5 to 10 mph, and that 90 percent are at impact speeds of 30 mph or less. Fifty-two percent of all pedestrian harm is attributable to collisions in the 0-30 mph impact speed range. As a result, we have directed our attention to accidents in this speed range. An additional reason for this focus of attention is that effectiveness of vehicle countermeasures intended to reduce injury severity will be greatly diminished when impact speeds exceed 30

*Numbers in parentheses indicate references.

Table 1. HARM—injury weighted by the economic consequence associated with that injury.

AIS Value	Harm Weight Factor
6	264.9
5	232.5
4	56.7
3	9.2
2	3.0
1	0.7

mph, considering our limited ability to address the consequences of accidents of such high severity.

Another important observation from the accident data was the very high frequency of young people, especially children, who are injured in pedestrian accidents. This is shown in Figure 2, which is a plot of the frequency of pedestrians injured each year, and the harm resulting from their injuries, as a function of their age, for impact speeds up to and including 30 mph. Young pedestrians, age 0-15 years, sustain 40 percent of all pedestrian injuries, and account for 30 percent of the total pedestrian harm, in this speed range.

The distribution of pedestrian harm by body region for collisions in the 0-30 mph impact speed range is presented in Figure 3. Also displayed is the distribution of injury severity associated with the harm for each body region.

Injuries to the abdomen were reclassified into two catagories, based on anatomical location: "hard" and "soft" abdominal injuries. Hard abdominal injuries consist of injuries to organs or other body regions protected by the lower thoracic cage. They account for 73% of all abdominal injuries. Soft abdominal injuries are those which result from contact in the soft tissue area, unprotected by the thoracic cage. Because hard abdominal injuries can be covered by an appropriate thorax injury criterion, we have included them in the thorax injury category.

Figure 3 shows that three-quarters of the pedestrian harm at impact speeds up to and including 30 mph results from head and thorax injury. For these two body regions, critical injuries (AIS 5) are the leading source of harm, producing 45%. About 13% of the harm occurs from leg injury, which, as previously mentioned, is being studied in another NHTSA program.

The distribution of harm by injury source, and injury severity for each source, appear in Figure 4 (as before, for impact speeds not exceeding 30 mph). Ground contacts account for 16% of the harm. In contrast to the other sources, a significant amount of ground-induced harm is contributed by minor (AIS 1) injuries.

"Other vehicle parts", which consist of vehicle undercarriage, tires and wheels, side surfaces, and

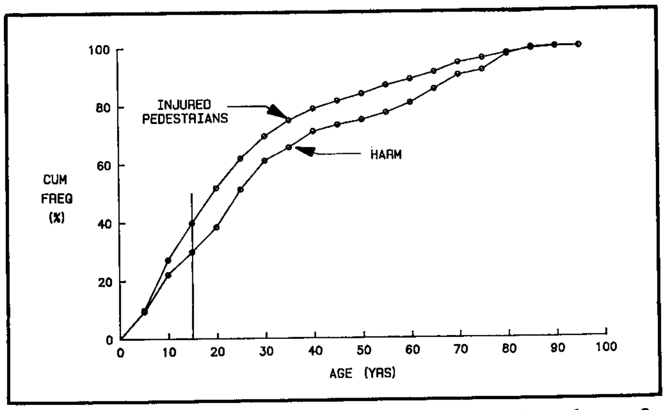

Figure 2. Cumulative freq. of injured pedestrians & harm for impact speeds ≤ 30 mph

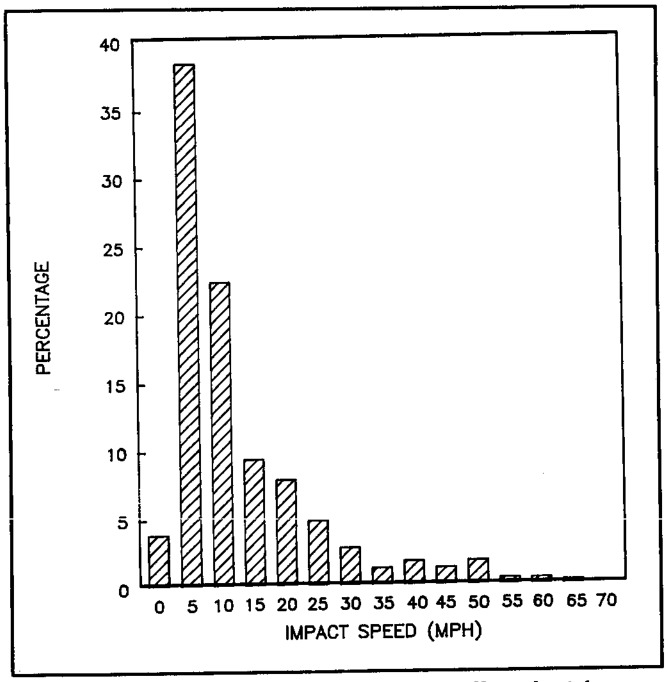

Figure 1. Impact speed distribution—all pedestrian accidents

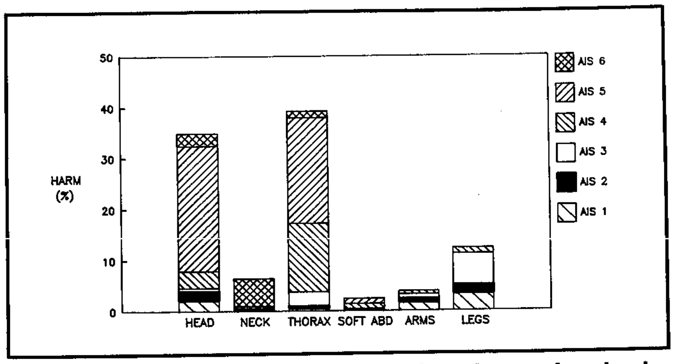

Figure 3. Distribution of pedestrian harm by body region and injury severity for impacts speeds ≤ 30 mph

rear surfaces, are responsible for nearly 12% of the harm. Over 80% of this harm is attributed to undercarriage and tires, implying that about 10% of all harm up to 30 mph results from pedestrians literally being "run over."

Non-contact injuries (10% of the harm) occur primarily to the neck and thorax. These neck injuries result from rapid excessive rotation of the head relative to the torso following contact to some body region other than the neck. Non-contact thorax injuries are undoubtedly induced by initial violent vehicle face/pelvis impacts. The bumper causes 7% of the harm, most of which results from AIS 3 leg injuries. Windshield areas, including glass, trim, A-pillar, and windshield wiper or mount contacts, cause 5%, mostly to the head.

This leaves about 50% remaining harm, which is caused by vehicle faces (grilles, headlight areas, and leading edges of hoods and fenders) and the top surfaces of hoods and fenders—vehicle areas in which practical injury-reducing structural modifications appear to be technologically feasible. Two-thirds of the face-hood-fender harm—i.e., one-third of the harm displayed in Figure 4—are caused by critical and maximum (fatal) injuries (AIS 5-6).

In Table 3, the data from the previous two figures are combined to show the most harmful injury source/body region contacts occuring at impact speeds of 30 mph and less. All contacts resulting in at least 2% of the harm are listed, thereby accounting for nearly 90% of the harm. Impacts of the head and thorax (including hard abdomen) against vehicle faces, hoods, and fenders are emphasized; they produce over 40% of the harm in this speed range.

Research Activities

Consistent with the results of the accident data analysis and vehicle modification feasibility, the research program is addressing pedestrian head and thorax injuries inflicted by vehicle front ends. Emphasis is being placed on children, due to their importance as accident victims. Both child and adult head impacts on vehicle faces and the tops of hoods and fenders (including cowls) are being studied. Child thorax injury resulting from impacts on vehicle faces and forward surfaces of hoods and fenders is also being researched.

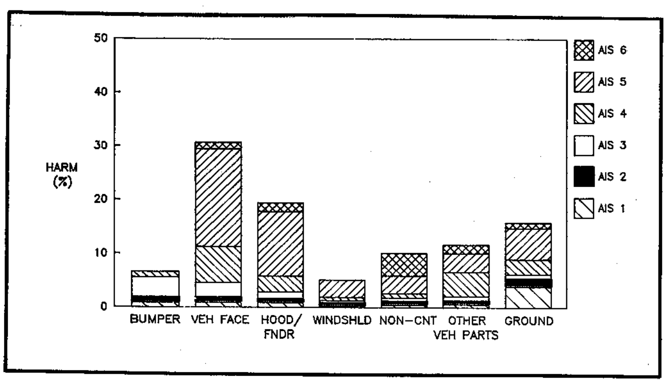

Figure 4. Distribution of pedestrian harm by injury source and injury severity for impact speeds ≤ 30 mph

Table 3. Most harmful injury source/body region contacts at impact speeds ≤ 30 mph.

Injury Source/Body Region	Harm (%)
Vehicle Face/Thorax	17.3
Hood-Fenders/Head	10.6
Ground/Head	10.4
Other Vehicle Parts/Thorax	8.7
Vehicle Face/Head	8.1
Hood-Fenders/Thorax	7.2
Bumper/Legs	6.5
Non-contact/Neck	5.1
Windshield/Head	4.5
Non-contact/Thorax	3.7
Vehicle Face/Legs	2.9
Ground/Thorax	2.2
	87.2%

Head Injury Reduction (Adult and Child)

The head injury research has been divided into the following areas:

- Test procedure development.
- Production vehicle testing.
- Development and evaluation of injury reduction counter-measures.

We have completed the development of test procedures and are nearing completion of the production vehicle testing. These activities, described in detail in Reference 5, will be highlighted here.

Test Procedure Development. A rigid variable mass impact device was developed for simulating adult and child head impacts on various vehicle surfaces. In order to derive an appropriate injury criterion to use with the device, we reconstructed specific accidents in the laboratory. We learned that simulating both the head impact velocity and the effective head mass was necessary to accurately reproduce the dynamic response; thus, the variable mass feature was essential to achieving successful accident reconstructions.

Both head injury criterion (HIC) and the newly revised translational mean strain criterion (TMSC)(6), measured from headform acceleration responses, correlated well with probability of fatality, determined from the three most severe head injuries suffered by the accident victims (see Table 2, taken from Reference 7). The relationship between probability of fatality and TMSC is shown in Figure 5. The coefficient of determination (R^2) for the exponential regression line is 0.832. Also shown in Figure 5 are three

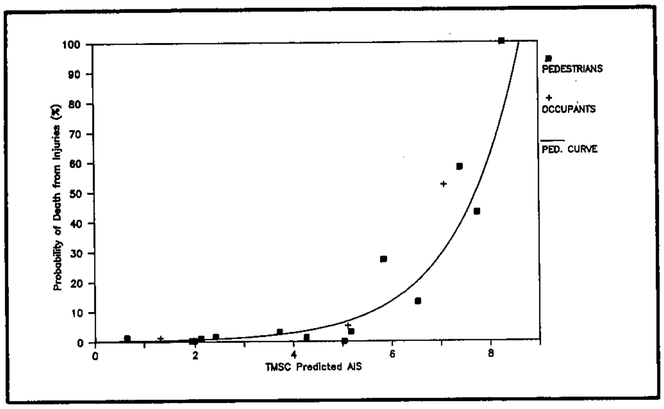

Figure 5. TMSC vs. probability of death—pedestrian & occupant head reconstructions

data points representing accident reconstructions of vehicle occupants, in which a different head impact device was used(8). These points are consistent with the pedestrian data, adding credence to the validity of the relationship.

The criteria agreed with each other, and provided consistent measures of injury severity, for HIC values up to approximately 2000. TMSC is more applicable to impact severities beyond this range, and is capable of providing information on the nature of the head injury. Consequently, we evaluated performance primarily on the basis of the TMSC.

Production Vehicle Testing. Having developed a head impact device, and having derived an injury criterion through accident reconstruction testing, we were ready

Table 2. 3-AIS rankings and probabilities of fatality.

AIS Index	AIS Ranking	Probability of Fatality (%)	AIS Index	AIS Ranking	Probability of Fatality (%)
1	1 0 0	0.1502	29	4 3 3	26.7080
2	1 1 0	0.3481	30	4 4 0	30.0853
3	1 1 1	0.8068	31	4 4 1	33.8896
4	2 0 0	0.9379	32	4 4 2	38.1750
5	2 1 0	1.2140	33	4 4 3	43.0022
6	2 1 1	1.5713	34	4 4 4	48.4399
7	2 2 0	2.0339	35	5 0 0	24.5181
8	2 2 1	2.6327	36	5 1 0	25.8821
9	2 2 2	3.4077	37	5 1 1	27.3220
10	3 0 0	1.8198	38	5 2 0	28.8420
11	3 1 0	2.0789	39	5 2 1	30.4465
12	3 1 1	2.3750	40	5 2 2	32.1403
13	3 2 0	2.7133	41	5 3 0	33.9283
14	3 2 1	3.0997	42	5 3 1	35.8158
15	3 2 2	3.5412	43	5 3 2	37.8083
16	3 3 0	4.0456	44	5 3 3	39.9117
17	3 3 1	4.6218	45	5 4 0	42.1321
18	3 3 2	5.2800	46	5 4 1	44.4759
19	3 3 3	6.0320	47	5 4 2	46.9502
20	4 0 0	9.1459	48	5 4 3	49.5622
21	4 1 0	10.3025	49	5 4 4	52.3194
22	4 1 1	11.6052	50	5 5 0	55.2300
23	4 2 0	13.0727	51	5 5 1	58.0236
24	4 2 1	14.7258	52	5 5 2	61.5461
25	4 2 2	16.5879	53	5 5 3	64.9700
26	4 3 0	18.6855	54	5 5 4	68.5844
27	4 3 1	21.0483	55	5 5 5	72.3999
28	4 3 2	23.7099	56	6 - -	100.0000

Probabilities of Fatality for Single Injuries

AIS Codes	Severity Code Descriptor	Probability of Fatality (%)
1 0 0	Minor	0.15
2 0 0	Moderate	0.94
3 0 0	Serious	1.82
4 0 0	Severe	9.15
5 0 0	Critical	24.5
6 0 0	Virtually Unsurvivable	100.0

to measure head injury potential from production automobiles. Thirteen 1985 and 1986 model year cars, representing a cross section of the current U.S. passenger car population, were selected for testing. Typical head impact sites for adults and children were chosen on vehicle faces, tops of hoods and fenders, and cowls. The head impact velocity was selected to represent a 30 mph vehicle/pedestrian collision (the upper bound of the 0-30 mph velocity range of interest).

To date, hood and fender testing is completed, and impacts on faces and cowls are approximately 50% completed. The principal findings from the hood and fender tests (some results of which are shown in Figures 6-9) are as follows:

1. A very large performance variation exists among today's production cars. Probability of fatality ranged from 5% to 100%, with the mean roughly at 50%.
2. Injury responses were typically the highest (50% to 100% probability of fatality) in the hood/fender interface regions.
3. Certain dynamic response characteristics associated with the "less injurious" impacts were observed. In general, impacts where the vehicle structure allowed the headform to travel through a stopping distance of at least 2 to $2\frac{1}{2}$ inches, where most of the energy (98 to 100%) was absorbed, and where structural stiffnesses were no more than 2000 pounds per inch, were predicted to be the least injurious.
4. The one plastic composite hood structure that we tested was far inferior to sheet metal (steel and aluminum), primarily because of much less energy absorption capability.
5. We identified important features of the hood reinforcing beam structure; a shallow cross-section in the typical hat section and a relatively sparse reinforcement pattern with beams arranged primarily longitudinally appeared to be best.
6. "Full cover" hoods (i.e., hoods which "wrapped over" the fenders, covering the entire width of the car), when tested in the hood/fender interface region, offered little improvement over more conventional designs. However, they allowed greater deflection and the opportunity for significant improvement with minimal design change. Very preliminary work in this important region indicates that responses similar to those in the best hood open areas (approximately 10% probability of fatality) are probably feasible.

Child Thorax Injury Reduction

The approach for addressing child thorax injuries is similar to that for the head; test procedures are being developed, production vehicles will be tested, and

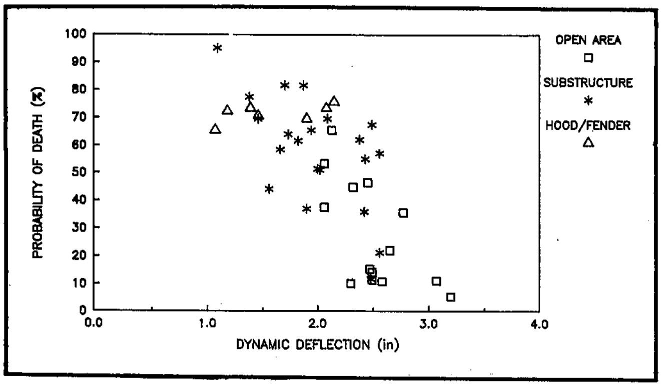

Figure 6. Production car test results—adult tests

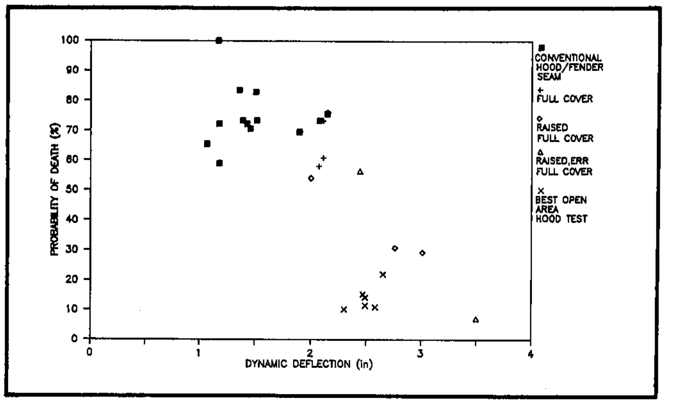

Figure 8. Hood/fender test results, compared with best hood open area results—adult tests

injury reducing concepts will be developed and evaluated. Currently, test procedure development is nearly completed. This work is outlined here; for greater detail, see Reference 9.

Test Procedure Development. The development of a suitable thorax impact device was much more difficult than it was for the head. The human thorax is softer, having a compliance which is comparable to that of the vehicle surfaces it is likely to contact; consequently, the compliance of the thorax device had to be similar to that of the human thorax. Also, in order to successfully derive an injury criterion by experimentally reconstructing accidents involving children of all ages and sizes, we had to build a family of thorax devices, representing children of different ages. The desired force-deflection characteristics for three, six, nine, and twelve year old children were derived by scaling adult cadaver response data(10). Four thorax impact devices were built, each one having the proper overall weight, rib to spine weight ratio, and compliance to simulate the child of the appropriate age. Accident reconstruction testing is now being done to derive an injury criterion to be used with these devices.

Definition of Target Population

Using some of the preliminary results of the head impact testing done to date, along with PICS accident data, we defined the "target population" of injuries—numbers of head injuries sustained by those pedestrian accident victims who might benefit from vehicle modifications.

Table 4 contains national estimates of numbers of pedestrian head and thorax injuries (not to be confused with numbers of injured pedestrians) known to be caused by those injury source/body area contacts that are addressed in our research, for vehicle/pedestrian impact speeds of 30 mph and less. Note that injury severity is expressed by AIS level and by the probability of fatality associated with that AIS level (bottom part of Table 2). Injuries with unknown source account for approximately 13% of all pedestrian injuries. Although some undoubtedly result from contacts in the categories listed in this table, no adjustment was made to include them.

Production vehicle head impact test results (Figures 6-9) indicate that it is possible to limit probability of fatality to 10% when 2 to $2\frac{1}{2}$ inches of stroking distance are available, and that there are some current production car hood locations where this has been achieved. It is assumed for purposes of these estimates that the hoods of all vehicles can be made to perform

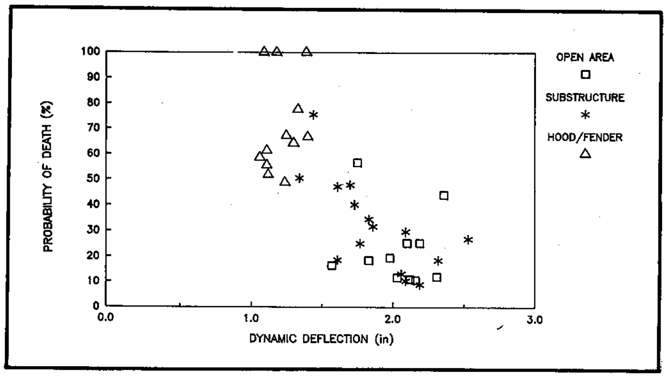

Figure 7. Production car test results—child tests

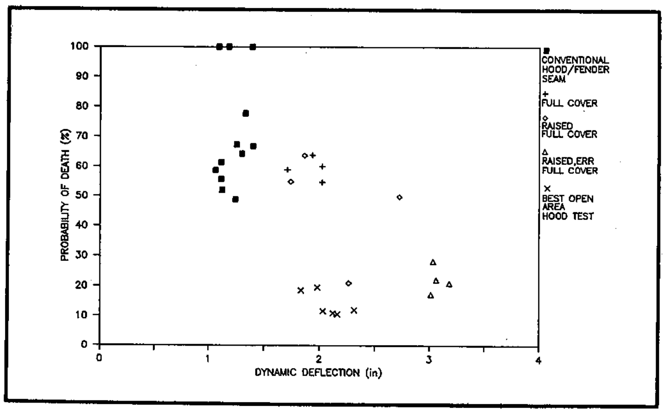

Figure 9. Hood/fender test results, compared with best hood open area results—child tests

Table 4. Numbers of injuries from head and thorax contacts with vehicle faces, hoods, and fenders, for impact speeds ≤ 30 mph.

AIS	1	2	3	4	5	6
Probability of Fatality (%)	0.15	0.94	1.82	9.15	24.5	100
Vehicle Face/Head	2376	1327	615	69	886	19
Hood-Fenders/Head	11834	2515	519	510	910	131
Total Head Injuries	14210	3842	1134	579	1796	150
Vehicle Face/Thorax	3164	635	1942	2787	1262	40
Hood-Fenders/Thorax	3624	700	2310	891	537	0
Total Thorax Injuries	6788	1335	4252	3678	1799	40

Table 5. Target population estimates—head contacts with vehicle faces, hoods, and fenders—impact speeds ≤ 30 mph.

AIS	1	2	3	4	5	6
Probability of Fatality (%)	0.15	0.94	1.82	9.15	24.5	100
Total Head Injuries (From Table 4)	14210	3842	1134	579	1796	150
Target Population of Head Injuries (0.60 x previous row)	8526	2305	680	347	1078	90

as well as the best ones, in those areas where $2\frac{1}{2}$ inches of stroke are available.

We measured the available stroking distance under the hoods of current cars. This was done on 10 vehicles, by placing clay cones at specific prominant locations under the hood (air cleaner, front suspension units, engine/accessory high points, etc.), and closing the hood, thereby compressing the clay. The average results are shown in Figure 10. We see that over 60% of the hood area of an average of today's cars has at least $2\frac{1}{2}$ inches of stroke available. Although specific measurements were not made in vehicle face, fender, and cowl regions, our observations indicate that extending the 60% - $2\frac{1}{2}$ inch estimate to those regions is reasonable. (Many current vehicles have full cover hoods and recessed cowls, helping to justify this estimate.)

From Table 4 we constructed Table 5 by multiplying "total head injuries" from Table 4 by 0.60. This produced the "target population" of head injuries—injuries that currently are being inflicted by regions of the vehicle surface estimated as having at least $2\frac{1}{2}$ inches of available stroke. The assumptions inherent in defining the target population in this manner are that 1) there is nothing that can be done to reduce injury in areas where stroke distance is less than $2\frac{1}{2}$ inches, and 2) it is impractical to provide a $2\frac{1}{2}$ inch stroke (or greater) over more than 60% of vehicle face, hood-fender, and cowl areas. The first of these assumptions, we feel, is conservative.

From our research to date, we feel that significant reductions may be possible in the numbers of serious to fatal head injuries now occurring as the result of pedestrian accidents in the U.S., through implementation of vehicle structural designs which depart very little from some of those currently in production. The number of thorax injuries is also substantial (see Table 4), suggesting that significant benefit may be possible in that body region as well.

Conclusions

1. Most pedestrians are struck at relatively low vehicle impact speeds. Ninety percent of all pedestrian accidents, and over 50% of all pedestrian harm, occur at or below 30 mph. For this reason, and because of practical constraints, the focus of attention in NHTSA's Advanced Pedestrian Protection Program is in this impact speed range.
2. Children are over-represented as pedestrian accident victims. Forty percent of pedestrian injuries, accounting for 30% of the harm, at impact speeds of 30 mph and less, are sustained by pedestrians from age 0 through 15 years.
3. Based on PICS data, the head and thorax (including "hard" abdomen) are the most frequently and severely injured pedestrian body areas. Vehicle faces (grilles, headlight areas, leading edges of hoods and fenders) and the tops of hoods and fenders cause most of these injuries. Over 40% of pedestrian harm occurring at impact speeds of 30 mph and less results from these contacts.
4. The PICS data, together with preliminary results from head impact tests on production vehicle hoods, indicate the presence of a substantial target population of head injuries—numbers of injuries occurring which may be reduced from practical vehicle modifications.
5. Plastic composite hood and fender designs are potentially more hostile to pedestrians than conventional sheet metal. High head injury severity

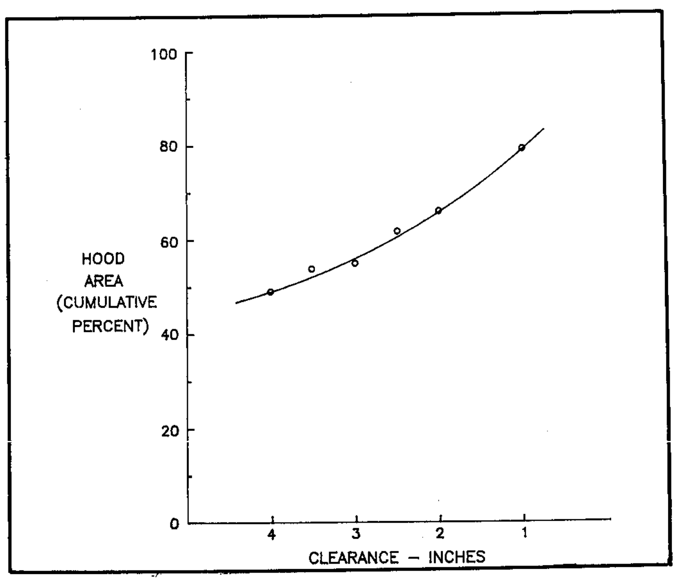

Figure 10. Available distance under the hood—average of 10 production cars

was predicted from tests on one such production car, due primarily to its very low impact energy absorption capability.

The discussion and conclusions in this paper represent the opinions of the authors and not necessarily those of the NHTSA.

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Risk Factor of the Road Accident Injuries' Seriousness for Car and Motorcycle Occupants and Pedestrians

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Abstract

The author has already studied original methodologies especially suited for evaluating the kind and seriousness of injuries in a road accident, including a practical method for data acquisition. One such methodology consists of calculating the relationship between the number of injuries in a single person run over by a car and the number of people we know who have suffered injuries (Coefficiente di Ripetibilità Lesiva: CRL $= \frac{nL}{nT}$). Such a relationship can be considered as "Seriousness Injuries Factor".

This meaning, which is generally accepted, is based on the evaluation by the traumatologist of any single minimum injury in an injured body related to the number of people injured. In a road accident, in which a pedestrian is involved, this methodology becomes more interesting and substantial because it outlines not only the number of injuries but also provides a practical indication about the parts of the human body which are more frequently involved. In the case of a car accident, the analysis of the injuries on the bodies of the occupants of the car points out that some well-identified parts of the body are more frequently involved than others because of the typical structure of the car. In the analysis of pedestrians, the multiple injuries are caused by a lot of factors which are related essentially with the collision trajectory, the type of vehicle and above all the speed; therefore the number of injuries is strictly dependent on the seriousness of the accident. This is of the maximum importance for the evaluation of the seriousness of the accident aimed to prevention needs. As a practical application, this methodology can be used for prevention purposes in a specific and well identified area of road accidents.

Introduction

The analysis of a road accident presents, in its totality, various aspects which can be studied more or less deeply depending on the different methodologies used in the study itself.

The real problem is to synthesize in an easy way the matter in order to obtain immediate evidence from it.

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In every case we have first to establish the limits of the problem; so we have to decide where it starts and what is the target.

Nowadays techniques and technologies are available to be able to have results by using very complicated elaborations in very small time; this was impossible a few years ago.

The first step is to obtain statistical data useful to collect information about situations and problems in order to have appropriate comparison.

We must start to define what a road accident is in our meaning.

The only correct way to examine the problem is to consider the road accident as an accident occurring in squares or streets and involving vehicles, pedestrians and/or animals in such a large entity that it causes police intervention to register the event.

Material and Method

The purpose of this study is to establish in the simplest way a methodology to assess the seriousness of a road accident; to do this we must analyze first the elements we have to consider.

They are: the environment, the mechanical vehicle, the persons and the injuries.

We consider the last point later in a particular way because in this study we also take into consideration the kind and the number of injuries and the relationship existing between them, the number of wounded and the number of people in the vehicle.

This correlation represents an original innovation.

In other terms to develop a correct analysis about this problem we need to collect a certain number of significant data about the accident. Therefore we do need to properly organize the available data.

The innovative aspect we have mentioned relates to the availability of the additional series of data concerning the number and the kind of injuries.

In synthesis we can say that the study is based upon the following two concepts:

- the first data about the accident are taken by the people called to succor the wounded. After a short training they will be able to report the essential data concerning the accident. To this scope the accident is defined via the first three aspects (environment, mechanical vehicle, people);
- the second part of the analysis consists in pointing out all the aspects of injuries; this part presents more difficulty and must be done in subsequent time by the traumatologists that have followed the various cases.

The cross-correlation between the number of injuries and the number of wounded, the number of wounded and the number of people in the vehicle, the number of injuries and the number of people in the vehicle, is able to give us an idea of the seriousness of the accident itself and consequently of the actions to be undertaken.

Consideration About the Seriousness of the Accident

As we have just mentioned, the originality of this study is represented by the introduction of some new parameters which properly intercorrelated can give us a careful idea of accident seriousness.

We have also evidentiated that all data concerning the environment, the mechanical, vehicle, the people and the injuries are to be collected via appropriate schedules.

In the first schedule (Figure 1) we report the number of occupants in relation to the number of wounded.

For completeness in another schedule concerning accidents involving pedestrians we report the number of pedestrians and the number of wounded (Figure 2).

From another schedule which we can call "traumatological schedule" (Figure 3) we are able to take out the number and the type of injuries related to occupants and/or pedestrians.

From the correlation of the number of injuries and the number of wounded we obtain the "Coefficiente di Ripetibilità Lesiva" "Coefficient of Lesive Repetition". (Figure 4).

This coefficient gives a clear indication of the accident seriousness (seriousness injuries factor). We can also correlate the number of injuries per wounded and the number of wounded people with the number of occupants involved in the accident.

The elaboration of these data can give us some suggestion for some preventive measures.

To perform this study we needed appropriate statistical data collected within a significant area of our country during an appropriate period of time.

We have considered the accidents occurred in Rome and surroundings during one solar year limiting the analysis to accidents involving persons heavily injured or dead.

This was the only way to obtain significant results from the study specially to account for the injuries in various parts of the human body.

In a totality of 820 road accidents observed, 54.9% involved cars and motorcycles (26% only cars, 28.9% only motorcycles) while the remaining 45.1% involved pedestrians with a 39.4% in accidents among cars and pedestrians and 5.7% in accidents among motorcycles and pedestrians (Figure 5-6).

On a totality of 3,202 injuries; 78.4% belong to car and motorcycle drivers while 21.6% belong to pedestrians (Figure 7).

FORM FOR TRAUMATIC INJURIES IN ROAD ACCIDENTS

DATE DD MM YY   TIME HH MM   LOCALITY .... DISTRICT ....

REPORT N. ....

GENERAL INFORMATION

A VENUE
- city street...... 1
- suburban road.... 2
- highway.......... 3

B TRAFFIC
- light............ 1
- normal........... 2
- heavy............ 3

C WEATHER CONDITIONS
- serene............ 1
- bright sunlight... 2
- fog............... 3
- smoke/dust........ 4
- rain/hail......... 5
- snow/ice.......... 6
- wind.............. 7
- twilight.......... 8

D SUSPECTED CAUSE
- over speed limit...... 1
- precedence not given.. 2
- pass.................. 3
- tailgating............ 4
- light blindness....... 5
- signs not observed.... 6
- skid.................. 7
- distraction........... 8

E ROAD SURFACE
- normal........... 1
- broken........... 2
- work in progress. 3
- other anomalies.. 4

F NATURE OF THE ACCIDENT
- between moving vehicles.......................... 1
- a moving vehicle and an inert one or other obstacle 2
- a moving vehicle and other obstacle – no impact.... 3

VEHICLES INVOLVED

A TYPE OF VEHICLE

	A	B	C	D
bicycle............	11	12	13	14
motorcycle..........	21	22	23	24
automobile.........	31	32	33	34
bus/pullman........	41	42	43	44
truck/van..........	51	52	53	54
TIR/trailer truck..	71	72	73	74

B CYLINDERS: cc. or WEIGHT: met.ton. — A, B, C, D

C YEAR OF FIRST MATRICULATION — A, B, C, D

D PART INVOLVED

	A	B	C	D
front............	11	12	13	14
back.............	21	22	23	24
right side.......	31	32	33	34
left side........	41	42	43	44

E DAMAGE TO VEHICLE

	A	B	C	D
none.............	11	12	13	14
body.............	21	22	23	24
interior.........	31	32	33	34
engine parts.....	41	42	43	44
destruction......	51	52	53	54

F SUSPECTED FAILURES

	A	B	C	D
none.................	11	12	13	14
suspension...........	21	22	23	24
brakes...............	31	32	33	34
tires................	41	42	43	44
steering column......	51	52	53	54
head-/taillights......	71	72	73	74
general..............	81	82	83	84

G SECURITY DEVICES

	A	B	C	D
seatbelts.......	11	12	13	14
headrests.......	21	22	23	24
absence thereof.	31	32	33	34
helmet..........	41	42	43	44
other...........	51	52	53	54

H no. of occupants — A, B, C, D

DRIVER

SEX / DATE OF BIRTH

	M	F
A	1	5
B	2	7
C	3	8
D	4	9

OWNER / YEAR LICENSE ACQUIRED

	YES	NO
A	11	12
B	21	22
C	31	32
D	41	42

PASSENGERS

VEHICLE A — SEX — DATE OF BIRTH

	M	F
T1	1	5
T2	2	7
T3	3	8
T4	4	9

VEHICLE B — SEX — DATE OF BIRTH

	M	F
T1	1	5
T2	2	7
T3	3	8
T4	4	9

VEHICLE C — SEX — DATE OF BIRTH

	M	F
T1	1	5
T2	2	7
T3	3	8
T4	4	9

VEHICLE D — SEX — DATE OF BIRTH

	M	F
T1	1	5
T2	2	7
T3	3	8
T4	4	9

WARNING: The various vehicles involved are identified by the letters A, B, C, D; remember this when filling out form, attach another if there are more than 4.

WARNING: indicate only injured passengers

Figure 1

FORM FOR TRAUMATIC INJURIES IN ROAD ACCIDENTS (PEDESTRIANS)

LOCALITY .... DISTRICT ......

DATE DD MM YY TIME HH MM REPORT N. ....

1- GENERAL INFORMATION

A VENUE
- city street...... 1
- suburban road.... 2
- highway.......... 3

B TRAFFIC
- light........... 1
- normal.......... 2
- heavy........... 3

C WEATHER CONDITIONS
- serene........... 1
- bright sunlight... 2
- fog.............. 3
- smoke/dust....... 4
- rain/hail........ 5
- snow/ice......... 6
- wind............. 7
- twilight......... 8

D SUSPECTED CAUSE
- over speed limit...... 1
- precedence not given.. 2
- pass.................. 3
- tailgating............ 4
- light blindness....... 5
- signs not observed.... 6
- skid.................. 7
- distraction........... 8

E ROAD SURFACE
- normal........... 1
- broken........... 2
- work in progress. 3
- other anomalies.. 4

F NATURE OF THE ACCIDENT
- between moving vehicles.......................... 1
- a moving vehicle and an inert one or other obstacle. 2
- a moving vehicle and other obstacle - no impact..... 3

G DIRECTION
- one-way.......... 1
- two-way.......... 2

H DIRECTION OF CROSSING PEDESTRIAN
- from left to right.............................. 1
- from right to left.............................. 2

L TYPE OF ROAD
- one lane......... 1
- two or more...... 2

M POSITION OF PEDESTRIAN
- on crosswalk or at traffic light.................. 1
- at LESS than 50 mt. from crosswalk or traffic light. 2
- at MORE than 50 mt. from crosswalk or traffic light. 3

N LOCATION
- near to school... 1
- near to church... 2
- industrial area.. 3
- shopping area.... 4
- near to hospital. 5
- suburbs.......... 6

P STREET LIGHTING
- none................. 1
- insufficient......... 2
- good................. 3

Q LIGHTS ON
- none................ 1
- parking lights...... 2
- dimmers............. 3
- bright lights....... 4

2- COLLISION VEHICLE

A TYPE OF VEHICLE
- bicycle........... 1
- motorcycle........ 2
- automobile........ 3
- bus/pullman....... 4
- truck/van......... 5
- TIR/trailer truck.. 6

B CYLINDERS: cc. or WEIGHT: met.ton.

C YEAR OF FIRST MATRICULATION

D PART INVOLVED
- front........... 1
- back............ 2
- right side...... 3
- left side....... 4

E DAMAGE TO VEHICLE
- none............ 1
- body............ 2
- interior........ 3
- engine parts.... 4
- destruction..... 5

F SUSPECTED FAILURES
- none................ 1
- suspension.......... 2
- brakes.............. 3
- tires............... 4
- steering column..... 5
- head-/taillights.... 6
- general............. 7

G SECURITY DEVICES
- seatbelts............ 1
- headrests............ 2
- absence thereof...... 3
- helmet............... 4
- other................ 5

3- DRIVER AND PEDESTRIAN

	SEX M	SEX F	DATE OF BIRTH	YEAR LICENSE ACQUIRED	OWNER YES	OWNER NO
DRIVER	1	2			1	2
PEDESTRIAN	1	2				

Figure 2

FORM FOR TRAUMATIC INJURIES IN ROAD ACCIDENTS

REPORT N. .... DATE ....

LOCALITY OF THE ACCIDENT ....
DISTRICT ....

ROLE

pedestrian	1
driver	2
passenger	3

SEX: M 1, F 2

DATE OF BIRTH

PROGNOSIS (in days) or
DECEASED (mark with an X)

MOST SERIOUS INJURY
(indicate by means of this
injury's code number)
fill in only when there
are multiple injuries

HEAD	
bruise	1
wound	2
fracture	3
FACE	
bruise	4
wound	5
fracture	7
eye injury	8
tooth injury	9
NECK and CERVICAL COLUMN	
bruise	10
wound	11
fracture	12
dislocation	13
whiplash	14
medullar injuries	15
CHEST and SPINAL COLUMN	
bruise	17
wound	18
rib fracture	19
vertebral fracture	20
internal injuries	21
LUMBAR REGION	
bruise	22
wound	23
fracture	24
medullar injuries	25
ABDOMEN	
bruise	27
wound	28
internal trauma	29
internal injuries	30

UPPER LIMBS	left	right
SHOULDER		
bruise	311	312
wound	321	322
fracture	331	332
dislocation	341	342
ARM		
bruise	351	352
wound	371	372
fracture	381	382
ELBOW		
bruise	391	392
wound	401	402
fracture	411	412
dislocation	421	422
LOWER ARM		
bruise	431	432
wound	441	442
fracture	451	452
WRIST		
bruise	471	472
wound	481	482
fracture	491	492
HAND		
bruise	501	502
wound	511	512
fracture	521	522
dislocation	531	532
VASCULAR INJURIES	541	542
NERVE INJURIES	551	552
AMPUTATION	571	572

LOWER LIMBS and PELVIS	left	right
PELVIS and HIPS		
bruise	581	582
wound	591	592
pelvic fracture	701	702
acetabulum fracture	711	712
dislocation	721	722
THIGH		
bruise	731	732
wound	741	742
FEMORAL NECK		
fracture	751	752
FEMORAL DIAPHISIS		
fracture	771	772
KNEE		
bruise	781	782
wound	791	792
knee-cap fracture	801	802
tibial plate fracture	811	812
tibial condyles fract.	821	822
other fractures	831	832
dislocation	841	842
LEG		
bruise	851	852
wound	871	872
fracture	881	882
INSTEP		
bruise	891	892
wound	901	902
fracture	911	912
dislocation	921	922
FOOT		
bruise	931	932
wound	941	942
fracture	951	952
AMPUTATION	971	972

Figure 3

$$CRL = \frac{n.l.}{n.t.}$$

Figure 4. Seriousness injuries factor as relation between injuries and number of people who suffered injuries

Accidents between two cars are 48.1%, among cars and motorcycles 30.3%, among cars and pedestrians 8.9% and finally 2.7% among motorcycles and pedestrians (Figure 8).

If we want to codify the injuries seriousness depending on their association in the some wounded, we must analyse the correlation existing between the number of injuries and the number of wounded related to every kind of accident.

Indicating by A-A an accident occurred between two cars, by A-M an accident between car and

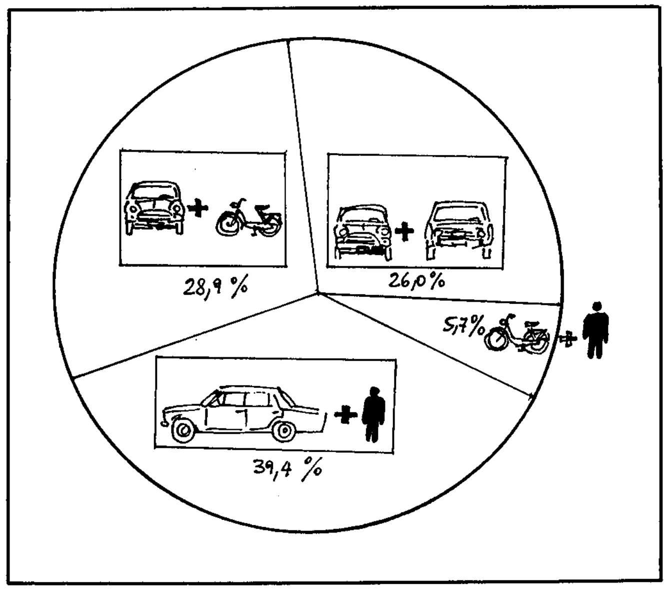

Figure 6. Kind of accidents (unassociated)

motorcycle, by A-P an accident between car and pedestrian we obtain, as you can see in Figure 9, the coefficients of lesive repetition (CRL).

You can immediately note how the major number of injuries per wounded relates to the motorcycle driver (CRL = 3.22).

Which means that every wounded motorcycle driver suffers about a double number of injuries suffered by a car driver. The average CRL value of all accidents in 2.57.

It means that in the Rome zone every wounded person due to road accident presents little less than three injuries (Figure 10).

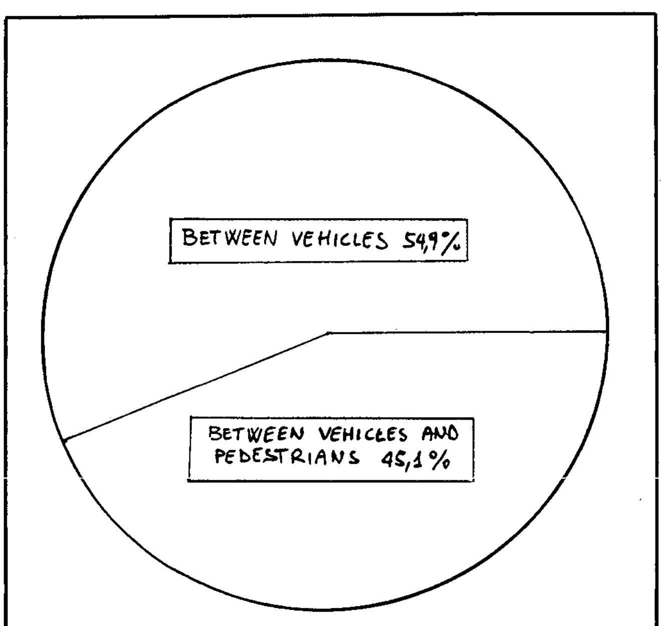

Figure 5. Kind of accidents (associated)

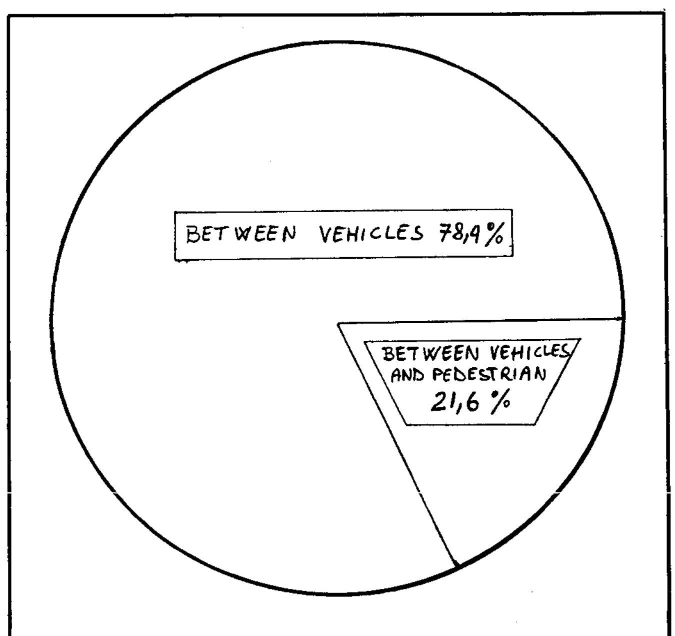

Figure 7. Injuries for kind of accident (associated)

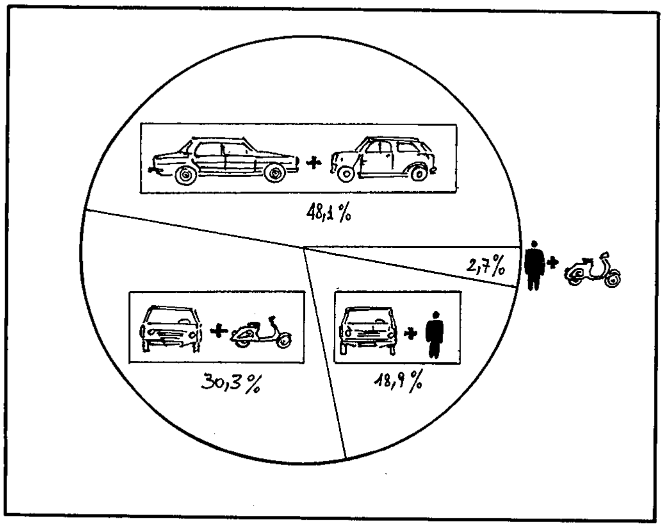

Figure 8. Injuries for kind of accident (unassociated)

$$CRL = \frac{3202}{1246} = 2{,}57$$

Figure 10. Total evolution of CRL in investigated cases

Prevention

With the introduction of these kind of parameters we are able to recognize the entity of the accident and so directly or indirectly to understand the accident's causes.

We can also provide these data to the officials working about road safety and help them to find relevant solutions to the problems to improve road safety in general.

We leave the various specialists to solve related different problems such as infrastructures, support vehicles, legislative and legal aspects.

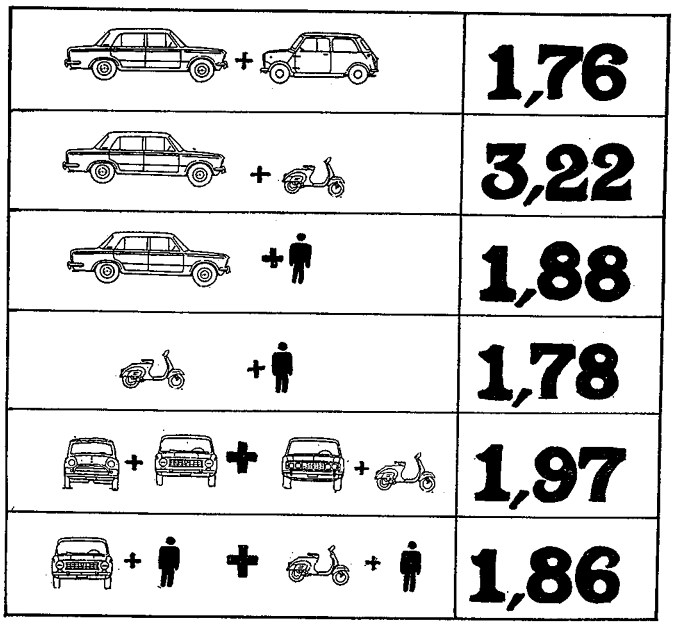

Figure 9. Results of CRL in relation to various kinds of accident

Conclusions

We have demonstrated by this study that it is possible to find very interesting parameters to study the entity of a road accident.

In particular it has been developed the concept to define the seriousness of a road accident by the ratio CRL.

We have also considered other characteristic parameters to evidentiate other aspects of accident seriousness specially when cars with many occupants or collective transportation vehicles are involved. Any how the study has confirmed the need for properly identified and collected data suitable to be elaborated to get the most interesting conclusions from them.

Finally the interpretation of the parameters by the various specialists like car manufacturers, people tasked to develop new regulations and laws to improve road safety etc., allows a practical use of this study.

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Development of Countermeasures to Reduce Pedestrian Head Injury

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Abstract

The National Highway Traffic Safety Administration is conducting research to develop methods of reducing pedestrian head injury due to automobile hood contact at speeds that would result from cars hitting pedestrians at less than or equal to 30 miles per hour (48kph). This paper describes the current status of work to develop vehicle designs which reduce pedestrian head injury severity. The development of techniques used to simulate head impacts on vehicle surfaces is briefly described. Tests were conducted on production hoods and fenders to: 1) determine the head injury potential of common hood and fender impact regions, and 2) understand how specific geometric and/or material characteristics influence injury severity and might be altered to reduce the severity of head impacts. Accident reconstruction research results identified an injury criterion which enables headform test responses to be related to injury severity with a high degree of confidence. The large variation in predicted head injury severity, obtained in production car testing, demonstrates that significant injury reduction could be achieved by making all cars perform as well as the best current production cars. The best head impact results, from 30 miles per hour (48kph) impact speeds, produced 10% probabilities of fatality with 2 to 2.5 inches (5.1-6.4 cm.) of dynamic deflection. The least injurious impacts occurred when nearly 100% of the impact energy was absorbed by the vehicle structure, acceleration pulse duration was 20 milliseconds or greater, and structural stiffness did not exceed 2000 pounds per inch (3500 N/cm). The potential for a significant raise in severe pedestrian head injury is indicated with widespread use of plastic composite or fiberglass materials in hoods and fenders.

Introduction

The National Highway Traffic Safety Administration (NHTSA) is addressing the pedestrian accident problem in the United States through a number of programs, one of which is the Advanced Pedestrian Protection Program, described in Reference 1. An analysis of accident data conducted as a part of this program indicated that head and thorax injuries account for most of the harm to pedestrians, and that vehicle faces (grilles, headlight areas, leading edges of hoods and fenders) and the top surfaces of hoods and fenders are the major sources of injury. Consequently, a major part of the Advanced Pedestrian

Protection Program is to develop methods of reducing adult and child pedestrian head injury due to contact with automobile hoods, fenders and faces. Thorax injury reduction is the subject of a concurrent study, which is reported in Reference 2.

The objective of this paper is to describe the current status of work to develop vehicle designs which reduce pedestrian head injury severity. Techniques for simulating pedestrian head impacts on vehicle surfaces were developed and tests were conducted on production hoods and fenders to 1) determine the head injury potential of common hood and fender impact regions, and 2) understand how specific geometric and/or material characteristics influence injury severity and might be altered to reduce the severity of head impacts. An underlying theme present during all the research was that any design modification intended to reduce pedestrian head injury must be practical and production feasible.

Head Impact Simulation

Impact Device

An impactor device, shown in Figure 1, was designed to simulate a pedestrian-vehicle head impact in a controlled and repeatable laboratory experiment. The impactor uses pneumatic pressure to accelerate a drive piston which accelerates the impacting ram and headform. The impacting ram is a free projectile confined to uniaxial motion. The headform, as seen in Figure 2, is a variable mass semi-spherical aluminum fixture covered with Hybrid III dummy skin.(3)

Head Injury Prediction

To derive a means of predicting head injury severity from impactor response measurements, the impactor was used in an attempt to reconstruct head dynamic responses experienced in 35 accident cases involving head injury. To achieve a successful accident reconstruction, it was necessary to duplicate the impact responses, including the vehicle damage, or dent, caused by impact of the pedestrian's head. It was found this could be done only if both the effective mass and impact velocity of the head were closely simulated. Head impact velocities were obtained from computer simulation, using, as input, the investigators' estimated vehicle velocities at impact. In general, impactor mass was varied, within a reasonable range of computed head impact velocities, to achieve the proper vehicle damage.

Reference 4 contains preliminary results of the head impact reconstruction experiments. Further analysis of the reconstruction test results and recent changes in the formulation of one of the injury criteria being evaluated(5) led to some revisions to the results of Reference 4. The most recent findings from the reconstruction work will be summarized here.

Fourteen of the adult accident cases were particularly well reconstructed. (i.e.,the vehicle damage, or dent, caused by each head impact was very accurately duplicated in the test.) These 14 cases were also uniformly distributed over a full range of injury severity. Therefore, results from these reconstructions were used to establish correlation between test responses and injury severity experienced in the accidents. Probability of death was determined from the three most severe head injuries for each of the 14 accidents victims. Table 1, taken from Reference 6, was used for this purpose. This was done to obtain a more accurate and continuous injury severity scale than is provided by the overall AIS level. Two parameters, Head Injury Criterion (HIC) and the Translational Mean Strain Criterion (TMSC)(5), calculated from the test responses, were found to correlate well with probability of death. Figure 3 shows a bi-linear curve fit between probability of death and HIC. The coefficient of determination (R^2) is 0.924. In Figure 4, probability of death is plotted against

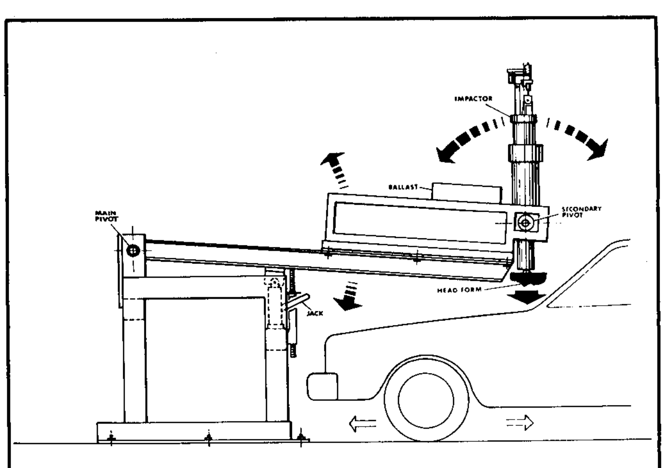

Figure 1. VRTC pedestrian head impact simulator

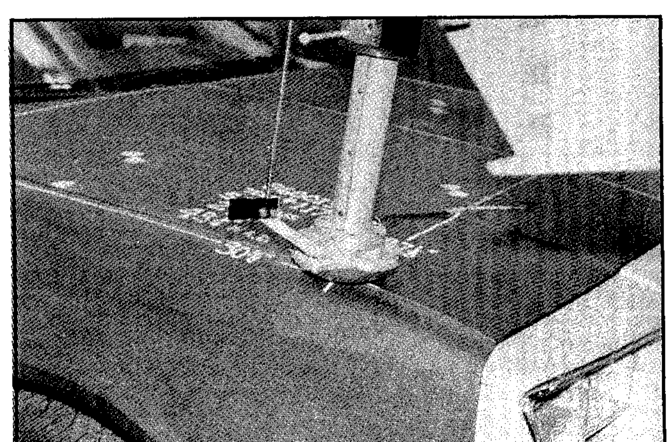

Figure 2. Variable mass headform of the head impact simulator

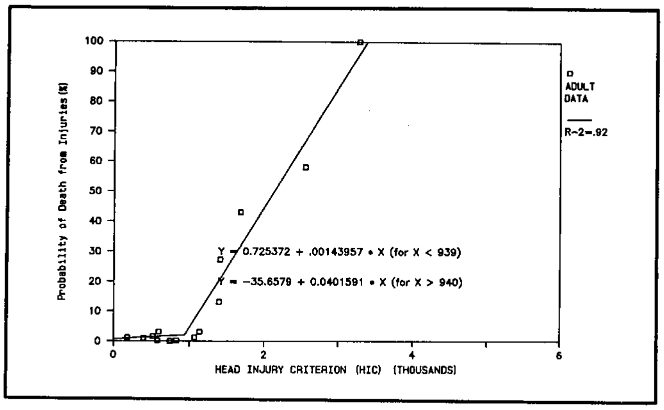

Figure 3. Variation of probability of death with HIC for the adult reconstructions

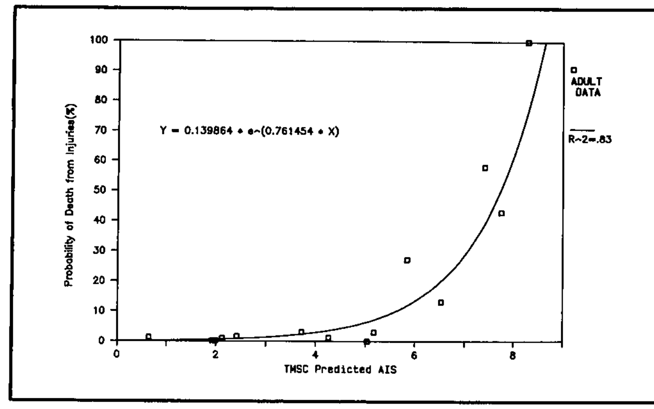

Figure 4. Variation of probability of death with TMSC for the adult reconstructions

TMSC. The relationship is described by an exponential regression line having a coefficient of determination of 0.832.

In Figures 5 and 6, 11 child head impact cases, representing the best of the child reconstructions, have been added to the 14 adult cases shown in Figures 3 and 4. The child cases, all at relatively low injury severity levels, are clustered around the lower ends of the curves, and appear to be consistant with the correlations derived from the adult cases.

Initially, in the production car testing (to be presented in the next section of this paper), HIC and TMSC values were calculated, and both relationships were used to predict injury severity. It was discovered, however, that many of the production car tests produced HIC values which greatly exceeded the normal range in which HIC can be used with confidence. TMSC is more applicable to high severity impacts, and has added capability of providing information on the nature, as well as severity, of head injuries. As a consequence, although both criteria continue to be used, more confidence is being placed in the TMSC, and results are presented in this paper on the basis of the TMSC.

Table 1. AIS rankings and probabilities of death.

INDEX	3-AIS RANKING	PROB. OF DEATH (%)	INDEX	3-AIS RANKING	PROB. OF DEATH (%)
1	100	0.1502	29	433	26.7080
2	110	0.3481	30	440	30.0853
3	111	0.8068	31	441	33.8896
4	200	0.9379	32	442	38.1750
5	210	1.2140	33	443	43.0022
6	211	1.5713	34	444	48.4399
7	220	2.0339	35	500	24.5181
8	221	2.6327	36	510	25.8821
9	222	3.4077	37	511	27.3220
10	300	1.8198	38	520	28.8420
11	310	2.0789	39	521	30.4465
12	311	2.3750	40	522	32.1403
13	320	2.7133	41	530	33.9283
14	321	3.0997	42	531	35.8158
15	322	3.5412	43	532	37.8083
16	330	4.0456	44	533	39.9117
17	331	4.6218	45	540	42.1321
18	332	5.2800	46	541	44.4759
19	333	6.0320	47	542	46.9502
20	400	9.1459	48	543	49.5622
21	410	10.3025	49	544	52.3194
22	411	11.6052	50	550	55.2300
23	420	13.0727	51	551	58.0236
24	421	14.7258	52	552	61.5461
25	422	16.5879	53	553	64.9700
26	430	18.6855	54	554	68.5844
27	431	21.0483	55	555	72.3999
28	432	23.7099	56	600	100.0000

PROBABILITIES OF DEATH FOR SINGLE INJURIES

AIS CODES	SEVERITY CODE DESCRIPTOR	PROB. OF DEATH (%)
100	MINOR	0.2
200	MODERATE	0.9
300	SERIOUS	1.8
400	SEVERE	9.1
500	CRITICAL	24.5
600	VIRTUALLY UNSURVIVABLE	100.00

Production Car Testing

Production car testing was the first step taken in the effort to develop countermeasures that reduce pedestrian head injury severity. Specifically, production car tests were done to determine head injury potential of common hood and fender impact regions and to understand how particular geometric and/or material characteristics influence injury severity. Knowledge in these areas would help determine if vehicle modifications might be made, in a production feasible and practical manner, which reduce head injury severity.

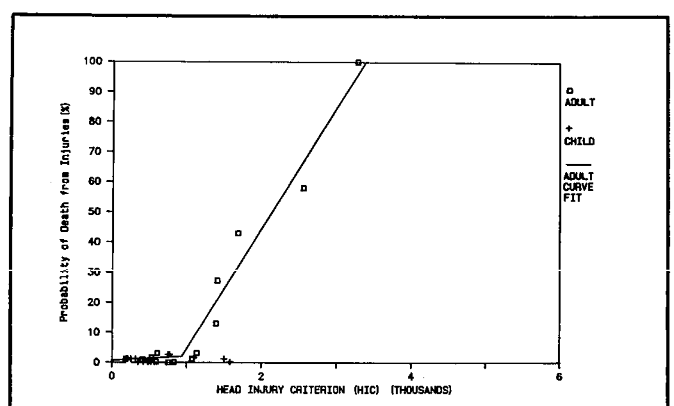

Figure 5. Variation of probability of death with HIC for the adult and child reconstructions

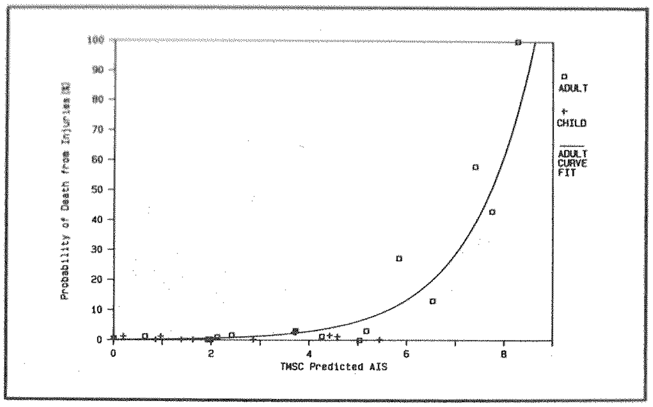

Figure 6. Variation of probability of death with TMSC for the adult and child reconstructions

Test Conditions

The primary criterion for the selection of vehicles for production car head impact testing was that they represent a reasonable cross-section of the current U.S. car fleet. Vehicle size, popularity and styling were important factors. Secondly, certain vehicles were selected because they had specific characteristics thought to be desirable or undesirable for pedestrian head impacts. For example, the Pontiac Fiero, whose hood underside is shown in Figure 7, was chosen because it has a plastic composite hood and fender design. Choosing the Fiero enabled a preliminary look at the effects that the inevitable increase in fiberglass and plastic materials will have on pedestrian safety. Similarly, a comparison between steel and aluminum hoods was possible using a 1985 Pontiac Sunbird.

Some design variations were also considered in selecting vehicles for testing. Certain cars, such as the Saab 900, have hoods that cover the entire width of the vehicle, thus placing the hood/fender seam on the side of the car, an area generally not involved in pedestrian impacts. The Saab 900 hood is shown in Figure 8. Figure 9 shows an Oldsmobile Ciera which has the more common feature of the hood/fender seam located on top. Both designs were tested to determine if the "full-cover" design provided a significant advantage over conventional hood/fender design. The vehicles used in the study are listed in Table 2.

Each test impact location was determined by a parameter called "wraparound" distance(3) which was based on average pedestrian height values. The wraparound distance was measured from the ground up over the bumper leading edge, to the hood edge, and back to the probable location of head impact on the hood or surrounding area. Figure 10 illustrates this measurement. The vehicle hoods were designated into adult and child regions; for adults, a wraparound distance of 157 to 183 centimeters was used and for children, 75 to 130 centimeters was used. Once the wraparound distance was determined, the impact location was varied across the width of the car being tested.

Vehicle parts tested within the adult and child regions to date are the hood and hood/fender seam. Tests done on unmodified hoods in their normal configuration are "baseline" tests. Two subclassifications used for the hood impacts are "open area" tests and "substructure" tests. An open area test is defined as a test where underlying substructure did not come into contact during the test with the hood or hood reinforcements directly beneath, or very close to, the impact site. Substructure tests are defined as any test

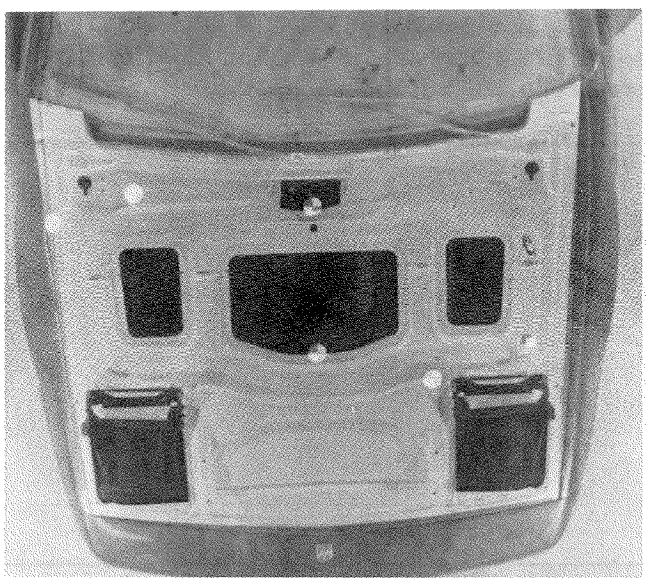

Figure 7. 1985 Pontiac Fiero

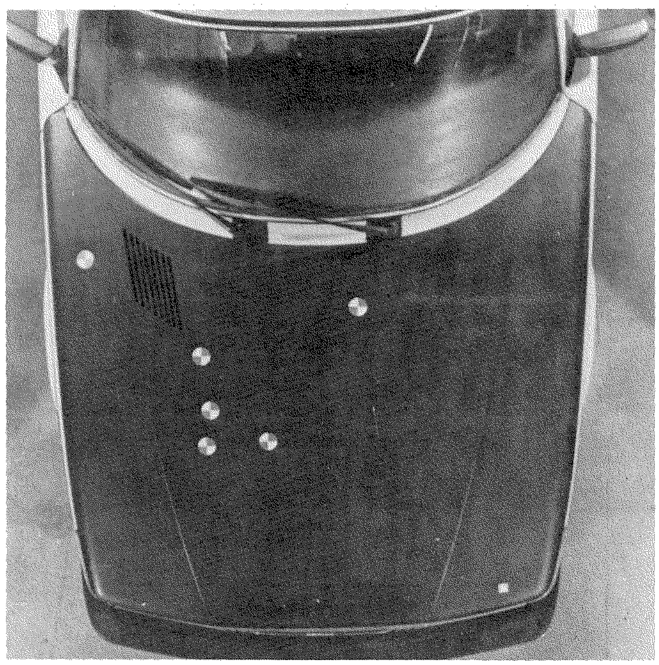

Figure 8. 1983 Saab 900

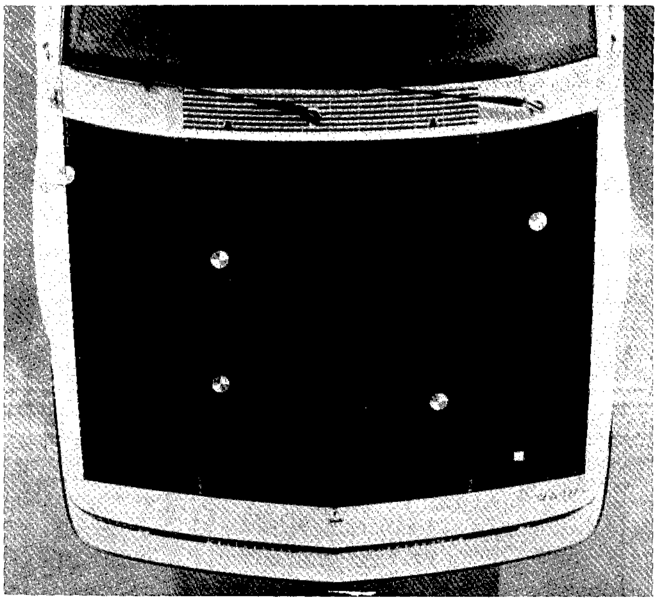

Figure 9. 1985 Oldsmobile Ciera

where an object directly below the impact site, such as an engine block, air cleaner or front suspension strut tower, was contacted during the test. Some tests have also been performed on hoods which were raised 4 inches above the fender. The hood was raised in order to evaluate its performance when absolutely no substructure interference occurred (either directly beneath, or remote from, the impact site).

The head impact velocity for the production car tests was based on a particular vehicle/pedestrian impact speed. Studies indicate ninety percent of all pedestrian auto accidents, and approximately half of the harm suffered by accident victims, occur when the vehicle strikes the pedestrian at 48 kilometers per hour (kph) or less.(1) Moreover, the feasibility of reducing pedestrian trauma by modifying vehicles becomes

Table 2. Production test vehicles

VEHICLE REGION/DESIGN TESTED	VEHICLES USED
HOOD AND CONVENTIONAL HOOD/FENDER SEAM LOCATION	1985 FORD ESCORT 1983 CHEVROLET CHEVETTE 1985 PONTIAC GRAND AM 1983 FORD THUNDERBIRD 1983 SAAB 900 1985 CHRYSLER LEBARON GTS 1985 PONTIAC SUNBIRD 1982 CHEVROLET CAVALIER 1985 OLDSMOBILE CIERA 1984 CHEVROLET CELEBRITY 1985 FORD MUSTANG SVO 1985 PONTIAC FIERO 1983 CHEVROLET CAPRICE
FULL-COVER HOOD	1983 SAAB 900 1983 RENAULT ALLIANCE 1983 ISUZU IMPULSE 1986 BUICK ELECTRA

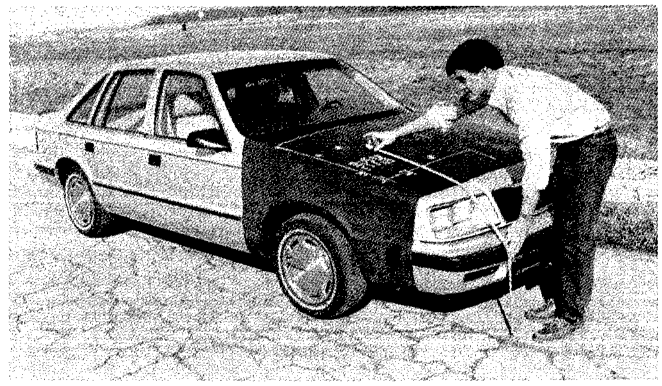

Figure 10. Illustration of wraparound distance measurement

increasingly less as pedestrian- car collisions exceed 48 kph.(4) Therefore, this was the vehicle/pedestrian collision speed chosen for the head impact simulations. Head impact velocity is usually no more than 90% of the vehicle/pedestrian impact velocity(7); hence a head impact speed of 43.2 kph was chosen.

The head masses used in the study were 3.59 kg for an adult test and 2.41 kg for a child test. The decision to use these head masses was based on results found in Reference 8.

Measured Parameters

In addition to obtaining predictions of the probability of death, using the TMSC relationship described above, five basic parameters were measured or calculated after performing a test. These were dynamic hood deflection, percentage of energy absorbed by the hood during impact, head impact acceleration pulse time duration, peak head impact force, and maximum material stiffness.

General Results

Figure 11 shows a wide variation in predicted injury severity to adults as the result of head impacts on production vehicles. Probability of death values range

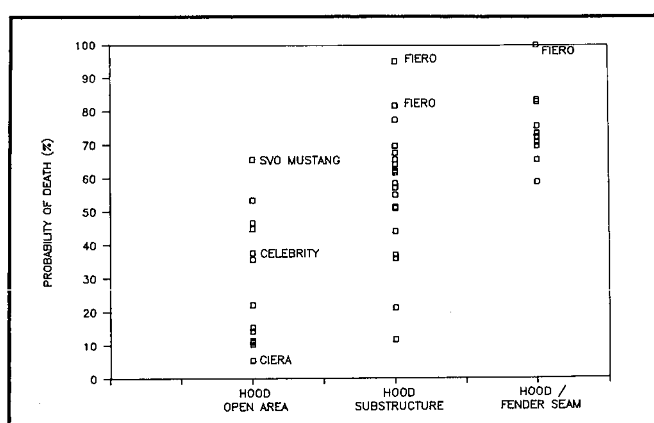

Figure 11. Variation in performance according to the type of impact for the adult tests

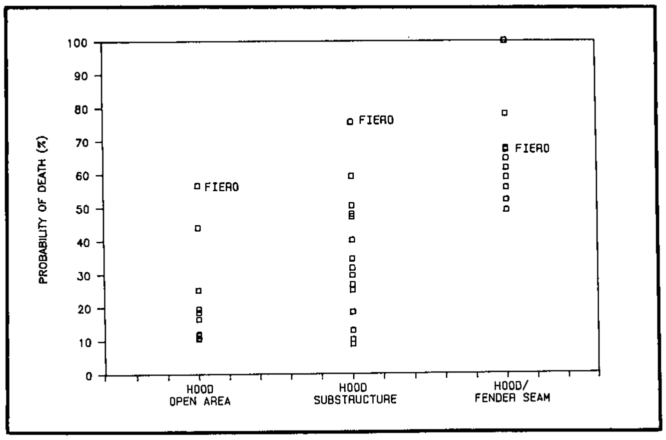

Figure 12. Variation in performance according to the type of impact for the child tests

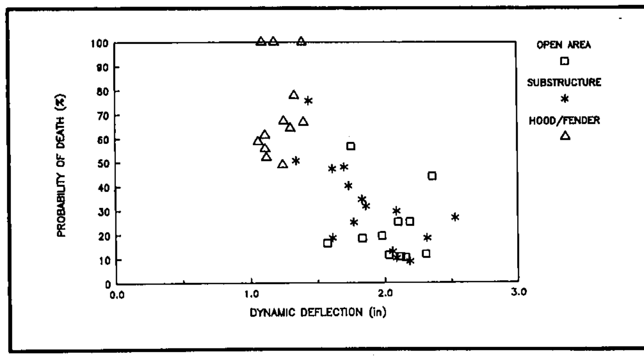

Figure 14. Variation in probability of death with dynamic head deflection for the child tests

from 5 to 100 percent, depending on the impact location. Clearly some existing designs are far less hostile than others. Figure 12, which shows the child data, is similar. (Vehicles identified on Figures 11 and 12 will be discussed in a subsequent section.) Generally, the lowest injury values were seen in tests done on hood open areas, followed by hood tests with substructure contact. Tests in hood/fender regions consistantly produced much greater injury severity. It is obvious, however, that considerable overlap exists among the different impact locations.

Analysis of baseline results revealed that there was some correlation, especially for the adult cases, between probability of death and the following test parameters: dynamic hood deflection, energy absorbed by the hood, peak head impact force, acceleration pulse duration, and maximum material stiffness. The observed correlations describe some of the important characteristics of the least injurious tests

- Dynamic hood deflection has a very strong influence on injury severity, for both adults and children. Figures 13 and 14 show that, on average, a one inch increase in dynamic deflection, in the deflection range between 1 and 3 inches, reduces probability of death by approximately 50%. Open area impacts indicate that achieving dynamic hood deflections of 2.5 inches or greater for adults, or 2 inches for children, can result in probabilities of death as low as 10%.
- It is significant that large variations in injury severity exist for given values of dynamic deflection. Note, for example, that adult open area impacts producing deflections of 2.5 inches result in probabilities of death ranging from 10% to nearly 50%; that adult substructure contacts undergoing just under 2 inches of deflection yield probabilities of death between 35% and 80%; that child contacts in hood/fender regions result in 1.25 inches of stroke but vary in death probability from 50% to 100%; etc. These results clearly demonstrate that significant improvement would be possible without increasing available stroking distance, by simply making all cars perform as well as the best current production cars.
- Increased energy absorption by the hood tends to decrease injury severity for both the adult and child. See Figures 15 and 16. This appears to be a very sensitive parameter in the 95% to 100% energy absorption range.

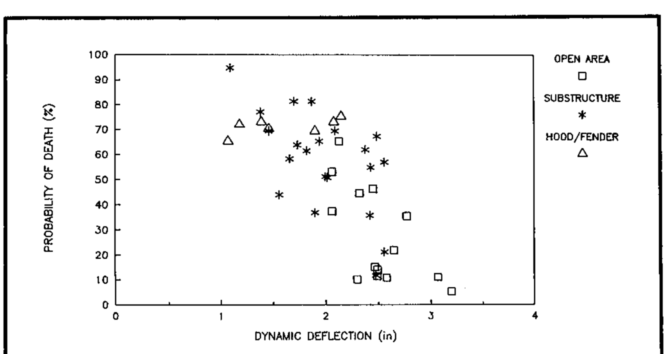

Figure 13. Variation in probability of death with dynamic head deflection for the adult tests

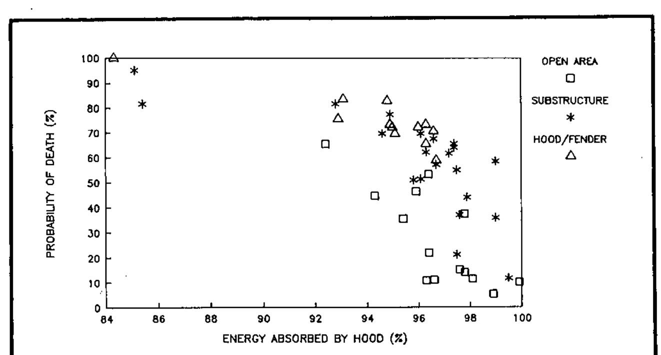

Figure 15. Variation of probability of death with energy absorption for the adult tests

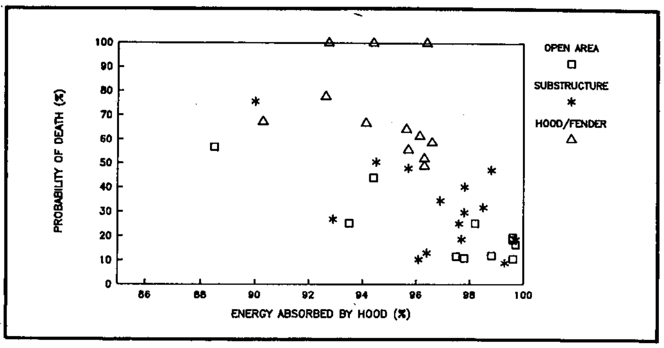

Figure 16. Variation in probability of death with energy absorbtion for the child tests

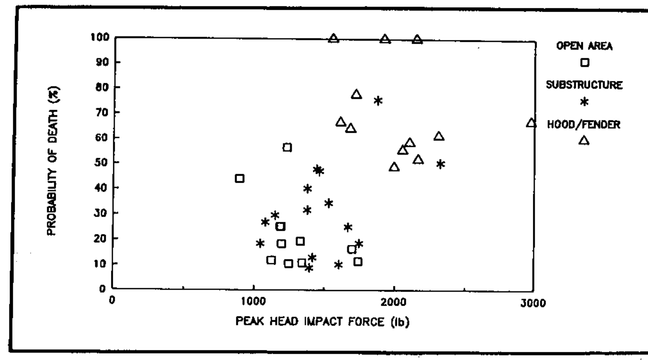

Figure 18. Variation in probability of death with peak head impact force for the child tests

- Figure 17 shows the variation in peak head impact force with injury severity for the adult tests. A peak force of approximately 1600 pounds appears to be a threshold for severe injury. The child data, shown in Figure 18, does not imply a similar threshold.
- Increased head impact acceleration pulse time duration tends to decrease injury severity for both adults and children, as shown in Figures 19 and 20. A pulse duration of 20 milliseconds appears to be a threshold for severe injury according to both the adult and child plots.
- Maximum hood stiffness as it relates to injury is plotted in Figure 21 for the adult cases. The data indicate that as the maximum hood stiffness decreases towards 2000 pounds per inch a lessening in injury severity is likely. The child data, seen in Figure 22, shows a similar, but less well defined, trend.

In general, the baseline test results show the importance of having between 2 and 3 inches of available stroke, avoiding hard strikes against rigid substructure, absorbing nearly all the impact energy, limiting structural stiffness, and maximizing acceleration pulse duration. They also indicate the wide variance in performance that exists in today's fleet. Finally, the

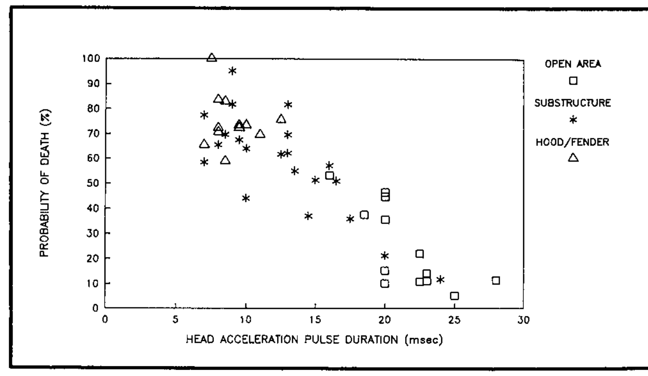

Figure 19. Variation in probability of death with acceleration pulse duration for the adult tests

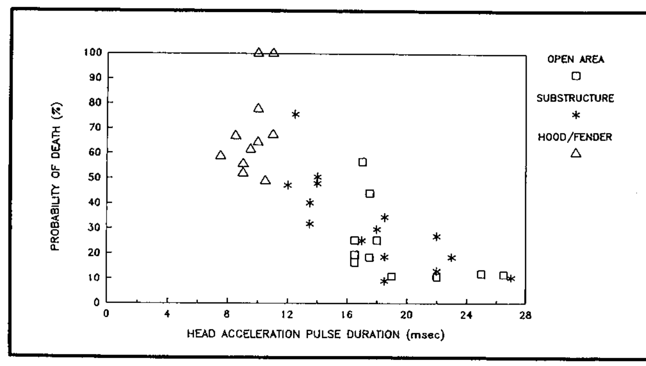

Figure 20. Variation of probability of death with acceleration pulse duration for the child tests

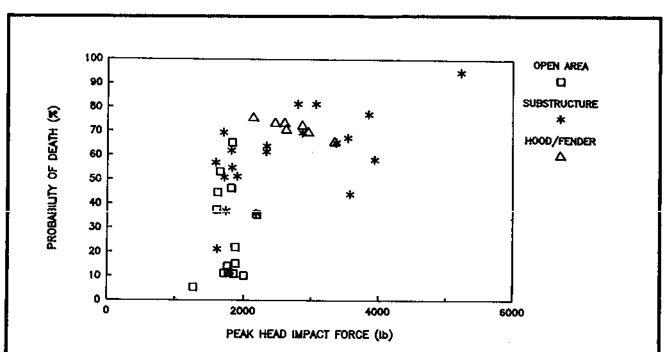

Figure 17. Variation in probability of death with peak head impact force for the adult tests

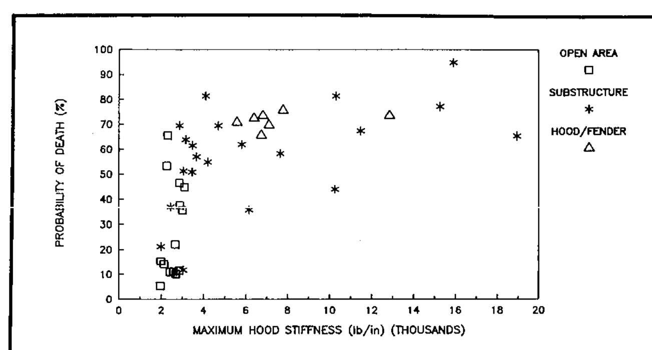

Figure 21. Variation of probability of death with hood stiffness for the adult tests

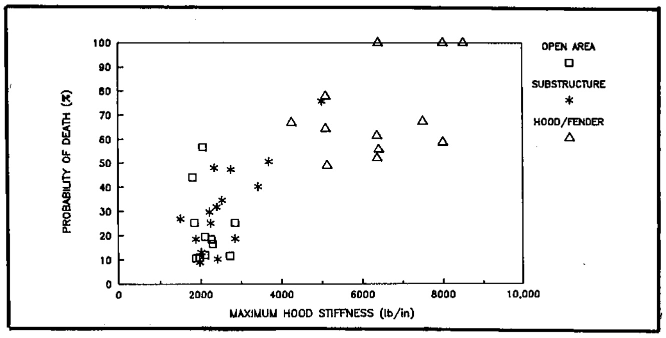

Figure 22. Variation of probability of death with hood stiffness for the child tests

hood/fender interface region appears to be especially hostile, and will require special consideration if total performance of the hood and fender tops is to be improved. Tables 3 and 4 list the adult and child data for all the production car hood and conventional hood/fender seam testing.

Specific Results—Hood Testing

Material Effects. Baseline test results on the Pontiac Fiero indicate that widespread use of fiberglass and plastic composite materials in hood and fender designs may significantly raise the potential for severe pedestrian head injuries. The material characteristics of the

Table 3. Production car test results - adult data.

IMPACT TYPE	TEST NO.	VEHICLE TYPE	TMSC AIS L-R	PROB. OF DEATH (%)	DYNAMIC DEFL. (in)	PEAK FORCE (lb)	PULSE WIDTH (msec)	ENERGY ABSORBED (%)	MAXIMUM HOOD STIFFNESS (lb/in)
HOOD/OA	312	1983 Caprice	5.63	10.13	2.30	1998	20.0	99.9	2696
HOOD/OA	324	1983 Chevette	7.81	53.16	2.06	1643	16.0	96.4	2250
HOOD/OA	298	1985 Grand AM	7.63	46.36	2.45	1815	20.0	95.9	2857
HOOD/OA	322	1985 LeBaron GTS	6.07	14.15	2.49	1759	23.0	97.8	2130
HOOD/OA	292	1985 Sunbird	7.28	35.52	2.77	2187	20.0	95.4	3000
HOOD/OA	295	1985 Ciera	4.78	5.31	3.20	1256	25.0	98.9	1956
HOOD/OA	323	1983 Saab 900	5.79	11.44	2.49	1778	28.0	98.1	2867
HOOD/OA	290	1983 Saab 900	6.65	22.00	2.65	1875	22.5	96.4	2667
HOOD/OA	288	1985 LeBaron GTS	6.17	15.27	2.47	1880	20.0	97.6	1981
HOOD/OA	321	1985 Mustang SVO	8.08	65.27	2.13	1825	16.0	92.4	2295
HOOD/OA	361	1984 Celebrity	7.35	37.47	2.06	1595	18.5	97.8	2885
HOOD/OA	320	1985 Escort	5.72	10.85	2.58	1854	22.5	96.3	2416
HOOD/OA	304	1985 Mustang SVO	7.58	44.63	2.32	1609	20.0	94.3	3103
HOOD/OA	308	1983 Thunderbird	5.75	11.10	3.07	1708	23.0	96.6	2580
HOOD/SS	313	1983 Caprice	5.82	11.70	2.48	1800	24.0	99.5	3043
HOOD/SS	305	1985 Mustang SVO	7.93	58.24	1.66	3941	7.0	99.0	7636
HOOD/SS	358	1985 Fiero	8.37	81.38	1.70	2795	13.0	85.4	4110
HOOD/SS	315	1982 Cavalier	8.16	69.37	2.09	1696	13.0	94.6	2857
HOOD/SS	311	1983 Thunderbird	8.12	67.29	2.49	3538	9.5	96.6	11470
HOOD/SS	319	1985 Sunbird	7.33	36.90	1.90	1735	14.5	97.6	2432
HOOD/SS	286	1985 LeBaron GTS	8.05	63.80	1.73	2332	10.0	97.4	3158
HOOD/SS	291	1983 Saab 900	7.90	56.92	2.56	1578	16.0	96.7	3673
HOOD/SS	299	1985 Grand Am	7.76	51.17	2.00	1896	15.0	96.1	3044
HOOD/SS	360	1985 Fiero	8.57	94.75	1.09	5227	9.0	85.1	15909
HOOD/SS	293	1985 Sunbird	7.56	43.95	1.56	3572	10.0	97.9	10263
HOOD/SS	316	1982 Cavalier	8.16	69.37	1.46	2861	8.5	96.1	4688
HOOD/SS	302	1985 Escort	7.29	35.79	2.42	2180	17.5	99.0	6154
HOOD/SS	309	1983 Thunderbird	7.85	54.80	2.43	1822	13.5	97.5	4200
HOOD/SS	307	1985 Mustang SVO	8.37	81.38	1.87	3063	9.0	92.8	10312
HOOD/SS	301	1985 Escort	8.00	61.42	1.82	2328	12.5	97.2	3478
HOOD/SS	318	1982 Cavalier	8.01	61.89	2.38	1811	13.0	96.3	5806
HOOD/SS	296	1985 Ciera	8.08	65.27	1.94	3365	8.0	97.4	18947
HOOD/SS	327	1983 Chevette	6.60	21.18	2.56	1602	20.0	97.5	1989
HOOD/SS	362	1984 Celebrity	8.30	77.16	1.38	3851	7.0	94.9	15273
HOOD/SS	325	1983 Chevette	7.75	50.79	2.02	1705	16.5	95.8	3461
HOOD/FENDER	317	1982 Cavalier	8.08	65.27	1.07	3337	7.0	96.3	6735
HOOD/FENDER	294	1985 Sunbird	8.16	69.37	1.90	2951	11.0	95.1	7105
HOOD/FENDER	326	1983 Chevette	8.23	73.16	1.39	2458	10.0	94.9	6818
HOOD/FENDER	287	1985 LeBaron GTS	8.18	70.43	1.46	2620	8.0	96.6	5581
HOOD/FENDER	363	1984 Celebrity	8.21	72.06	1.18	2861	9.5	95.0	6383
HOOD/FENDER	289	1983 Saab 900	8.27	75.42	2.15	2133	12.5	92.9	7778
HOOD/FENDER	310	1983 Thunderbird	8.23	73.16	2.08	2598	10.0	94.9	12857
HOOD/FENDER	314	1983 CAPRICE	7.94	58.68	1.18	2810	8.5	96.7	6522
HOOD/FENDER	303	1985 ESCORT	8.23	73.16	1.52	2704	9.5	96.3	6000
HOOD/FENDER	297	1985 CIERA	8.21	72.06	1.43	2842	8.0	96.0	6000
HOOD/FENDER	306	1985 SVO MUSTANG	8.4	83.26	1.36	3970	8.0	93.1	6428
HOOD/FENDER	359	1985 FIERO	8.78	111.15	1.17	4166	7.5	84.3	3600
HOOD/FENDER	300	1985 GRAND AM	8.39	82.63	1.51	2449	8.5	94.8	5567

OA = open area
SS = substructure contact

Table 4. Production car test results - child data.

IMPACT TYPE	TEST NO.	VEHICLE TYPE	TMSC AIS L-R	PROB. OF DEATH (%)	DYNAMIC DEFL. (in)	PEAK FORCE (lb)	PULSE WIDTH (msec)	ENERGY ABSORBED (%)	MAXIMUM HOOD STIFFNESS (lb/in)
HOOD/OA	333	1983 Saab 900	5.81	11.64	2.03	1743.00	26.50	97.50	2727.00
HOOD/OA	331	1985 Mustang SVO	6.83	25.30	2.19	1188.00	18.00	93.50	2857.00
HOOD/OA	344	1983 Caprice	6.42	18.52	1.83	1193.00	17.50	99.60	2273.00
HOOD/OA	346	1985 Celebrity	6.49	19.54	1.98	1329.00	16.50	99.60	2112.00
HOOD/OA	356	1985 Fiero	7.89	56.69	1.75	1227.00	17.00	88.50	2054.00
HOOD/OA	349	1983 Thunderbird	5.85	12.00	2.31	1122.00	25.00	98.80	2113.00
HOOD/OA	351	1985 LeBaron GTS	5.69	10.63	2.16	1246.00	22.00	99.60	1885.00
HOOD/OA	336	1985 Sunbird	6.27	16.52	1.57	1699.00	16.50	99.70	2308.00
HOOD/OA	340	1985 Ciera	6.83	25.30	2.10	1182.00	16.50	98.20	1846.00
HOOD/OA	329	1985 Escort	5.73	10.96	2.12	1343.00	19.00	97.80	1978.00
HOOD/OA	328	1983 Chevette	7.56	44.10	2.36	891.00	17.50	94.40	1800.00
HOOD/SS	348	1983 Chevette	7.04	29.69	2.09	1145.00	18.00	97.80	2222.00
HOOD/SS	337	1985 Sunbird	6.82	25.11	1.77	1669.00	17.00	97.60	2250.00
HOOD/SS	330	1985 Escort	7.67	47.95	1.70	1443.00	14.00	95.70	2337.00
HOOD/SS	338	1983 Cavalier	7.65	47.23	1.61	1463.00	12.00	98.80	2748.00
HOOD/SS	350	1983 Thunderbird	7.44	40.25	1.73	1377.00	13.50	97.80	3428.00
HOOD/SS	352	1985 LeBaron GTS	5.46	8.92	2.19	1394.00	18.50	99.30	1978.00
HOOD/SS	342	1985 Grand Am	5.96	13.05	2.06	1416.00	22.00	96.40	2013.00
HOOD/SS	343	1985 Grand Am	7.24	34.57	1.83	1529.00	18.50	96.90	2535.00
HOOD/SS	334	1983 Saab 900	5.65	10.31	2.09	1605.00	27.00	96.10	2416.00
HOOD/SS	332	1985 Mustang SVO	7.74	50.57	1.34	2322.00	14.00	94.50	3692.00
HOOD/SS	341	1985 Ciera	6.42	18.52	2.32	1043.00	23.00	99.70	1882.00
HOOD/SS	345	1983 Caprice	6.43	18.66	1.61	1749.00	18.50	97.70	2857.00
HOOD/SS	364	1985 Sunbird - AL	6.91	26.89	2.53	1077.00	22.00	92.90	1500.00
HOOD/SS	347	1985 Celebrity	7.13	31.79	1.86	1376.00	13.50	98.50	2400.00
HOOD/SS	357	1985 Fiero	8.27	75.69	1.44	1872.00	12.50	90.00	5000.00
HOOD/FENDER	109	1983 Caprice	7.94	58.68	1.06	2101.00	7.50	96.59	8000.00
HOOD/FENDER	110	1983 Chevette	8.06	64.29	1.30	1679.00	10.00	95.63	5096.00
HOOD/FENDER	111	1985 Ciera	8.00	61.42	1.11	2307.00	9.50	96.13	6369.00
HOOD/FENDER	112	1985 Escort	8.11	66.78	1.40	1608.00	8.50	94.12	4255.00
HOOD/FENDER	113	1982 Cavalier	7.78	51.96	1.12	2161.00	9.00	96.30	6369.00
HOOD/FENDER	114	1984 Celebrity	7.87	55.64	1.11	2051.00	9.00	95.70	6410.00
HOOD/FENDER	115	1985 Grand Am	8.87	100.00	1.18	1917.00	11.00	96.40	6400.00
HOOD/FENDER	116	1985 Sunbird	7.70	48.89	1.24	1992.00	10.50	96.30	5128.00
HOOD/FENDER	117	1985 LeBaron	8.92	100.00	1.39	1550.00	10.00	94.40	8510.00
HOOD/FENDER	118	1984 Mustang SVO	9.10	100.00	1.09	2146.00	10.00	92.73	8000.00
HOOD/FENDER	119	1983 Thunderbird	8.31	77.75	1.33	1717.00	10.00	92.62	5095.00
HOOD/FENDER	031	1985 Fiero	8.12	67.29	1.25	2972.00	11.00	90.29	7500.00

OA = open area
SS = substructure contact
AL = aluminum hood

plastic hood and fenders of the Fiero lead to more serious pedestrian head injuries than those of ordinary sheet metal hoods. Figures 11 and 12 show that the Fiero design is much more injurious than ordinary sheet metal designs regardless of the impact location. Probability of death predictions range from 56 to 100 percent. Figures 23 and 24 show the Fiero design

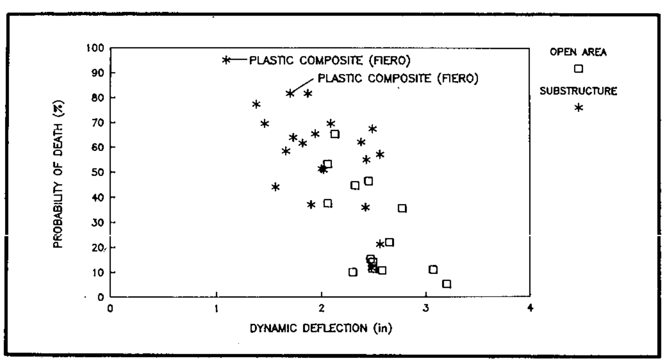

Figure 23. Variation of probability of death with dynamic hood deflection for the adult tests with the Fiero tests identified

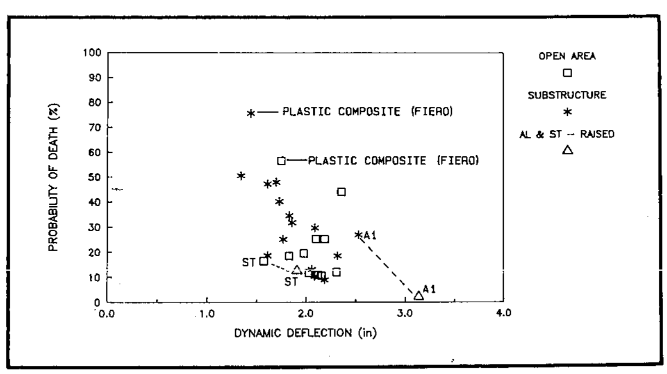

Figure 24. Variation of probability of death with dynamic hood deflection for child tests with raised hood steel and aluminum tests added

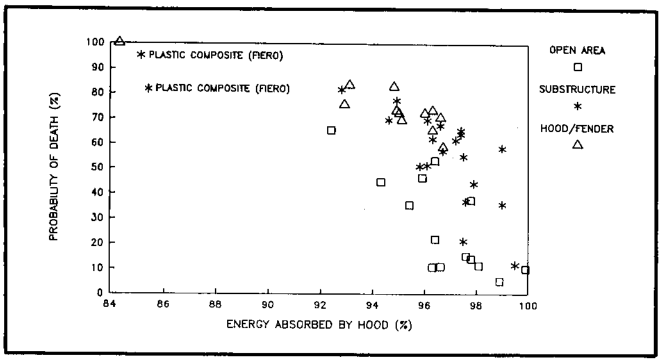

Figure 25. Variation of probability of death with energy absorption for the adult tests

restricting dynamic hood deflection for adults and children, causing increased injury severity. Especially significant is the low energy absorption capability of the Fiero's hood (Figures 25 and 26). It is likely that this is the primary characteristic responsible for the Fiero's poor performance. As plastic vehicle exteriors become more prevalent, it is highly desirable that they be designed to absorb greater amounts of impact energy.

Steel and aluminum hoods on the Pontiac Sunbird are compared in Figure 24 (similar to Figure 14, but with two Sunbird raised hood tests added). The aluminum baseline test predicted a larger deflection but a more severe injury value than the steel baseline test. The aluminum allowed too much deflection in the baseline position, causing substructure to be contacted and, consequently, a larger injury prediction. The raised aluminum test produced a much larger deflection and a lower injury value than the raised steel test, since no substructure contact occurred. Firm conclusions are not possible from steel vs. aluminum comparisons on only one hood configuration; none-the-less, these tests suggest that aluminum hoods may be more flexible and therefore preferable to steel only if sufficient stroking distance were available.

Reinforcement Effects. Differences in adult injury severity between the Oldsmobile Ciera and Ford SVO

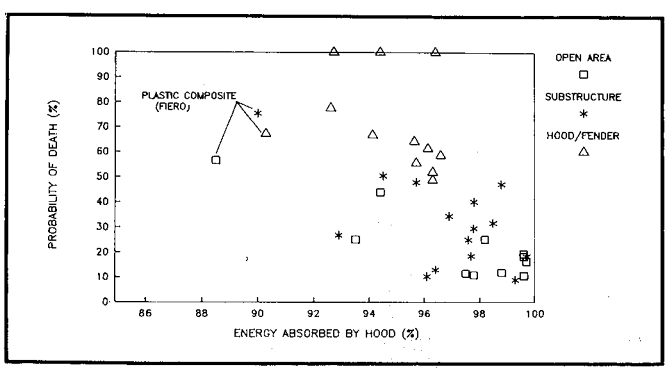

Figure 26. Variation in probability of death with energy absorption for the child tests

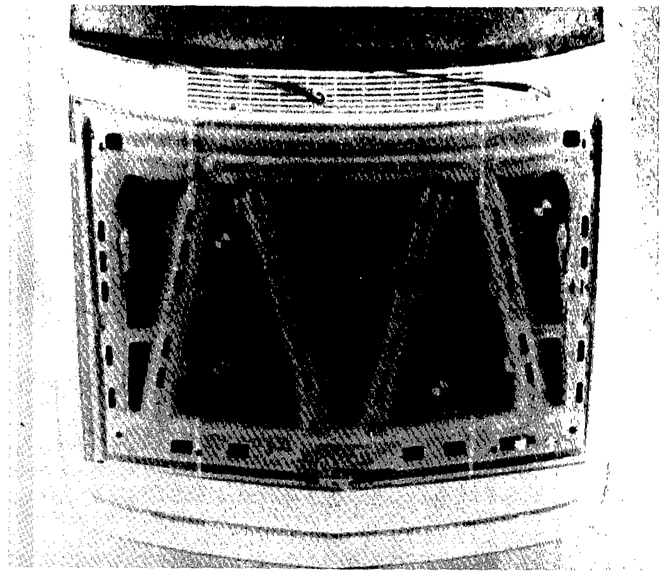

Figure 27. 1985 Oldsmobile Ciera

Mustang hood open areas, as observed in Figure 11, are remarkable. Also, the large difference between Ciera and Chevrolet Celebrity injury severities was interesting, since these two cars are very similar, having exterior hoods nearly identical in size and shape. Visual examination of these hoods suggested the hood reinforcement network was a contributor to these differences. Figures 27 and 28 show the hood reinforcement patterns for the Ciera and Mustang. The Mustang, with its turbo intake vent, has a much more extensive reinforcement network than the Ciera. The Ciera has no major lateral connecting reinforcements. The Celebrity, shown in Figure 29, has a much more extensive reinforcement network than does the Ciera. Reinforcement cross-sections of the Ciera,

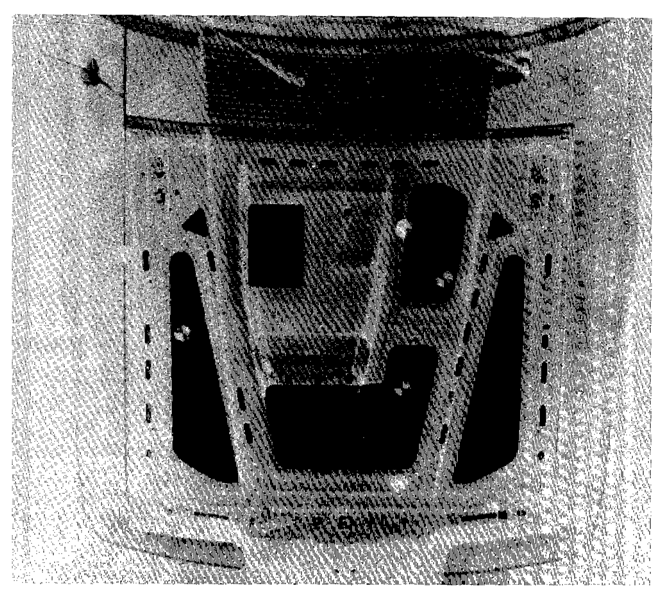

Figure 28. 1985 Ford Mustang SVO

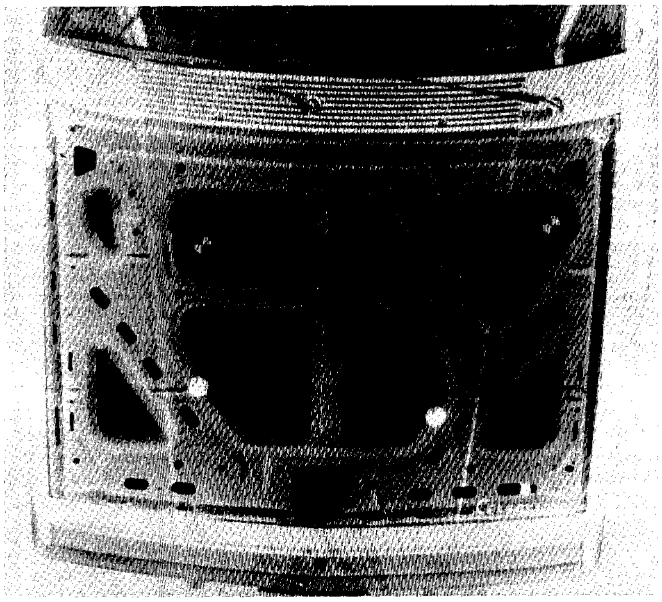

Figure 29. 1984 Chevrolet Celebrity

Mustang and Celebrity are shown in Figure 30. The Ciera cross-section differs from those of the Mustang and Celebrity (and most other cars, as well) by having only about half the depth.

The Ciera baseline test discussed above (no. 295) had an impact location which was within 2 inches of a hood reinforcement. This produced the lowest injury prediction in the baseline hood testing. To further investigate the effect of the Ciera's hood reinforcement, another test (no. 403) was run directly above that reinforcement at a location with the same wrap-around distance as test 295. These tests were repeated

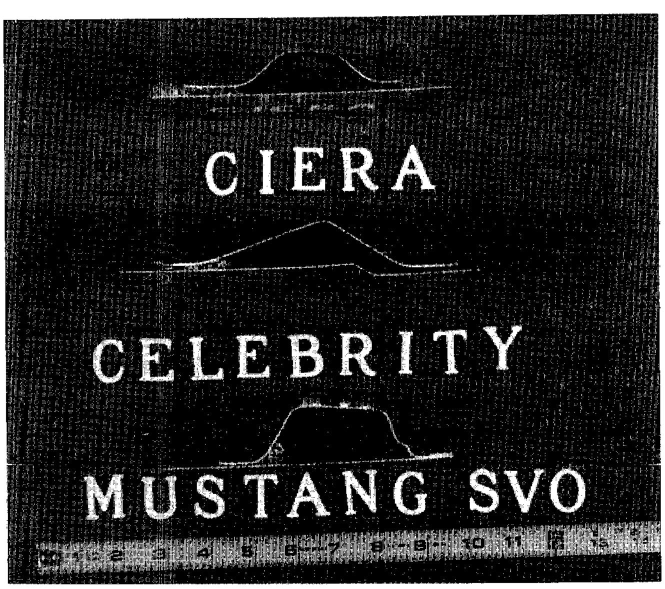

Figure 30. Hood reinforcement cross-sections of 1985 Oldsmobile Ciera, 1984 Chevrolet Celebrity and 1985 Ford Mustang SVO

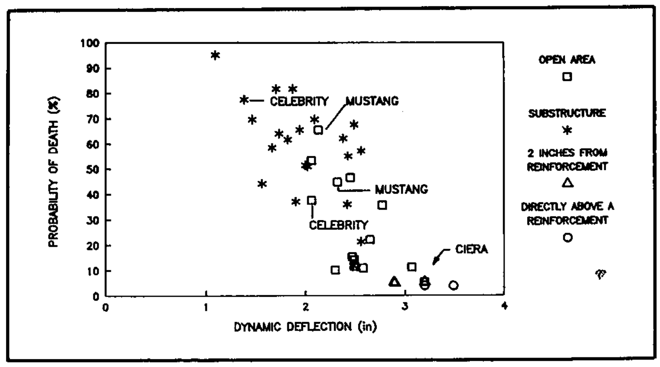

Figure 3l. Variation of probability of death with dynamic hood deflection for the adult tests with Ciera reinforcement tests added

to ensure that results were accurate. Test 401 and 402 were repeats of test 295. Test 426 was a repeat of test 403.

Relationships between tests above the reinforcement and the other baseline tests can be seen in Figure 31. These results clearly indicate that impacting directly over the Ciera style reinforcement does not increase the potential for severe head injury. Dynamic deflection remained constant or increased slightly for the reinforcement test. Although a deflection of 3 inches or more may be impractical for some hood areas, the fact that the reinforcement did not adversely affect the impact remains. It is reasonable to assume that if the Celebrity had a Ciera style reinforcement design it would have produced similar results. Table 5 summarizes reinforcement-evaluation testing results.

Formulation of a "best" hood reinforcement design for the central areas of the hood was based on the testing described above. Since this type of reinforcement is present in a production car the design is production feasible. Figure 32 illustrates a cross-sectional view of this "best" reinforcement. Observa-

Table 5. Ciera hood reinforcement testing results hood open area—adult tests.

Test No./Location	Probability Of Death (%)	Dynamic Hood Deflection (in.)	Energy Absorbed By Hood (%)
295/ 2 in. from hood reinforcement	5.32	3.20	96.5
401/ 2 in. from hood reinforcement	5.15	2.89	99.6
402/ 2 in. from hood reinforcement	4.93	2.90	98.8
403/ directly above a hood reinforcment	3.84	3.20	96.5
426/ directly above a hood reinforcement	3.75	3.49	96.7

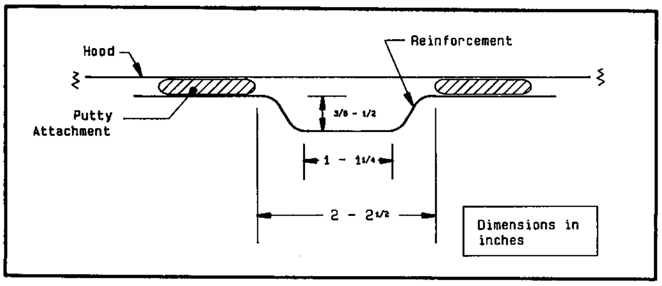

Figure 32. End view of ideal hood reinforcement cross-section

tions from this Ciera cross-section design indicate that it is desirable for the putty attachments to be pliable yet stiff enough to prevent the hood and reinforcement from contacting each other when impacted. The reinforcement sheet metal thickness is .0295 inches and the hood sheet metal thickness is .0306 inches.

Specific Results—Hood/Fender Region Testing

Figure 33 shows the variation in adult probability of death with dynamic hood deflection from impacts in hood/fender regions for the following designs and condition: conventional hood/fender seams, full-cover hoods, full-cover hoods raised 4 inches, and full-cover hoods raised 4 inches with edge reinforcements removed. The best open area hood tests are also shown for comparison. The tests on full-cover hoods demonstrate essentially no improvement over the more conventional hood/fender seam designs. Low injury severities (less than 10% probability of death) were obtained on raised full-cover hoods with edge reinforcements removed only when dynamic deflections were very high (3.5 inches or greater). However, test 382 (on an un-altered full-cover hood raised 4 inches) predicted a relatively low probability of death value of 30% with a more reasonable dynamic deflection of 2.75 inches, approaching the best open area hood results.

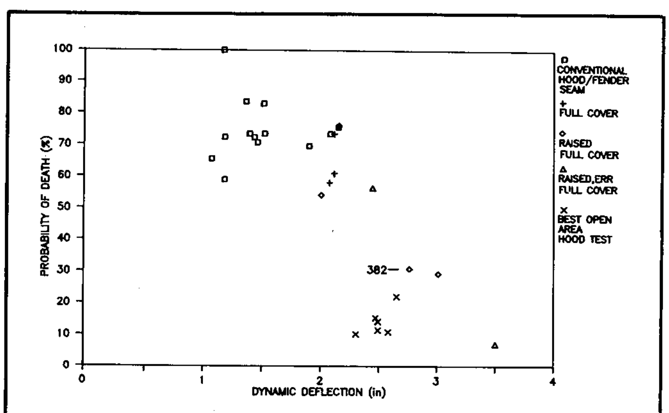

Figure 33. Variation of probability of death with dynamic hood deflection for the hood-fender region—adult tests

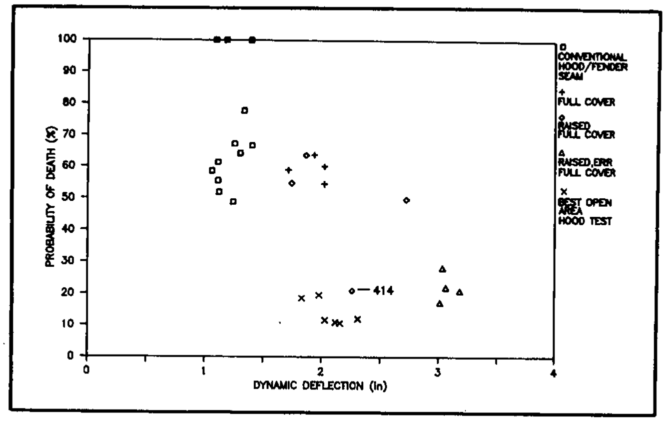

Figure 34. Variation of probability of death with dynamic hood deflection for the hood-fender region child tests

The child head impact simulations are seen in Figure 34. Full cover hoods show no injury reduction benefit over most of the conventional designs, although dynamic deflections are nearly an inch greater. Indeed, deflections were almost the same as for the best open area impacts, where probability of death was as low as 10%, demonstrating that sufficient stroking distance is available. Tests on raised hoods with edge reinforcements removed resulted in reduced injury severity, but excessive deflections. One raised un-altered full-cover hood, however, yielded a 20% probability of death with only 2.25 inches of stroke (Test 414), demonstrating the feasibility of significant improvement in this area.

More work is needed in the hood/fender area. It is clear, however that the full-cover hood design increases the potential of attaining injury values which are equivalent to the less severe values seen in the central areas of the best performing hoods (about 10% probability of death with 2 to 2.5 inches of deflection). Tables 6 and 7 detail the results.

Table 6. Hood/fender region results—adult data.

HOOD & IMPACT TYPE	TEST NO.	VEHICLE TYPE	TMSC AIS L-R	PROB. OF DEATH (%)	PEAK FORCE (lb)	DYNAMIC DEFL. (in.)	ENERGY ABSORBED (%)	MAX. STIFF. (lb/in)	PULSE WIDTH (msec)
CONVENTIONAL HOOD/FENDER DESIGN	294	1985 SUNBIRD	8.16	69.62	2951	1.90	95.1	7105	11.0
	317	1982 CAVALIER	8.08	65.50	3337	1.07	96.3	6735	7.0
	300	1985 GRAND AM	8.39	82.93	2449	1.51	94.8	5567	8.5
	363	1984 CELEBRITY	8.21	72.32	2861	1.18	95.0	6383	9.5
	359	1985 FIERO	8.78	100.00	4166	1.17	84.3	3600	7.5
	326	1983 CHEVETTE	8.23	73.42	2458	1.39	94.9	6818	10.0
	314	1983 CAPRICE	7.94	58.88	2810	1.18	96.7	6522	8.5
	310	1983 THUNDERBIRD	8.23	73.42	2598	2.08	94.9	12857	10.0
	303	1985 ESCORT	8.23	73.42	2704	1.52	96.3	6000	9.5
	289	1983 SAAB 900	8.27	75.69	2133	2.15	92.9	7778	12.5
	287	1985 LEBARON GTS	8.18	70.68	2620	1.46	96.6	5581	8.0
	306	1985 SVO MUSTANG	8.4	83.56	3970	1.36	93.1	6428	8.0
	297	1985 CIERA	8.21	72.32	2842	1.43	96.0	6000	8.0
FULL-COVER	417	1983 ALLIANCE	8.23	73.42	3292	2.11	94.5	1111	10
	410	1983 ISUZU	7.92	57.96	1643	2.07	95.7	3750	17
	289	1983 SAAB 900	8.27	76.15	2133	2.15	92.9	7778	12.5
	409	1983 ISUZU	7.98	60.89	1637	2.11	95.6	3428	13.5
FULL-COVER RAISED	418	1983 ALLIANCE	7.02	29.28	1039	3.01	97.6	1454	24
	382	1983 SAAB 900	7.09	30.84	1490	2.76	99.2	1799	17
	412	1983 ISUZU	7.82	54.05	1710	2	96.2	4444	12.5
FULL-COVER RAISED REINFORCEMENTS MODIFIED	425	1983 SAAB 900	5.15	7.06	944	3.5	98.9	1818	21
	413	1983 ISUZU	7.88	54.46	1411	2.44	97.7	1778	16
	421	1983 ALLIANCE	3.82	2.56	849	4.88	99.0	1200	25

Table 7. Hood/fender region testing—child data.

HOOD & IMPACT TYPE	TEST NO.	VEHICLE TYPE	TMSC AIS L-R	PROB. OF DEATH (%)	PEAK FORCE (LBS.)	DYNAMIC DEFL. (IN.)	ENERGY ABSORBED (%)	MAX. STIFF. K (LB/IN)	PULSE WIDTH (MSEC)
CONVENTIONAL HOOD/FENDER DESIGN	109	1983 CAPRICE	7.94	58.68	2101	1.06	96.59	8000	7
	110	1983 CHEVETTE	8.06	64.29	1679	1.30	95.63	5096	10
	111	1985 CIERA	8.00	61.42	2307	1.11	96.13	6369	10
	112	1985 ESCORT	8.11	66.78	1608	1.40	94.12	4255	9
	113	1982 CAVALIER	7.78	51.96	2161	1.12	96.30	6369	9
	114	1984 CELEBRITY	7.87	55.64	2051	1.11	95.70	6410	9
	115	1985 GRAND AM	8.87	100.00	1917	1.18	96.40	6400	11
	116	1985 SUNBIRD	7.70	48.89	1992	1.24	96.30	5128	11
	117	1985 LEBARON	8.92	100.00	1550	1.39	94.40	8510	10
	118	1984 MUSTANG	9.10	100.00	2146	1.09	92.73	8000	10
	119	1983 THUNDERBIRD	8.31	77.75	1717	1.33	92.62	5095	10
	031	1985 FIERO	8.12	67.29	2972	1.25	90.29	7500	11
FULL-COVER	422	1983 SAAB 900	8.04	63.82	1207	1.93	95.5	1905	11
	411	1983 ISUZU	7.84	54.61	1577	2.02	90.5	4000	15
	406	1986 ELECTRA	7.94	58.92	2374	1.71	97.4	2286	9
	416	1983 ALLIANCE	7.96	60.00	1926	2.02	97.3	1333	11
FULL-COVER RAISED	419	1983 ALLIANCE	7.71	49.75	1059	2.72	96.2	1793	15
	423	1983 SAAB 900	8.04	63.66	1244	1.86	96.1	1569	12
	407	1986 ELECTRA	7.84	54.79	2457	1.74	97.9	2222	11
FULL-COVER RAISED REINFORCEMENTS MODIFIED	424	1983 SAAB 900	6.32	17.24	911	3.01	99.5	1778	17.5
	414	1983 ISUZU	6.58	21.00	1369	2.26	95.6	3750	22.5
	420	1983 ALLIANCE	6.97	28.31	1342	3.03	98.6	1600	15
	415	1983 ISUZU	6.58	21.00	768	3.18	94.8	2105	22.5

Conclusions

1. A rigid variable-mass head impact device was developed. Through accident reconstruction testing, an injury criterion was derived which enables headform test responses to be related to injury severity with a high degree of confidence.
2. A wide variation in predicted head injury severity was obtained among production vehicles. These vehicles were impacted in various hood and fender locations simulating both adult and child pedestrian accidents. This large variation clearly demonstrates that significant injury reduction could be achieved in the accident environment without increasing available stroking distance, by making all cars perform as well as the best current production cars.
3. Head impacts resulting from 30 mph vehicle/pedestrian impact speeds produced 10% probabilities of fatality with 2 to 2.5 inches of dynamic deflections, in hood central areas (the best performing vehicle regions). The least injurious impacts occurred when nearly 100% of the impact energy was absorbed by the vehicle structure, acceleration pulse duration was 20 milliseconds or greater, and structural stiffness did not exceed 2000 pounds per inch.
4. Widespread use of fiberglass and stiff plastic composite materials in hood and fender designs will significantly raise the potential for severe pedestrian head injuries, unless these materials can be designed to absorb much greater amounts of impact energy.
5. Widespread use of a hood reinforcement similar in design to that found in the central areas of the 1985 Oldsmobile Ciera hood will reduce the potential for severe pedestrian head injuries.
6. For head impacts in hood-fender areas the full-cover hood design offers the potential for significant injury reduction over the conventional hood/fender design. Although more work is needed in this area, it appears that injury severities as low as those obtained in the best central hood regions are attainable.

The discussion and conclusions in this paper represent the opinions of the author and not necessarily those of the National Highway Traffic Safety Administration.

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Car to Pedestrian Impact Energies and Their Application to Sub-System Testing

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Abstract

In a pedestrian accident the severity of injury to different parts of the body varies considerably with respect to the shape of the car front. Consequently for the areas of a car important for pedestrian safety (the bumper, bonnet leading edge, and bonnet top) the protection requirements will also vary considerably according to the car shape.

Existing U.K. proposals for guidelines, specify sub-system tests that examine the effectiveness of protection at each of these three locations, by using impactors to represent sections of a pedestrian's body. The sub-system test must match the impact energy at each location, which depends on the shape of the car. The mass of the impactor should be the mass which best represents the pedestrian striking the car at that point.

Experimental data on impact energy and effective mass at each phase of impact has been obtained from full scale tests with an adult pedestrian dummy and a range of simulated car shapes. The results are given in this paper, and are discussed with respect to their implication for sub-system testing. They are also compared with mathematical simulation data used for compiling the basic sub-systems tests.

The results show that bonnet height strongly influences impact energy for all phases of the impact. Bumper height influences the impact energy for bumper and bonnet leading edge contacts.

The effective mass of impact is seen to vary throughout an impact and also with respect to car shape, but suggested values for sub-system impactors are:-

10 kg for single leg to bumper impacts.
16 kg for upper leg to bonnet leading edge impacts.
6.5 kg for head to bonnet top impacts.

Pedestrian protection should be considered at an early stage in the production of a new car. At the design stage, the results in this paper can help in selecting an overall shape and can also be used in conjunction with human tolerances to calculate crush depth and stiffness at the main contact points.

Introduction

In a pedestrian accident the severity of injury to different parts of the body varies considerably with respect to the shape of the car front.

Mathematical computer simulations of cars striking pedestrians(1) have predicted certain trends in the changes of the impact severity that occur with respect to car shape for each phase of a car to pedestrian collision. These trends were used as the basis of guidelines proposing sub-systems tests that examined the effectiveness of pedestrian protection on cars at each phase of an impact (the bumper, bonnet leading edge and the bonnet top)(2). The proposed tests consist of an impactor representing an appropriate segment of a pedestrian striking these car sections at the required test velocity. In the guidelines the requirements of the sub-systems tests were selected to match the impact energy at each phase of impact which depends on the shape of the car front. Values for impactor mass and test velocity were proposed so that the tests could impart the correct input conditions.

The mathematical computer simulation used in the above work was validated for its accuracy on dummy kinematics and impact accelerations(3) but not for impact energy, nor the effective mass of impact. Both of these aspects are important for defining the requirements of sub-systems tests.

In order to obtain experimental information on the effective masses and energies of impact, TRRL have conducted a series of full scale tests on a range of simulated car shapes. Impact velocities, force and acceleration were measured at each impact point and these have been used to derive effective mass and energy at each phase of an impact. This paper summarises initial results of tests at 32 km/h using an adult dummy giving impact values for the upper and lower legs and head. Further research will also include tests using a dummy representing a six year old child and also at increased speeds. The test speed of 32 km/h, which is at the lower end of the range normally used for pedestrian dummy testing, was chosen to prove the test method, with minimal risk of damage to the instrumentation or dummy.

The results are discussed with respect to their implication for sub-systems testing. They are also compared with mathematical computer simulation data although further tests at higher speed are required before absolute values may be compared.

Description of Test Procedure

A range of simulated car shapes were mounted on a trolley to impact a standing adult dummy, see figure 1. The dummy was supported from an overhead gantry and released electrically by the approaching trolley just before impact.

Pedestrian Dummy

The pedestrian dummy used was an Ogle 50th percentile adult male, 1.7 metres tall weighing 75 kg. Modified knee joints were fitted to simulate knee ligaments which gave a more realistic performance(3). Impact was to the left side, with the left leg backwards and the right leg forwards. To give the most satisfactory test conditions, the standing dummy was turned to face 20 degrees towards the car from directly sideways on. This stance gave a near lateral presentation of the legs, but by the time the head contacted the bonnet the body had rotated to near face down presentation of the head without interference from the left shoulder.

Miniature uni-directional accelerometers were attached directly to the steel leg bones under the flesh, their height was adjusted to be in the centre of bumper and bonnet impact points for each car shape. Tri-axial accelerometers were used to record head accelerations.

The Impact Trolley

The impact trolley weighed 1050 kg and consisted of an unsprung steel frame running on railway lines. The nominal test speed was 32 km/h and it was braked from impact at a rate of 0.7 g.

The car shapes simulated included variations in bonnet height, bumper lead, bumper height and vertical depth (see Fig. 2). In all cases the bonnet top sloped downwards to the front at 10 degrees to the horizontal.

Force Measurement

The bumper, front and leading edge of bonnet and rear of bonnet impact areas were all separately mounted on load measuring transducers and covered in dense energy absorbing foam (Plastazote). The thickness of the foam at these locations was 112, 62 and 112 mm respectively which gave long duration force and acceleration responses and stiffness characteristics appropriate to pedestrian protection. The coefficient of restitution of the foam used was low giving little rebound velocity. The long duration time histories and low coefficient of restitution improved the accuracy with which force and acceleration curves could be used to define energy. It also simplified the task of identifying the instant of common velocity during an impact and thereby the maximum energy absorbed by a structure.

Horizontal impact force on the bumper was measured using a pair of load cells.

The bonnet leading edge and the first 0.57 metres of the bonnet top were mounted on a frame with four bi-axial load cells. The load cells measured forces normal to both the bonnet top surface and the leading edge. In the results these forces have been resolved into horizontal and vertical forces.

A section of the rear bonnet top was mounted on three uni-axial load cells to measure force normal to the bonnet surface. The location of the 300 mm square segment was adjusted to coincide with the head impact point for each test. Tests were repeated if necessary to determine the correct location.

Results

The test data was analysed at 0.1 ms time intervals and the results with respect to the different car shapes studied are shown in tables 1 to 12 and figures 3 to 7.

To allow for cross referencing, the results from each run are shown in an identical position in all the tables.

Figures 3 and 4 show plots of all the forces measured on the trolley from two tests, having bonnet heights of 770 mm and bumper leads of 150 and 250 mm respectively. Comparison of the results shows that with a 250 mm bumper lead, significantly higher bumper and reduced bonnet edge contacts occurred together with some increase in head contact.

Figures 5, 6 and 7 show plots of the most significant trends of variation of impact energy with respect to car shape. Bumper and bonnet height are the most important features.

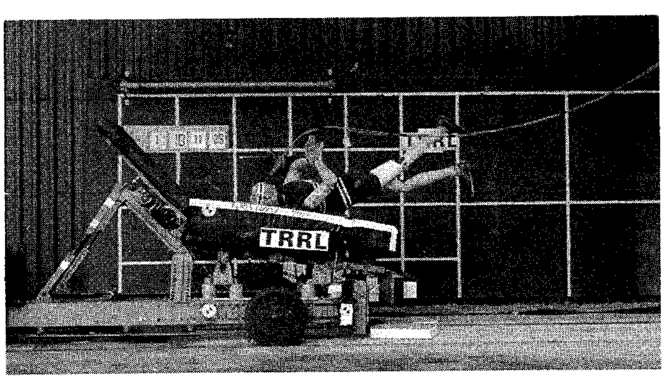

Figure 1. Typical test

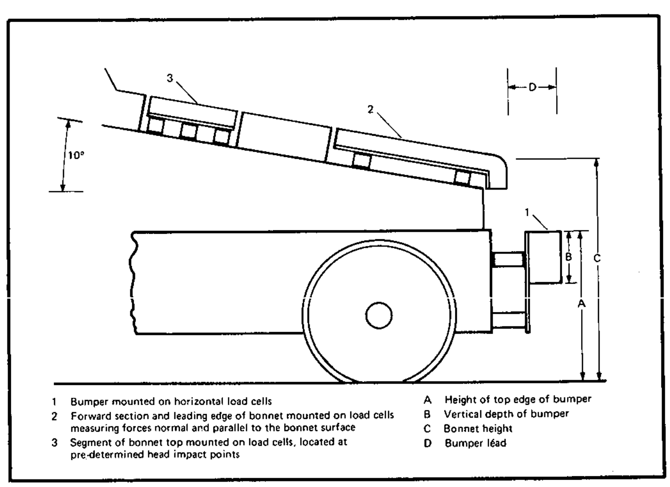

Figure 2. Vehicle simulation

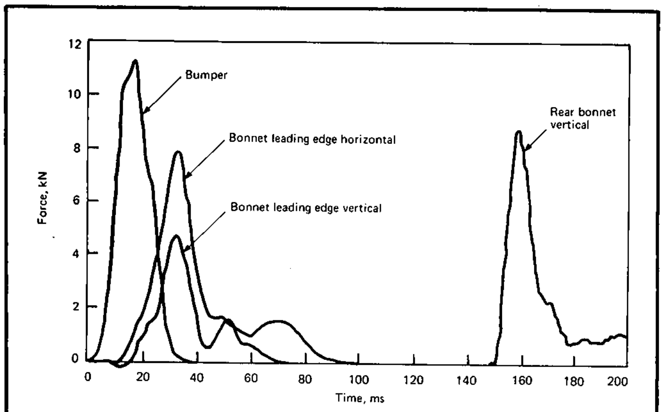

Figure 3. Trolley force time characteristics for test with 770mm bonnet height and 150mm bumper lead

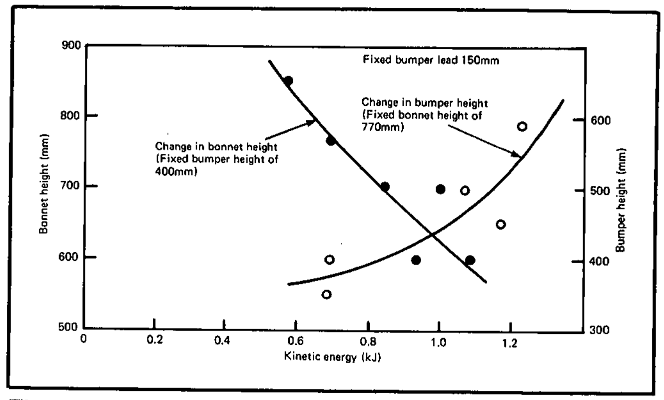

Figure 5. Energy absorbed by bumper with respect to bumper and bonnet height

Table 1 gives the measured impact speed for each test at first impact.

Table 2 shows the total horizontal kinetic energy dissipated by the trolley through the bumper and bonnet edge. These were derived from measuring bumper and bonnet horizontal forces and calculating trolley displacement from impact speed.

Table 3 gives calculated kinetic energy input to the first lower leg struck and the total to both legs.

The values were obtained from the kinetic energy given up by the trolley (Table 2) through the bumper using the basis that when a very heavy body strikes a small stationary body then at the instant of common velocity, approximately half the kinetic energy lost by the large body has gone to structural deformation and half to accelerating the smaller body.

Values for first leg contact were only calculated for the occasions when the individual impact forces of the two legs to the bumper were separated in time.

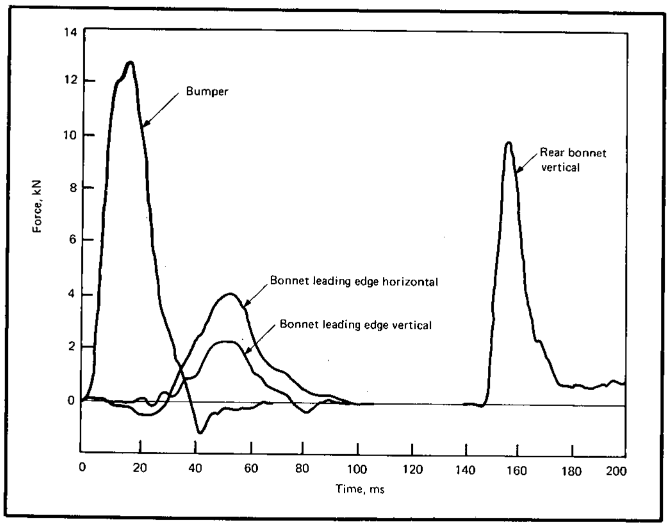

Figure 4. Trolley force time characteristics for test with 770mm bonnet height and 250mm bumper lead

Table 4 gives the calculated average and peak effective mass of the first lower leg struck and the total of both legs. As for Table 3 single leg values were only calculated where individual leg impacts were separated in time.

In both cases the average effective mass for single and double leg contacts was derived from the energy changes shown in Table 3 and velocity change (Table 1).

The peak values of instantaneous effective mass for the first leg struck are shown in brackets. The values were obtained from bumper force and the acceleration of the leg in contact with the bumper. (As there is a danger of dividing by small numbers, the instantaneous effective mass was only calculated up to the time when 90 per cent of impact energy had been transmitted.)

Table 5 shows the velocity change of the section of the upper leg in contact with the bonnet edge and resulting from this contact.

The value was derived from the accelerometer mounted on the upper leg whose position was adjusted to coincide with the centre line of the bonnet edge for each test.

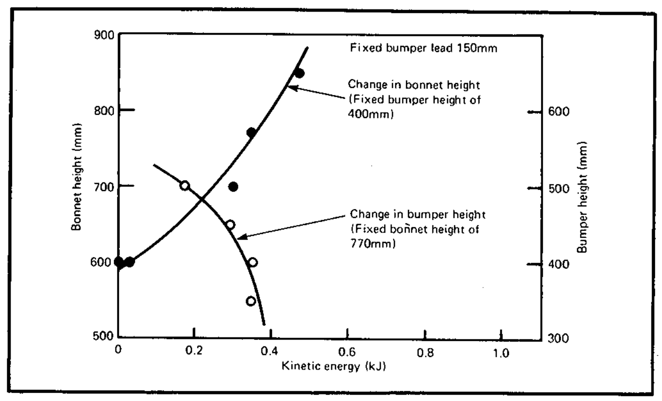

Figure 6. Upper leg kinetic energy from bonnet leading edge impact with respect to bumper and bonnet height

Table 1. Measured trolley impact speed for each test with respect to car shape.

Bonnet Dimensions mm: Height of leading edge	Bumper Dimensions mm: Height of top edge	Bumper Dimensions mm: Vertical depth / Bumper lead	Trolley Velocity m/s: 50	150	250
600			8.6	8.6	
			8.6	8.6	
700	400	200	8.5	8.8	
			8.4	8.3	
770				8.5	8.6
850			8.2	8.1	
770	350	200		8.2	
	400	200		8.5	
	450	120		8.6	
	500	200		8.4	
	590	300		8.5	

Table 6 gives the 3 ms exceedence for the instantaneous resultants of the forward bonnet forces normal to the top surface and leading edge.

Table 7 shows the ratio of the peak horizontal force compared to the peak vertical force on the front section of the bonnet, together with the angular direction of the resultant to the horizontal.

Table 8 gives the kinetic energy imparted into the upper leg from bonnet contact. It was only calculated for the cases where the second leg did not make contact with the bonnet.

The value was obtained from bonnet edge resultant force and upper leg displacement. Upper leg displacement was derived from the output from the leg mounted accelerometer in contact with the bonnet edge.

Table 9 gives the average and peak instantaneous effective masses of the upper leg striking the bonnet edge for the cases where the second leg did not make contact with the bonnet.

The average effective masses were calculated from the impact energies shown in Table 8 and the velocity changes in Table 5.

Table 2. Horizontal kinetic energy dissipated by trolley through bumper and bonnet edge with respect to car shape.

Bonnet dimensions mm: Height of leading edge	Bumper dimensions mm: Height of top edge	Bumper dimensions mm: Vertical depth / Bumper lead	Total horizontal energy dissipated during impact — Through Bumper kJ: 50	150	250	Through Bonnet edge kJ: 50	150	250
600			1.58	1.86		0.74	0	
			1.64	2.16		1.62	0.32	
700	400	200	1.92	2.00		1.34	0.90	
			1.46	1.68		1.38	0.70	
770				1.38	1.98		1.50	0.96
850			1.54	1.14		1.56	1.56	
770	350	200		1.36			1.18	
	400	200		1.38			1.5	
	450	120		2.34			1.04	
	500	200		2.14			0.90	
	590	300		2.46			0.88	

Table 3. Kinetic energy imparted to lower legs by bumper with respect to car shape.

Bonnet Dimensions mm: Height of leading edge	Bumper Dimensions mm: Height of top edge	Bumper Dimensions mm: Vertical depth / Bumper lead	Kinetic energy imparted to lower legs — First Leg kJ: 50	150	250	Both Legs kJ: 50	150	250
600						0.77	0.93	
			0.37	0.37		0.82	1.08	
700	400	200	0.42			0.96	1.0	
						0.73	0.84	
770							0.69	0.97
850				0.29		0.77	0.57	
770	350	200					0.68	
	400	200					0.67	
	450	120		0.46			1.17	
	500	200		0.45			1.07	
	590	300					1.23	

Peak effective masses were derived from the resultants of the front bonnet forces and upper leg acceleration. (As with Table 4 the instantaneous effective mass was only calculated up to the time when 90 per cent of impact energy had been transmitted.)

Table 10 gives the vertical component of head to bonnet impact velocity. Analyses of high speed films of each test (400 frames/sec) showed that at impact, the head was moving vertically downwards. The spinal axis of the head and thorax was essentially horizontal, but the fore and aft axis of the head was inclined sideways at angles of up to 20 degrees from vertical. Consequently the velocities were derived from the lateral and fore and aft headform accelerations only, with a correction factor included to allow for sideways inclination of the head with respect to the vertical. Omitting the spinal head accelerations removed the confusion which could result from forces originating at the lower thorax and transmitted through the neck.

Table 11 gives the kinetic energy lost by the head when striking the bonnet top. It was derived from the

Table 4. Average effective mass of a single lower leg and both lower legs striking bumper, with respect to car shape.

Bonnet Dimensions mm: Height of leading edge	Bumper Dimensions mm: Height of top edge	Bumper Dimensions mm: Vertical depth / Bumper lead	Lower Leg to Bumper Impact Average Leg Mass — First Leg kg: 50	150	250	Both Legs kg: 50	150	250
600			–	–		21.2	25.1	
			9.9(21)	9.9(39)		22.0	29.2	
700	400	200	11.5(18)	–		26.4	25.8	
			–	–		20.7	24.2	
770				–	–		19.0	26.8
850				8.7(30)		22.9	17.4	
770	350	200					20.2	
	400	200					19.0	
	450	120		12.4(30)			31.5	
	500	200		12.6(20)			30.3	
	590	300					34.0	

Note – Peak values are shown in brackets

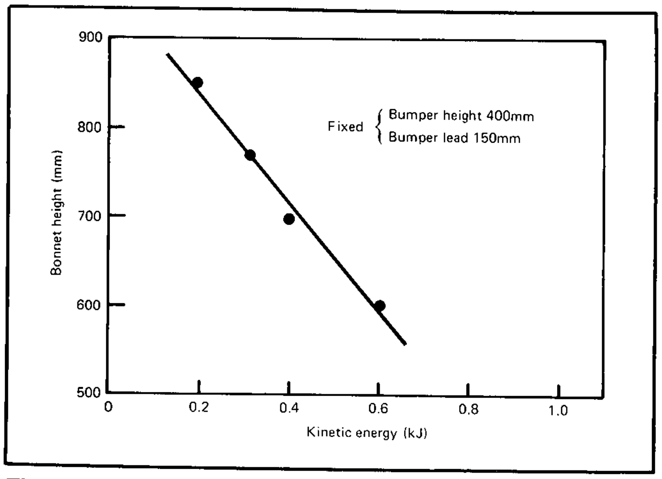

Figure 7. Energy absorbed by bonnet from head impact with respect to bonnet height

force normal to the rear bonnet and the head displacement derived from the headform vertical velocity (Table 10).

Table 12 shows the average headform effective mass and the maximum instantaneous values of effective mass obtained.

The average effective mass was derived from impact energy (Table 11) and impact velocity (Table 10).

Instantaneous values of effective mass were obtained from headform acceleration and bonnet force. (As with table 4 the instantaneous effective mass was only calculated up to the time when 90 per cent of impact energy had been transmitted.)

Comments on Results

The Impact Trolley Velocity

The trolley velocity, measured just before impact, was used to calculate bumper and bonnet leading edge horizontal displacement. The variations in trolley speed (see Table 1) result from small frictional changes in the running gear and braking system.

Table 6. Peak resultant upper leg to bonnet edge force with respect to car shape.

Bonnet Dimensions mm	Bumper Dimensions mm		Resultant upper leg to bonnet edge force · 3ms exceedence kN		
Height of leading edge	Height of top edge	Vertical depth / Bumper lead	50	150	260
600			3.4 2.9	0 1.2	
700	400	200	7.9 8.4	5.3 3.5	
770				8.9	4.6
850			15.1	11.9	
770	350	200		9.4	
	400	200		8.9	
	450	120		8.2	
	500	200		5.3	
	590	300		4.3	

A nominal impact velocity of 32 km/h was the aim for all tests. This value which is at the lower end of the range normally used for pedestrian dummy tests was chosen to enable the dominant trends to be identified, with minimal risk of damage to the instrumentation or the dummy.

Bumper Contact

The kinetic energy lost by the trolley during each impact shown in Table 2, accelerates the dummy and also deforms the impact surfaces.

The total kinetic energy lost (addition of bumper and bonnet energies) ranged between 1.86 kJ for a 600 mm high bonnet and 3.38 kJ for a 770 mm high bonnet. Generally the total value reduces as the bumper lead increases.

With short bumper leads the energy lost was approximately evenly distributed between the bumper and the bonnet edge, but increased bumper lead combined with a low bonnet or increasing the bumper height both increased the influence of the bumper and reduced that of the bonnet edge.

Comparisons of the kinetic energies imparted to the first lower leg struck (where this could be isolated)

Table 5. Velocity change of upper leg from impact with bonnet leading edge, with respect to car shape.

Bonnet Dimensions mm	Bumper Dimensions mm		Upper Leg to Bonnet Impact Velocity m/s		
Height of leading edge	Height of top edge	Vertical depth / Bumper lead	50	150	250
600			8.9 8.0	No-contact 1.8	
700	400	200	– 7.8	5.5 4.5	
770				7.0	4.4
850			8.1	7.6	
770	350	200		7.5	
	400	200		7.0	
	450	120		6.5	
	500	200		4.6	
	590	300		5.8	

Table 7. Ratio of horizontal to vertical forces on front section of bonnet and angular direction of their resultant with respect to horizontal.

Bonnet Dimensions mm	Bumper Dimensions mm		Bonnet Edge Forces: Ratio $\frac{\text{horizontal}}{\text{vertical}}$			Angular direction of resultant from horizontal degrees		
Height of leading edge	Height of top edge	Vertical depth / Bumper lead	50	150	250	50	150	250
600			2.2 2.4	n/c 2.0		24 23	n/c 26	
700	400	200	2.2 3.4	1.8 2.2		24 16	29 24	
770				2.7	2.5		20	22
850			4.3	3.6		13	15	
770	350	200		2.4			23	
	400	200		2.7			20	
	450	120		2.0			26	
	500	200		1.9			28	
	590	300		2.2			24	

and the total to both lower legs, Table 3, showed that between 34 and 50 per cent of the total kinetic energy to the lower legs was received by the first leg to be struck. Generally the percentage of the energy received by the first leg increases as the bonnet edge is raised and this probably results from the impact occurring closer to the pelvis.

For both legs the energy is approximately constant for all cases with a bumper lead of 50 mm, but for a lead of 150 mm the kinetic energy increases as the bumper height is raised or the bonnet height is lowered, Fig. 5.

During each impact the effective mass of the first lower leg struck, where this could be isolated, started at a low value and increased to the peak values shown in Table 4. These were up to four times the average effective mass for the first leg struck, (where these could be determined) and these average values ranged between 8.7 and 12.6 kg.

The average values, for both legs ranged from 17 kg for a high bonnet with a 150 mm bumper lead increasing to approximately 30 kg as the bonnet edge is lowered or the bumper raised. With a short bumper lead the effective mass was with one exception 21 to 23 kg.

Bonnet Leading Edge Contact

The velocity of contact with the bonnet edge, Table 5, was greatest with a short bumper lead, where it approximated to trolley impact speed. It reduced to zero as the bumper lead increased and the bonnet edge was lowered. Raising the bumper height also reduced the impact velocity by up to 20 per cent.

The resultant impact force to the bonnet edge, Table 6, was a maximum at 15 kN for a 850 mm high bonnet and low bumper lead of 50 mm. Increasing the bumper lead and reducing the bonnet height to 600 mm reduced this impact force to zero. Raising the bumper height also caused a reduction from 9.4 kN to 4.3 kN. For the tests with an 850 mm high bonnet, the impact was to the top of the femur adjacent to the pelvis.

The ratio of the horizontal to vertical components of the bonnet edge force, Table 7, shows that the horizontal force was dominant and reached a maximum of 4.3 times the vertical force with a high bonnet and short bumper lead. The horizontal force decreased to 1.8 times the vertical force as the bumper lead was increased and the bonnet edge was lowered. Correspondingly the angular direction of the resultant of these forces ranged from 13 degrees to the horizontal with a high bonnet and short bumper lead to 29 degrees with a low bonnet and increased bumper lead.

Where the kinetic energy imparted to the upper legs could be determined, Table 8, the values closely followed the trend identified for bonnet edge impact velocity; that is, it was a maximum of 0.53 kJ with an 850 mm bonnet. Either lowering the bonnet or increasing the height of the bumper reduced the impact energy, Fig. 6. There was insufficient evidence to show the influence of a 50 mm bumper lead.

Table 9 gives the average and peak values of effective mass of the upper leg. The average effective mass, ranged from 16.3 kg for a high bonnet down to zero for a 600 mm bonnet with a 150 mm bumper lead. Reducing bumper height gave a reduction in the average effective mass from 16.1 to 12.4 kg for bumper top edge heights of 590 to 350 mm respectively.

Peak effective masses were generally between 1.5 to 3 times the average effective mass.

Head to Bonnet Impact

With a 150 mm bumper lead, the head impact velocity, Table 10, increased as the bonnet was lowered, ranging from 8.9 to 13.6 m/s for bonnet heights of 850 to 600 mm respectively.

No distinct changes in head velocity were observed from changes in bumper height or in those tests with a 50 mm bumper lead.

Table 8. Kinetic energy change of upper legs to bonnet edge impact with respect to car shape.

Bonnet Dimensions mm	Bumper Dimensions mm		Kinetic energy change of upper legs from bonnet edge impact kJ		
Height of leading edge	Height of top edge	Vertical depth \ Bumper lead	50	150	250
600	400	200	–	0	
			–	0.03	
700			–	0.30	
			0.24	–	
770				0.35	–
850			0.53	0.47	
770	350	200		0.35	
	400	200		0.35	
	450	120		0.29	
	500	200		0.17	
	590	300		–	

Table 9. Average mass of upper leg striking bonnet edge with respect to car shape.

Bonnet Dimensions mm	Bumper Dimensions mm		Upper leg to bonnet edge impact average leg mass kg		
Height of leading edge	Height of top edge	Vertical depth \ Bumper lead	50	150	250
600	400	200	–	0	
			–	18.5(23)	
700			–	19.8(33)	
			7.9(22)		
770				14.3(34)	–
850			16.2(43)	16.3(46)	
770	350	200		12.4(41)	
	400	200		14.3(34)	
	450	120		13.7(31)	
	500	200		16.1(32)	
	590	300		–	

Note – Peak values are shown in brackets

Table 10. Head to bonnet vertical impact velocity with respect to car shape.

Bonnet Dimensions mm	Bumper Dimensions mm		Head to Bonnet Impact Velocity m/s		
Height of leading edge	Height of top edge	Vertical depth \ Bumper lead	50	150	250
600			9.7	–	
			9.0	13.6	
700	400	200	11.4	11.6	
			11.7	–	
770				10.8	11.7
850			9.2	8.9	
770	350	200		10.8	
	400	200		10.8	
	450	120		10.1	
	500	200		10.3	
	590	300		11.5	

Head impact energy, Table 11, showed a similar pattern to the changes in head velocity. At a bumper lead of 150 mm the impact energy ranged from 0.19 kJ at a bonnet height of 850 mm to 0.60 kJ for a 600 mm high bonnet, Fig. 7. Again neither changes in bumper lead nor in bumper height showed any distinct changes in impact energy.

During each head impact the effective mass of the head increased approximately linearly from zero at initial contact to the peak values shown in Table 12.

Average values of effective mass were with one exception between 60 to 70 per cent of the peak values.

The greatest average value was 6.5 kg for a 600 mm bonnet and 150 mm bumper lead. Increasing the bonnet height to 850 mm reduced the mass progressively to 4.7 kg. Raising the bumper height gave a slight increase in effective mass; changing the bumper lead generally gave small changes.

Discussion

The results show that at each point on a car striking a pedestrian, (bumper, bonnet leading edge or bonnet top) the effective mass of the pedestrian contact varies throughout an impact and also the average effective mass varies with respect to the shape of the particular car involved.

The velocity of impact to the bonnet leading edge and the bonnet top also vary considerably with respect to the shape of the vehicle.

The changes in these quantities follow a trend which may be used to specify the requirements of sub-system tests.

Clearly for component testing it would be impracticable to try to represent all changes in effective mass, but satisfactory test procedures can be developed around three impactors (one each for the bumper, bonnet leading edge and bonnet top). The test velocity chosen, together with the impactor mass would conform to the impact energy requirement for the section of the car under test and vehicle shape.

Sub-system tests on this basis would check the structure for energy absorbing capability, but a drawback would be that in some cases velocity sensitive material would not be assessed at the correct velocity.

Bumper Test

When specifying a bumper test the decision must be taken whether the impactor represents the mass of one leg, or of both. If it is assumed to simulate one leg (approximately 10 kg), this implies that the second leg strikes the bumper at a different location from that of the first. Alternatively both legs may be represented as striking the bumper together, one on top of the other in which case an impactor of just over 20 kg would be appropriate.

The results show that for the car shapes studied, bumper sub-systems test energy should be varied with respect to bumper and bonnet heights.

Bonnet Leading Edge Test

For the tests reported in this paper with the dummy initially standing sideways on to the approaching car, the impact to the bonnet edge essentially resulted in

Table 11. Kinetic energy of vertical head impact to bonnet with respect to car shape.

Bonnet Dimensions mm	Bumper Dimensions mm		Head to Bonnet Impact Energy kJ		
Height of leading edge	Height of top edge	Vertical depth \ Bumper lead	50	150	250
600			–	–	
			0.23	0.60	
700	400	200	0.37	0.4	
			0.38	–	
770				0.31	0.41
850			0.22	0.19	
770	350	200		0.3	
	400	200		0.31	
	450	120		0.27	
	500	200		0.32	
	590	300		0.38	

Table 12. Average and peak mass of head striking bonnet top with respect to car shape.

Bonnet Dimensions mm	Bumper Dimensions mm		Head to Bonnet Impact Head Mass					
			Peak Mass kg			Average Mass kg		
Height of leading edge	Height of top edge	Vertical depth \ Bumper lead	50	150	250	50	150	250
600			–	–		–	–	
			9.0	11.0		5.6	6.5	
700	400	200	8.0	10.0		5.7	5.9	
			9.0	–		5.6	–	
770				8.5	14.0		5.3	6.0
850			8.0	6.5		5.0	4.7	
770	350	200		8.5			5.1	
	400	200		8.5			5.3	
	450	120		7.5			5.3	
	500	200		9.5			6.0	
	590	300		10.5			5.7	

contact with one leg only. In cases when the second leg made contact, the impact energy involved was relatively low. An impactor representing the upper part of a single leg would therefore be appropriate having a mass of about 16 kg.

The relative magnitudes of the bonnet leading edge horizontal and vertical components of force are shown in Table 7 to vary with respect to car shape. The bonnet leading edge sub-systems tests would therefore be more representative if the impact directions varied with respect to car shape to reflect the direction of resultant force, shown in Table 7 to range between 13 and 29 degrees to the horizontal depending upon the bonnet height and the bumper lead.

The results show that for the car shapes studied, sub-systems test energy should be varied with respect to bumper and bonnet height.

Rear Bonnet Test

The most appropriate head mass for rear bonnet sub-systems test would be 6.5 kg which was the maximum average effective mass found in the tests.

If a lighter headform was used as a standard then for large head impact energy test conditions a very high headform impact velocity would be necessary which may be difficult for standard test houses to achieve.

The results show that for the car shapes studied rear bonnet sub-system test energy should be varied with respect to bonnet height.

Impact Energy

When determining the impact energy performance for each sub-system test, the requirement is to specify the energy absorbed by the vehicle structure.

In the case of the bumper and bonnet top the required values are approximately equal to the kinetic energy change of the lower leg and the headform respectively and typical values are given in Tables 3 and 11.

In the case of the bonnet leading edge, because of the nature of the impact, the exact value of energy absorbed in deforming the structure may be difficult to determine experimentally. Further work is needed to isolate the vertical components of the impact energy, but a simpler method may be to use mathematical computer simulations.

Validation of Computer Simulation

Test recommendations for assessing pedestrian protection performance of cars(2), were based on computer simulation(1). Appendix 1 compares the results from tests given in this paper with those from the computer simulations.

Conclusions

The input conditions for sub-systems tests representing pedestrians struck by cars must be properly selected for each shape of car front. The presently reported work is a contribution towards this and concludes that:-

(1) These results support the general trends in changes of impact energy with respect to car shape, previously reported from mathematical simulation(1) (2), and used as guidelines in proposed sub-system tests, although some tests at 40 km/h will be necessary before the absolute values may be compared.

(2) In the adult tests at a nominal test speed of 8.3 m/s the kinetic energy imparted to the lower legs by the bumper ranged between 0.29 to 0.45 kJ for a single leg contact and between 0.57 and 1.23 kJ for a double leg contact. The average effective mass ranged between 8.7 to 12.6 kg for a single leg and between 17.4 and 34 kg for both legs.

(3) For the bonnet leading edge the peak vertical forces ranged between 23 and 55 per cent of the peak horizontal forces. The kinetic energy imparted to the upper legs ranged between 0 and 0.53 kJ and the average effective mass between 0 and 19.8 kg.

(4) For head to bonnet contact, the head impact velocity ranged between 8.9 and 13.6 m/s, the head impact energy between 0.19 and 0.6 kJ, and the corresponding average effective mass between 4.7 and 6.5 kg.

(5) Variations in the impact energies (Figs. 5 to 7) showed clear trends with respect to bonnet height for all phases of the impact and with respect to bumper height for the bumper and bonnet leading edge contacts. This strong relationship between impact energy and vehicle shape may be used as a basis for determining the test requirements of sub-system tests.

There is insufficient evidence to show if full scale testing supports the lesser trends identified for bumper lead in the computer simulation, and further research will be necessary to study this aspect.

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Appendix

Validation of Computer Simulation

Comparisons of the present tests at 8.3 m/s with the mathematical simulations at 11.1 m/s impact previously reported(1)(3) show the following:

(a) The average effective masses of impact were similar. For the lower leg to bumper contact the effective masses ranged from 8.7 to 12.6 kg for the full scale tests and 8 to 10 kg for the simulation.
For the upper leg striking the bonnet leading edge, the effective mass ranged between 7.9 and 19.8 kg for the tests and between 5 and 16 kg for the simulation.
In the tests the effective mass of the headform striking the bonnet top ranged between 4.7 and 6.5 kg which compares with the 4.9 kg deadweight of the dummy head and the 6.45 kg deadweight of the dummy head plus neck.

(b) With respect to energy distribution, the full scale tests show the bumper absorbed between 0.57 and 1.17 kJ for tests at 8.3 m/s. For the computer modelling of simulations at 11.1 m/s the corresponding values were 0.28 to 1.17 kJ.
For the bonnet leading edge no comparative values are available. However, the variation of energy absorbed by the bonnet edge with respect to car shape, obtained by computer simulation, is of a similar form to the variations of upper leg to bonnet impact velocity and force, obtained by full scale test.

(c) Head impact velocities were generally similar in test and simulation.
They ranged from 8.9 to 13.6 m/s in the full scale tests at a test speed of 8.3 m/s, and between 8.1 to 15.8 m/s for computer modelling at 11.1 m/s.
In both sets of results the variations in head impact velocity with respect to car shape follow similar trends, with the exception of the cases involving a 600 to 700 mm high bonnet and a 50 mm bumper lead.

Experimental Study of Thoracic Injury in Child Pedestrians

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Abstract

A substantial portion of the motor vehicle-related injuries in the United States are sustained by pedestrians; pedestrians comprise about 16 percent of the total number of traffic fatalities each year. It has been determined that an important classification of pedestrian injuries is thoracic injuries in children struck by the vehicle face. The NHTSA is currently conducting a research program to investigate child pedestrian thoracic injuries. The planned approach to solving the problem involves three phases. The first phase is to reconstruct actual pedestrian accidents using a child thoracic surrogate device. Therefore, such a surrogate device had to be developed. This paper describes the development of a family of child lateral thoracic impact surrogate devices representing 3, 6, 9, and 12 year old children. The masses of the devices are adjustable such that the thoraces of all ages of children between 2 and 13 years old can be simulated. The devices have all performed favorably thus far.

The next step in the first phase of the program is to use the child thoracic surrogates to perform the reconstruction tests. This testing, which has already begun, involves reproducing actual pedestrian accident cases in which children sustained thoracic injuries resulting from contact with the vehicle face. From these tests, an injury criterion can be determined for the thoracic surrogates and a methodology for reproducing pedestrian accidents in the laboratory can be verifiably developed.

The second phase of the program will then involve testing a representative sample of current production cars and determining their potential for causing or preventing pedestrian injuries. The final phase of the NHTSA pedestrian injury study will be to incorporate the knowledge gained from the first two phases into developing practical vehicle face designs which will alleviate thoracic injuries to struck pedestrians.

Introduction

As part of its Advanced Pedestrian Protection Program, the National Highway Traffic Safety Administration (NHTSA) is conducting a research program aimed at reducing the severity of pedestrian thoracic injuries inflicted by vehicle front ends[1].

Based on the accident data analysis of this program, it has been determined that pedestrian accidents involving children being struck in the thorax area by the vehicle face at vehicle/pedestrian impact speeds of 48 kph (30 mph) or less are of major importance. The overall objective of the research addressed in this paper is to examine the causes of thoracic injuries in child pedestrians with the ultimate goal of developing practical countermeasures to reduce pedestrian injuries. This pedestrian thoracic injury research program has a parallel program currently in progress to investigate another important type of pedestrian injuries, that being head injuries[2].

The methodology chosen for this research is threefold. The first step is to reconstruct actual pedestrian accidents involving thoracic injuries to children. This procedure provides a method to verify that pedestrian accidents can accurately be simulated in the laboratory, including both checking the performance of the thoracic surrogate device(s) used and developing an injury criterion for use with the device. The second step in this research program is to test currently available production vehicles to determine which factors make certain vehicle face configurations either friendly or harmful to struck pedestrians. The final step is to incorporate the information gathered in the previous two steps and develop prototype vehicle face designs which would reduce pedestrian injuries. The work completed to date includes the development of several thoracic surrogate devices and preliminary accident reconstruction testing with those devices. This paper describes the development of the latest generation of child thoracic surrogate devices and outlines the reconstruction testing being performed with those devices. Also presented is an overview of the future production vehicle testing and injury reducing design phases of the program.

Side Impact Overview

In this study it was assumed that most pedestrians struck by motor vehicles are travelling in a direction perpendicular to the striking vehicle at the time of impact. Therefore, the child thoracic surrogate devices developed for this program have been designed to simulate lateral impact. This section provides an overview of the work performed to date in the field of lateral thoracic impact surrogate development.

Two basic approaches have been taken in the development of lateral thoracic impact devices. The first is to connect two masses by a spring-mass system as shown schematically in Figure 1. A small mass is used to represent the effective mass of the ribs while a larger mass is used to account for the remaining effective mass of the thorax. The two masses are connected by some combination of springs and dampers which simulate the stiffness and damping charac-

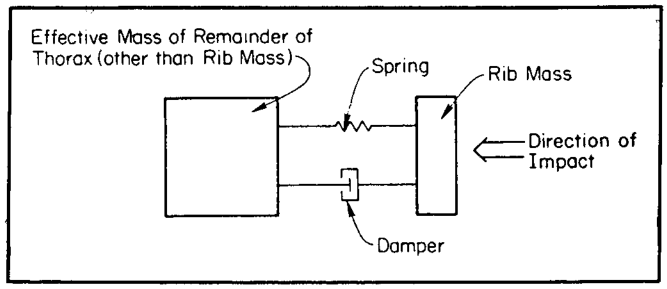


Figure 1. A lumped mass model of a lateral thoracic impact device

teristics of the thorax. The spring-damper combination shown in Figure 1 is one simple case. Generally, a more complicated system is needed to effectively model the thorax. This discrete element approach is essentially a physical realization of the mathematical model developed by Kroell and Lobdell[3]. General Motors[4], Ford[5], and the Motor Vehicle Manufacturers' Association (MVMA)[6] have utilized the discrete element method in the development of their respective lateral thoracic surrogates.

The second approach that has been taken in developing a thoracic side impact device is to design a distributed mass rib cage structure that attempts to duplicate the physical structure of the human thorax. This is the approach taken in the UMTRI/NHTSA Side Impact Dummy (SID) as shown in Figure 2. The thorax consists of steel ribs lined with damping

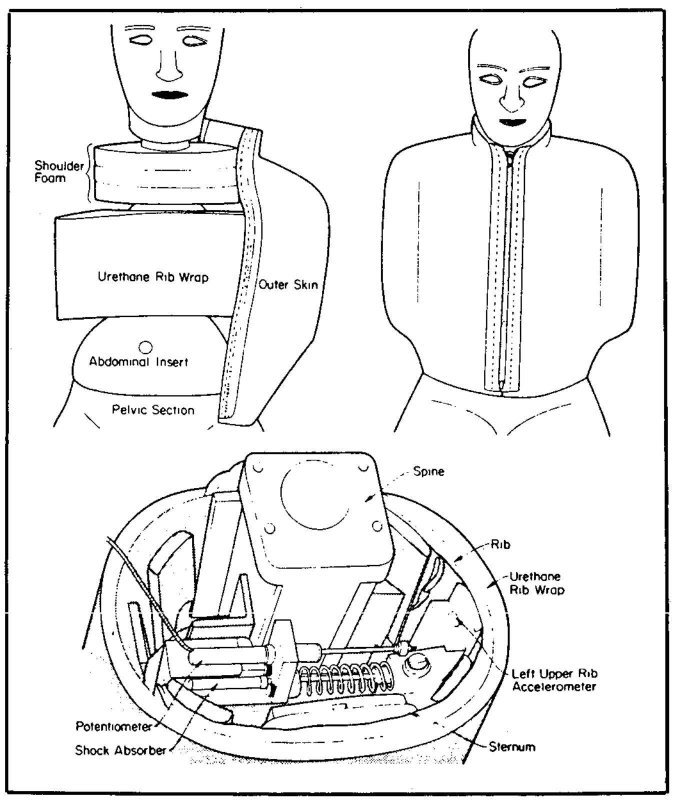


Figure 2. The side impact dummy (SID) thorax

material which create a continuous distribution of rib mass, stiffness, and damping characteristics. Additional spring and damping elements are present within the thorax to tune the device to give the desired response. The EUROSID thorax has also utilized this distributed parameter approach.

Biomechanical Database

There is only a limited amount of biomechanical data available pertaining to lateral thoracic impact. Two primary sources have been used in developing most of the thoracic surrogates mentioned previously, i.e., the Ford, GM, and MVMA devices. These are the APR (Association Peugeot-Renault) lateral cadaver drop tests[7] and a series of side impact sled tests performed at the University of Heidelberg in 1982[8]. The problem encountered is that these tests involved impacts with adult cadavers while the specimens of interest in this study are children ranging in age from 2 to 13 years. (The specific reason for being interested in this age range is given in the "Reconstruction Testing" section of this paper.) It was therefore necessary to manipulate the adult cadaver data to make it representative of children. A procedure was devised for this purpose which incorporated the normalization procedure developed by Mertz[9]. This procedure, which is described in Reference 10, produced proposed force-time and deflection-time lateral thoracic impact responses for fiftieth percentile 3, 6, 9, and 12 year old children. The data used in the analysis was that obtained from the APR cadaver drop tests. These proposed 3, 6, 9, and 12 year old force-time and deflection-time responses were used as the desired response behavior in developing the child thoracic surrogate devices.

Pedestrian Child Thorax Development

The initial approach taken in the development of the pedestrian child thorax surrogate device was to modify the thorax of the Humanoid six year old child dummy thorax as described in Reference 11. However, it was found that the modified Humanoid device had certain drawbacks which prevented it from being fully functional in the desired application. The main problem was that the device could not be impacted at speeds greater than about 27 kph (17 mph) without permanent deformation occurring to the ribs. This presented a problem since a goal of the NHTSA pedestrian research program is to study pedestrian accidents involving impact speeds up to 48 kph (30 mph). A second problem encountered with the modified Humanoid device is that the damping material bonded to the ribs began to develop cracks after about ten impacts at speeds between 16 kph (10 mph) and 24 kph (15 mph). These cracks propagated enough to make the rib response unpredictable after about 20 impacts at these same speeds. It was therefore decided to design a new child thorax surrogate which could withstand impacts up to 48 kph (30 mph) and have a longer useful life than the modified Humanoid child thorax.

In developing the new child thoracic surrogate device, the lumped parameter approach shown in Figure 1 was taken. The design is very similar to that of the thorax of the Side Impact Body Block (SIBB) developed by the Ford Motor Company. Since one goal of the current NHTSA study is to reconstruct pedestrian accidents involving various ages of children, thoracic surrogates for 3, 6, 9, and 12 year old children have been developed. The six year old thorax device is shown in Figure 3. As seen in Figure 3, the design includes a front plate which simulates rib mass on which are mounted guide rods which travel through a carrier frame on bearing surfaces made of Rulon. The carrier frame simulates the remaining thoracic mass as well as supporting the device on its hydraulic firing ram. Mounted on the front plate is a solid cylindrical piece of Dow Ethafoam 220, 2 inches thick which simulates the compliance of the skin and muscles surrounding the ribs. The compliance of the ribs and thoracic contents are simulated by a cylindrical section of Dow Ethafoam 600 approximately 3.5 inches thick. As seen in Figure 3, this thoracic foam piece has five holes drilled through it in the axial direction. The smallest of these holes is to allow room for the displacement potentiometer which measures relative motion between the front plate and carrier frame, i.e., relative displacement between the ribs and spine. The guide rods on the front plate pass through the other four smaller holes in the foam. The large hole in the center of the thoracic foam is to create the desired compliance behavior. Originally, it was thought that, to obtain the proper response from the device, some combination of different foams in series and/or in parallel would have to be used to create the desired response. This is because Dow Ethafoam was thought to have very little damping and the human thorax is very heavily damped. Thus, various combinations of Sorbothane, Ensolite, and different densities of Ethafoam were combined and tested. The large hole in the center of the thoracic Ethafoam compliance element allowed for the insertion of different materials in parallel with the Ethafoam. However, it was found that the best response was obtained by leaving this hole vacant.

The instrumentation used on the thoracic surrogate device can be seen in Figure 3. As previously mentioned, a linear potentiometer is used to measure relative displacement between the ribs and spine. An accelerometer mounted on the back of the front plate measures rib acceleration. Spinal acceleration is measured somewhat redundantly by two accelerometers

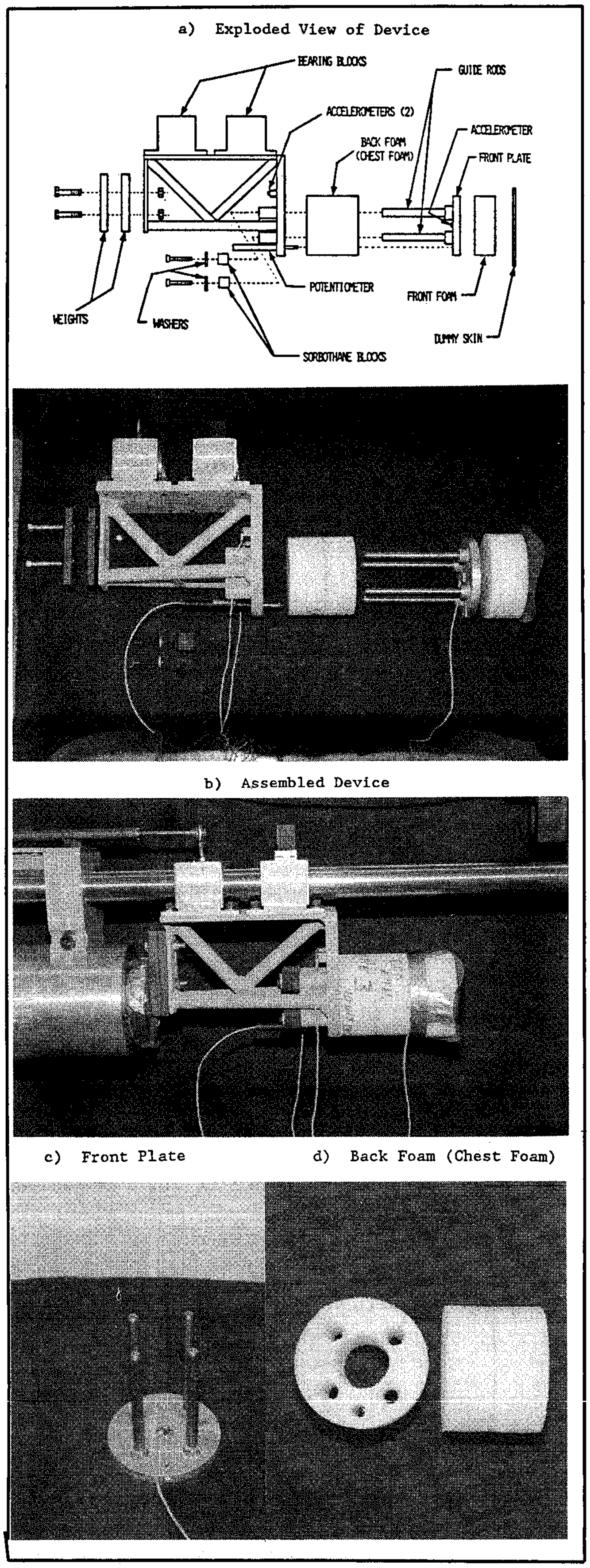


Figure 3. Six year old child lateral thoracic surrogate

mounted on the carrier frame. Two accelerometers were used in an attempt to observe if off-axis loading of the front plate would be transmitted to the carrier frame. It was found that when off-axis loading is applied to the front plate at up to 45 degree angles, there is no indication of the loading being non-axial from the spinal accelerometers, i.e., both accelerometers give essentially the same reading regardless of the loading conditions.

The mass of the carrier frame can be altered using the weights shown in Figure 3. Small weights are used to adjust the mass of the front plate. The 6, 9, and 12 year thoracic surrogates are all produced from the apparatus shown in Figure 3. The same compliance elements, i.e., 12.7 cm (5.0 in) diameter Ethafoam 220 front foam and 12.7 cm (5.0 in) diameter Ethafoam 600 chest foam, are used; only the masses are changed among the 6, 9, and 12 year old surrogates. The 3 year old device is similar to the device shown in Figure 3, but is built from a smaller carrier frame and front plate. The compliance elements used in the 3 year old device are of the same materials as those used in the 6, 9, and 12 year old devices, but are only 11.4 cm (4.5 in) in diameter. Figure 4 shows the 3 year old thoracic surrogate device along with the device used to simulate the thoraces of 6, 9, and 12 year old children.

The carrier frame ("fixed mass") and front plate ("moving mass") masses are shown in Table 1 along with the fixed and moving masses for the Ford, GM, and MVMA thoracic side impact devices. The devices developed for this study are called the "VRTC devices" in the table since they were developed at the Vehicle Research and Test Center (VRTC) in East Liberty, Ohio. Table 1 also shows the mass of each device as compared to the mass of the human subject being modelled. Also given is the relationship between the moving mass and the fixed mass for each device. As seen in Table 1, the total mass of the 3 year old

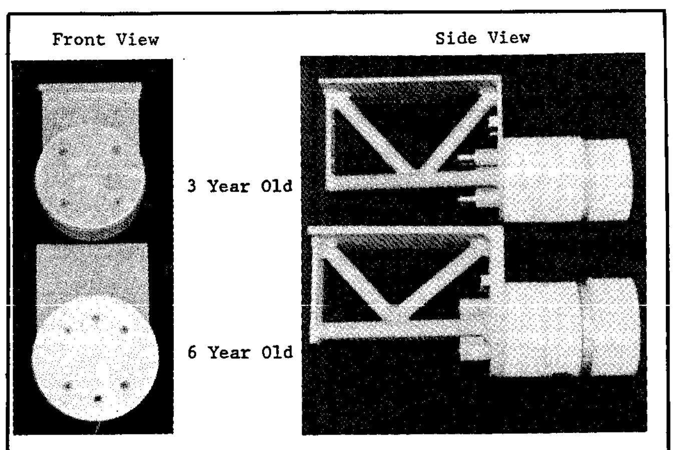


Figure 4. Size comparison of three and six year old thoracic surrogate devices

Table 1. Mass distribution of side impact thoracic surrogates.

Device	Age	Total Mass of Impactor (Kg)	Total Mass of Impactor (lb)	Total Mass of 50th Percentile Human (Kg)	Total Mass of 50th Percentile Human (lb)	Total Mass of Impactor Percentage of Total Mass of Human (%)	Moving Mass (Kg)	Moving Mass (lb)	Moving Mass as Percentage of Impactor Mass (%)
VRTC	3 Yrs	5.0	11.1	14.0	30.9	35.9	0.75	1.65	14.9
VRTC	6 Yrs	6.6	14.5	20.0	44.1	32.9	0.98	2.15	14.8
VRTC	9 Yrs	8.0	17.5	28.6	63.0	27.8	1.09	2.4	13.7
VRTC	12 Yrs	10.2	22.6	39.1	86.2	26.2	1.41	3.1	13.7
GM	Adult	24.0	53	76.3	168.3	31.5	3.63	8	15.1
Ford	Adult	17.2	38	76.3	168.3	22.6	2.72	6	15.8
MVMA	Adult	33.8	74.5	76.3	168.3	44.3	7.37	16.25	21.8

Table 2. Impact surface dimensions of side impact thoracic surrogates.

Device	12 Years	Width of Impact Surface (cm)	Width of Impact Surface (in)	Chest Depth of Human (cm)	Chest Depth of Human (in)	Ratio of Impact Surface Width to Human Chest Depth
VRTC	3 Years	11.4	4.5	11.4	4.5	1.0
VRTC	6 Years	12.7	5.0	12.8	5.0	1.0
VRTC	9 Years	12.7	5.0	14.7	5.8	0.86
VRTC	12 Years	12.7	5.0	15.8	6.2	0.81
GM	Adult	19.0	7.5	19.7	7.8	0.96
Ford	Adult	19.0	7.5	19.7	7.8	0.96
MVMA	Adult	19.6	7.7	19.7	7.8	0.99

thorax surrogate device is about 36% of the total body mass of the fiftieth percentile 3 year old child. This ratio of impactor mass to total body mass decreases to 33% for the 6 year old device. It decreases further to 28% for the 9 year old device, and finally to 26% for the 12 year old thoracic surrogate device. This occurrence was not necessarily intentional; it was essentially a result of altering the carrier frame mass to obtain the desired responses. There is no known information available which can be used to check the validity of this phenomenon of thoracic mass becoming a lesser portion of total body mass with age. However, it has been observed that the human thorax is somewhat rounded in cross section at birth and gradually assumes its fully developed, more elliptical shape[12]. Thus, the relative masses of the 3, 6, 9, and 12 year old thoracic surrogates are not inconsistent with this observation of the changing shape of the human thorax as it develops. However, as stated, no quantitative information is available to substantiate this point. The total thoracic mass of the Ford and MVMA impactors are, respectively, about 23% and 44% of the total mass of the adult male specimen being modelled. The values of thoracic mass as a percentage of total body mass for the child thorax devices are all within the range defined by those percentages observed in the Ford and MVMA devices. Furthermore, all values seem to center about the 31.5% figure observed in the GM impactor. Thus, while there is currently no way to validate the masses of the child thoracic surrogates, the values used are not unreasonable based on both intuition and the masses used in the adult thoracic surrogates. The moving mass for each device except the MVMA device is around 14-16% of the total device mass. The moving mass of the MVMA device is about 22% of its total mass. While the MVMA device has the higher mass percentage, all the values are relatively consistent.

Table 2 compares the impact surface width of each device to the chest depth of the human subject being modelled. Since these are all lateral thoracic impact devices, the goal is to have the width of the impact surface be the same as the chest depth of the human subject being simulated. The only remarkable information in this table is that the 9 and 12 year old front plate widths are the same as the front plate width for the 6 year old. This was done to avoid having to machine two additional front plates for the 9 and 12 year old devices. As previously stated, and as seen in Table 1, the mass of the front plate is altered among the 6, 9, and 12 year old devices, although dimensionally the front plate remains constant. Table 2 shows that even for the 12 year old, the surface is over 80% of its desired value and it is assumed that minimal error is caused by this slight dimensional error.

Results

Since the database used to develop the child thoracic surrogate devices is the APR cadaver drop test data, an attempt was made to emulate the APR test conditions when testing the devices. The APR tests of interest consisted of dropping cadavers onto assumed rigid surfaces from heights of 1.0 m (3.3 ft) and 2.0 m (6.6 ft). These drop heights produced impact velocities of approximately 16 kph (10 mph) and 22.4 kph (14 mph) respectively. Thus, the child thorax devices were impacted at these two speeds into a rigid surface (a wood block) as shown schematically in Figure 5. Prior to testing, each foam element undergoes a "break-in" period of ten impacts at about 24 kph (15 mph) into the rigid surface. Ethafoam is a closed-cell foam and thus requires these initial impacts to develop a consistent response. Following the break-in impacts, each foam sample was found to have a useful life of about

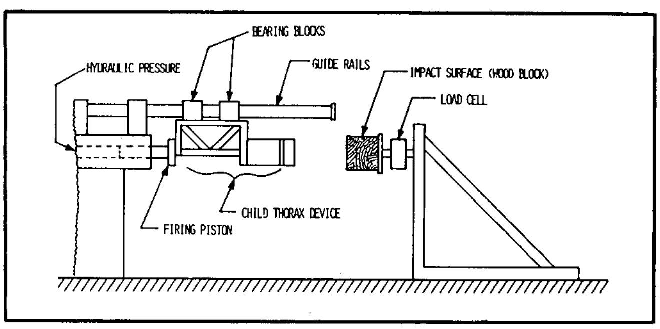
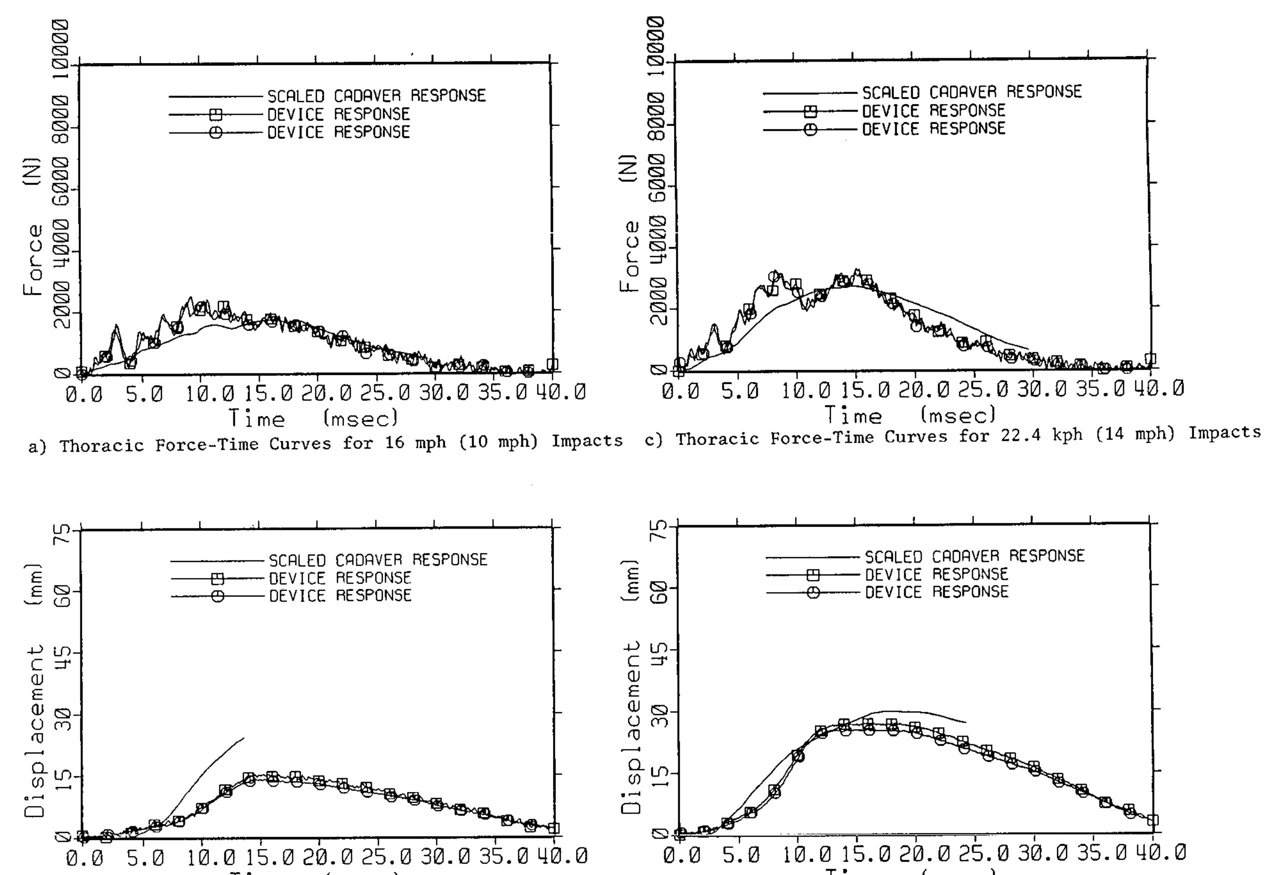


Figure 5. Schematic of test set-up to test child lateral thoracic impact device

50 impacts before its response degraded. In addition to this 50 impact repeatability, the foam behavior was also found to be quite reproducible, i.e., substituting new (broken-in) foam samples in the devices did not change the impact responses. Furthermore, it was found that blunt impacts up to 48 kph (30 mph) could be performed without damaging the thoracic surrogates.

Figure 6(a) shows the force-time responses and Figure 6(b) the deflection-time responses of two tests with the 3 year old device impacting the wood block at 16 kph (10 mph). Overlaid with these curves is the scaled and normalized cadaver data from the 1.0 m (3.3 ft) drop tests. Figure 6(c) and (d) show the results of two 22.4 kph (14 mph) impacts of the 3 year old device overlaid with the 2.0 m (6.6 ft) drop test scaled and normalized cadaver data. Figures 6(a) and (c) show reasonably good correlation between the force-time responses of the device and the cadaver data. Reasonable correlation between the deflection-time responses can be seen in Figures 6(b) and (d). Also, excellent repeatability of the device is seen in all cases. It should be noted that, as described in Reference 9, the cadaver data normalization procedure is not totally reliable when applied to the deflection data. Thus, while the scaled and normalized force-time cadaver data has some credibility, the deflection-time data is being used only as a rough guideline and is not necessarily claimed to be completely representative. Figures 7, 8, and 9 show the responses of the 6, 9, and 12 year old devices, each compared to its respective scaled and normalized cadaver data. In all cases, reasonably good correlation is seen between the devices and the cadaver data as well as good repeatability of each device. For the 9 and 12 year old devices impacting a 22.4 kph (14 mph), inertial spikes begin to appear in the force-time traces. While these spikes do not make the devices' force-time responses deviate intolerably from the cadaver data, they do seem to indicate that perhaps the chest foam should be slightly



a) Thoracic Force-Time Curves for 16 mph (10 mph) Impacts

c) Thoracic Force-Time Curves for 22.4 kph (14 mph) Impacts

b) Relative Displacement Between Impacted Ribs and Spine Versus Time for 16 mph (10 mph) Impacts

d) Relative Displacement Between Impacted Ribs and Spine Versus Time for 22.4 kph (14 mph) Impacts

Figure 6. Comparison of 3 year old child lateral thoracic surrogate device with scaled and normalized APR cadaver data

more heavily damped, especially for the 12 year old device. In general, however, the responses of all the devices compare favorably with the cadaver data.

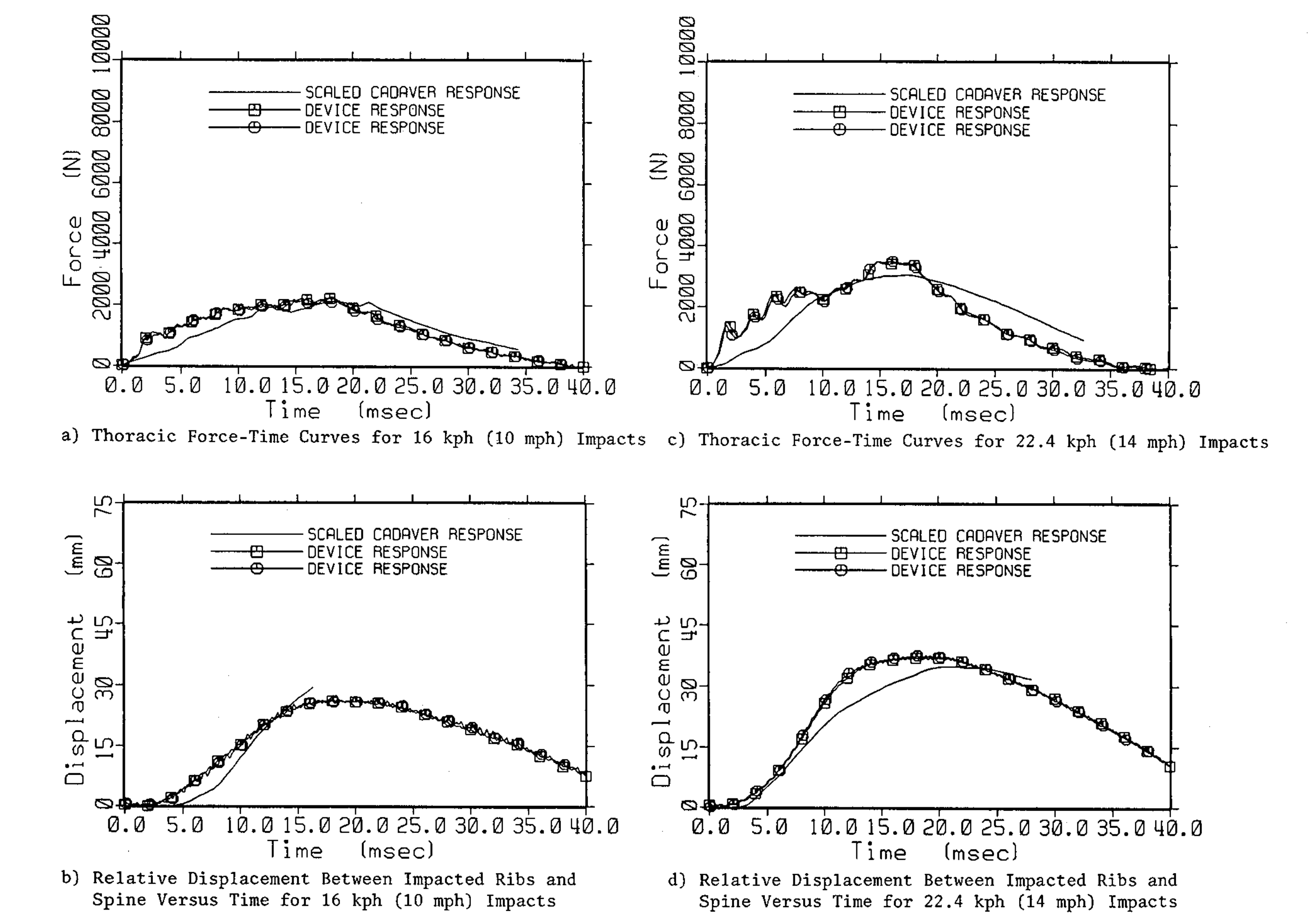
Overview of Remainder of Pedestrian Thoracic Injury Program

Reconstruction Testing

With the child surrogate devices developed, the next step in the NHTSA pedestrian thorax research program is to conduct accident reconstruction tests. The purpose of the accident reconstruction testing is to reproduce, in a laboratory setting, actual documented pedestrian accidents that involved thoracic injuries to children. This provides a method to verify that pedestrian accidents can be accurately simulated in the laboratory. Also, the performance of the child thoracic devices can be examined and a verifiable injury criterion can be determined. These accident reconstruction tests are currently being performed at VRTC. A brief description of the accident reconstruction methodology is presented here; a more thorough description will be given in a future publication which presents the results of these reconstruction tests.

The first step in the accident reconstruction testing was to obtain a collection of well documented pedestrian accident cases in which children received thoracic injuries from contact with the vehicle faces of passenger cars or light trucks. The sources used for these cases were the Pedestrian Injury Causation Study (PICS)[13] and the Pedestrian Accident Investigation Data Support (PAIDS)[14]. Fourteen cases were selected, 12 of which involved passenger cars impacting the child pedestrians; the vehicles in the remaining two cases were a pickup truck and a van. The children in the cases selected range in age from 2 to 13 years old. This is why it was desired to develop thoracic surrogate devices that could simulate the thoraces of children in this age range.

As stated previously, the goal of this reconstruction testing is to reproduce these actual pedestrian accidents in the laboratory. For each case, this is accom-



a) Thoracic Force-Time Curves for 16 kph (10 mph) Impacts

c) Thoracic Force-Time Curves for 22.4 kph (14 mph) Impacts

b) Relative Displacement Between Impacted Ribs and Spine Versus Time for 16 kph (10 mph) Impacts

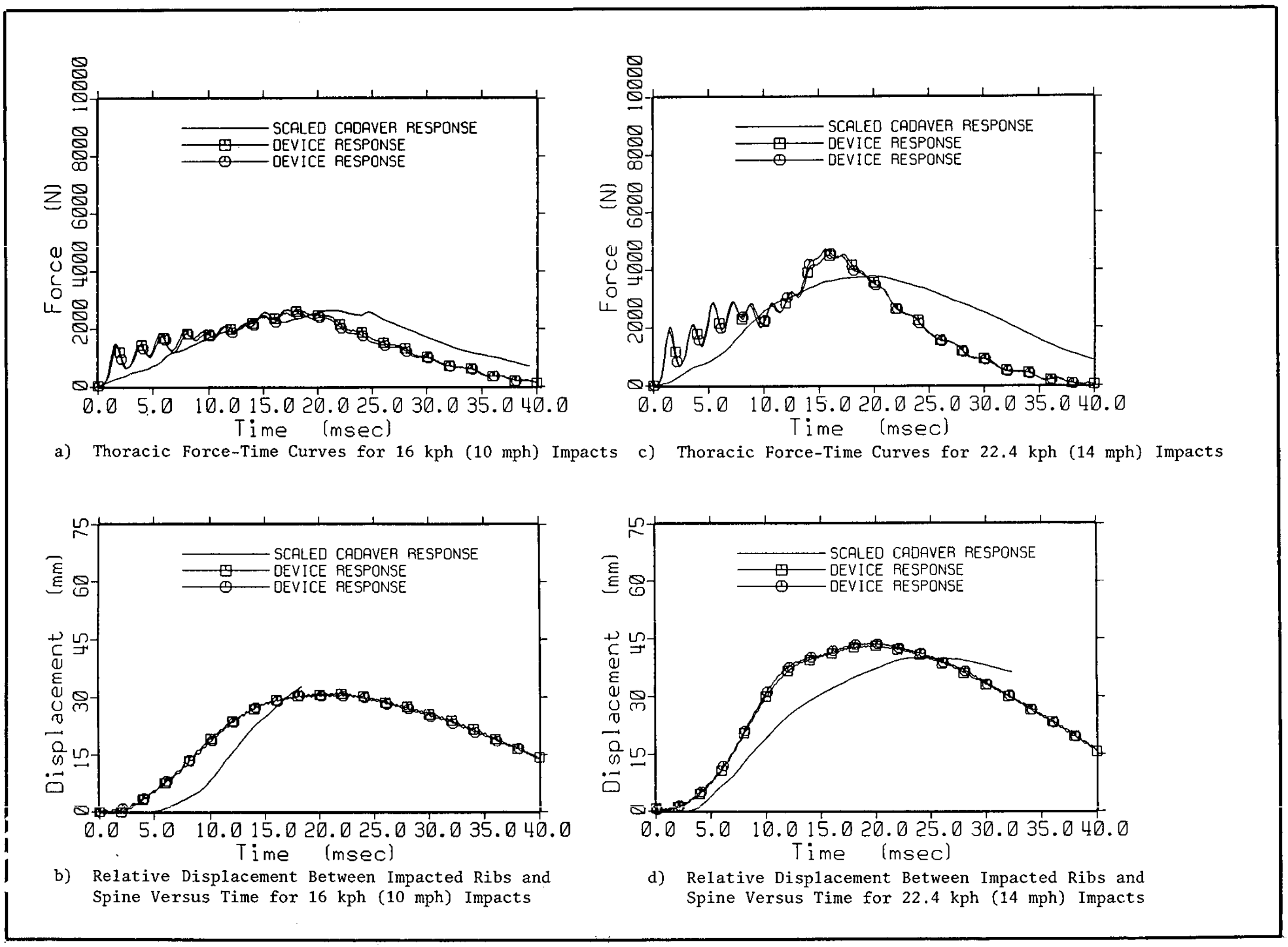
d) Relative Displacement Between Impacted Ribs and Spine Versus Time for 22.4 kph (14 mph) Impacts

Figure 7. Comparison of 6 year old child lateral thoracic surrogate device with scaled and normalized APR cadaver data

plished by taking a vehicle which is identical to that involved in the accident and impacting it with the child thoracic surrogate device that most closely represents the thorax of the child involved in the accident. The selection of the proper thoracic surrogate device is based on the height and weight of the child as opposed to the child's age. The impact location on the vehicle is determined from the damage sustained by the vehicle involved in the actual accident as well as the accident investigator's assessment of the pedestrian's kinematic behavior during the accident. The velocity with which the child's thorax impacted the vehicle face is determined by the accident investigator's calculation or estimate of initial impact speed and by MADYMO[15] computer simulations of the accident. If the reconstruction vehicle obtains a different damage pattern than that of the actual accident vehicle, the impact speed is reassessed and the reconstruction is performed again. When impacting the reconstruction vehicle with the thoracic surrogate device produces damage which is the same as (or reasonably close to) the damage to the actual accident vehicle, the accident is assumed to be reconstructed properly. After each accident case has been reconstructed as well as possible, the various displacement, velocity, and acceleration information obtained from the thoracic surrogate devices can then be compared with the thoracic injuries sustained by the children involved in the actual accidents. In this way, a verifiable injury criterion will have been determined for the thoracic surrogate devices, either by application of an existing criterion or by development of a new one.

Production Vehicle Testing

The overall goal of this pedestrian injury research program is to determine the causes of pedestrian injuries and develop practical countermeasures to reduce a struck pedestrian's propensity for injury. At the conclusion of the reconstruction tests it is hoped that a workable injury criterion will have been developed for the family of foam thoracic surrogate



a) Thoracic Force-Time Curves for 16 kph (10 mph) Impacts

c) Thoracic Force-Time Curves for 22.4 kph (14 mph) Impacts

b) Relative Displacement Between Impacted Ribs and Spine Versus Time for 16 kph (10 mph) Impacts

d) Relative Displacement Between Impacted Ribs and Spine Versus Time for 22.4 kph (14 mph) Impacts

Figure 8. Comparison of 9 year old child lateral thoracic surrogate device with scaled and normalized APR cadaver data

devices. The next step will be to examine a cross section of currently available production cars to determine their injury potential to a struck pedestrian. From these tests and the accident reconstruction tests, it can be determined what vehicle face features tend to cause pedestrian thoracic injuries and what features prevent those injuries.

Injury Reducing Design Concepts

The final phase of this research will involve taking all the information gathered in the accident reconstructions and the production testing and incorporating it into vehicle face designs which should reduce pedestrian thoracic injuries. Then, these designs will be tested in the laboratory to determine which designs offer the greatest injury reduction potential. Finally, attempts will be made to develop these injury reducing designs into practical designs that can be produced by the automotive industry with little or no interference in the price or the styling of the vehicles.

Summary and Conclusions

This paper has described a segment of the NHTSA Advanced Pedestrian Protection Program which is concerned with reducing thoracic injuries sustained by children as a result of contacts with vehicle front ends. The first phase of this program involves reconstructing actual pedestrian accidents using child thoracic surrogate devices. The main purpose of this paper has been to describe the development of these surrogate devices.

As mentioned in the paper, it was assumed that the typical pedestrian/vehicle collision is a lateral impact to the pedestrian. Therefore the thoracic surrogate devices developed for this program were designed to simulate lateral impact. Since one goal is to reconstruct pedestrian accidents involving children covering a wide age range, four devices were developed, representing fiftieth percentile 3, 6, 9, and 12 year old children. The biomechanical database used was the Association Puegeot-Renault (APR) lateral cadaver

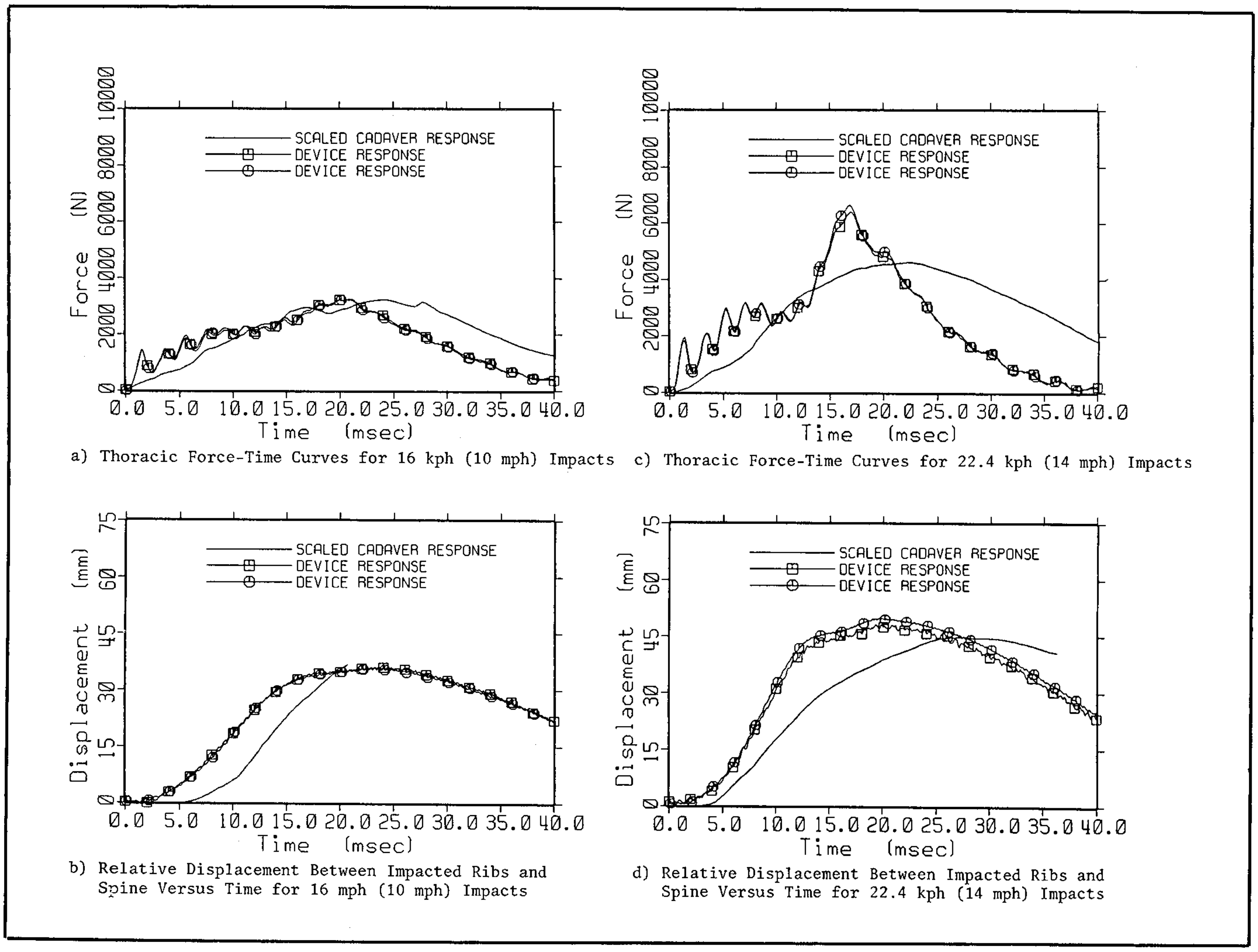


Figure 9. Comparison of 12 year old child lateral thoracic surrogate device with scaled and normalized APR cadaver data

impact data. This data was scaled and normalized as described in Reference 10 to produce thoracic response information representative of the four ages of children being modelled.

A lumped-mass approach was taken in the development of the thoracic surrogates as shown in Figures 1 and 3. A small front mass was used to represent the effective rib mass and a larger back mass was used to represent the spinal mass and the remainder of the effective thoracic mass. The compliance properties of the thorax were simulated in the devices with Dow Ethafoam. As was seen in Tables 1 and 2, the masses and dimensions of the devices are consistent with anthropometry data and with masses and dimensions of previously developed thoracic surrogate devices. Figures 6 through 9 showed that the devices responded reasonably well as compared to the scaled and normalized APR cadaver data. Also, the devices were found to maintain good repeatability up to about 50 impacts before the Ethafoam needed to be replaced. When the Ethafoam compliance elements were replaced, no appreciable response change was observed. It was also found that the thoracic surrogate devices could withstand impacts up to 48 kph (30 mph) without becoming damaged. It was concluded that, based on the scaled and normalized APR data, the child thoracic surrogate devices simulate child thorax impact response sufficiently well to be used in the accident reconstruction testing.

The discussion and conclusions in this paper represent the opinions of the author and not necessarily those of the NHTSA.

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Evaluation of Vehicle-Cyclist Impacts Through Dummy and Human Cadaver Tests

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Abstract

The TNO Road-Vehicles Research Institute is involved in a long-term research programme on pedestrian and cyclist safety. Until now 45 full-scale dummy and cadaver tests with simulated vehicle fronts and real passenger cars have been performed. Part of the programme is presented in this paper. Cyclist impacts with real passenger cars are compared with cyclist impacts with simulated vehicle fronts. Also comparisons are made between tests with cyclist-dummies and tests with cyclist-cadavers. Special attention is given to the influence of the initial bicycle speed. The results of these experimental simulations are compared with the results obtained from 3D mathematical model simulations conducted with the MADYMO CVS package. Recommendations are made with respect to future regulations in this field.

Introduction

In most European countries, unprotected road users account for a significant proportion of the road accident casualties. In almost every country, pedestrians are the most frequently involved[1]. In the Netherlands, however, where the bicycle is very popular, more cyclists than pedestrians are killed or severely injured. In the past years, international research was focused mainly on pedestrian safety. Based on this research, various recommendations for the front structure design of passenger cars were developed.

The TNO Road-Vehicles Research Institute has started a long-term research programme, concerned with cyclist safety in particular. The final objective of the programme is to reduce the number and severity of injuries sustained by all unprotected road users in a vehicle impact, by decreasing the aggressiveness of vehicle fronts. A research project has been formulated in which several research methods are integrated, including full-scale tests, body segment impactor tests, and mathematical model simulations.

In a previous study presented at the 10th ESV Conference[2], the influences of the vehicle impact speed and vehicle geometry on the cyclist and the pedestrian kinematics were analysed experimentally, as well as by mathematical model simulations. The influence of bicycle mass and initial bicycle speed was predicted by mathematical model simulations. The results of pedestrian and cyclist tests were compared. It was shown that the integration of cyclist and pedestrian injury prevention research is not only necessary but also feasible, considering the large number of similarities.

The results obtained in this previous study were based on tests with simple standard test dummies and simulated vehicle fronts. The cyclist in this previous test programme was initially not moving. Mathematical model simulations were conducted with the MADYMO 3D CVS package in order to evaluate and extrapolate the results of the experimental simulations. The study presented here extends this analysis to:

- moving-bicycle tests;
- tests with real passenger cars impacting a cyclist;
- tests with human cadavers seated on a bicycle.

The cadaver tests were performed by INRETS in France.

Test Method

Introduction

The general test set-up has been described earlier [2,3] and will be reviewed briefly in the next sections. Most of the conditions were selected on the basis of the analysis of real vehicle-cyclist and vehicle-pedestrian accidents. The test programme to be analysed here is summarized in Table 1.

Vehicle

The tests in the previous study were conducted with a moving barrier (without suspension) provided with a simulated vehicle front. This front consisted of three sections simulating the bumper, the grill, and the hood (see Figure 1). These sections were made of homogeneous polyurethane foam (density of 50 kg/m3 in the tests presented here). In the previous test programme the bumper height, hood-edge height, and bumper protrusion were varied. The simulated vehicle front tests conducted in the present study were performed with the standard vehicle geometry, i.e. a

Table 1. Variations of test parameters.

Test Number	Cylist	Bicycle speed (km/h)	Vehicle
8303/8306*	dummy	0	simulated front
8415/8416	dummy	0	Audi 100
8501/8502	dummy	15	simulated front
2R01/2R02/2R03/2R04/2R05	cadaver	0	simulated front

* reference test; see [2].

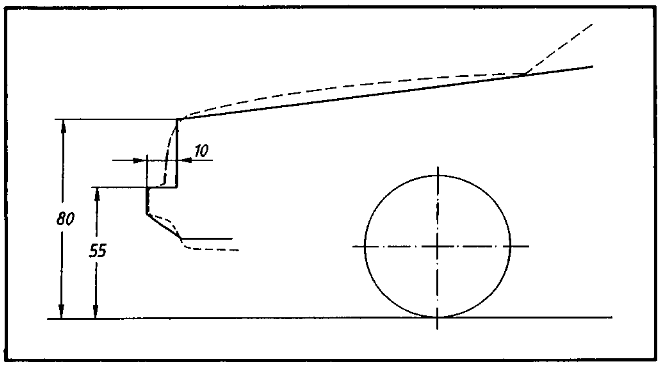

Figure 1. Dimensions (cm) of the simulated vehicle front (—) and the front of the real passenger car (---- = Audi 100)

bumper height of 55 cm, a hood-edge height of 80 cm and a bumper protrusion of 10 cm (see Figure 1).

The real passenger car tests were conducted with an Audi 100 and a Volvo 343. However, only the results of the Audi tests will be presented here. The suspension of the Audi was locked in a driving position in test number 8416 and in a braking position in test number 8415. The geometry of this vehicle in driving position, which is illustrated in Figure 1, closely resembles the geometry of the simulated vehicle front. The bumper height and hood-edge height are 6 cm lower in the braking position (48 cm and 74 cm respectively).

The vehicle impact speed was varied in the previous test programme. In the present tests the vehicle impact speed is 30 km/h. The vehicle starts to brake at initial vehicle/cyclist contact. The maximum vehicle deceleration due to braking is approximately 0.7 g's.

The vehicle impacts the cyclist laterally, 90 degrees between longitudinal axes of the bicycle and the vehicle.

Bicycle and initial position

The bicycles are standard Dutch men's bicycles, without chain, mudguards, lighting, etc. The remaining bicycle mass is 12.5 kg. The dummy sits in a prescribed standard position with its leg on the impact side stretched downwards (see Figure 2). In the tests with a stationary bicycle the dummy is supported by a cable fixed to the neck-bracket. The cable is disconnected by an electro-magnetic release system 10–40 ms before the impact. In the moving-bicycle tests the bicycle and dummy are placed inside a trolley, which is driven at 50% of the vehicle impact speed (see Figure 3). The bicycle wheels and saddle-pin and the cable fixed to the neck-bracket of the dummy are guided by the trolley before the impact. Then the trolley is stopped and the bicycle with the dummy is driven out of the trolley; the wheels and dummy support-cable are guided until the end of the trolley has been reached. The last 0.5 second before the impact, the bicycle and dummy are not supported and are driving freely.

The cadaver tests are performed with a stationary bicycle. The heights of the handlebar and saddle are adjusted to the cadaver dimensions; the leg on the impact side is stretched downwards and both arms are stretched (see also Figure 2). Prior to the impact the cadavers are supported by several cables fixed to a strap around the neck.

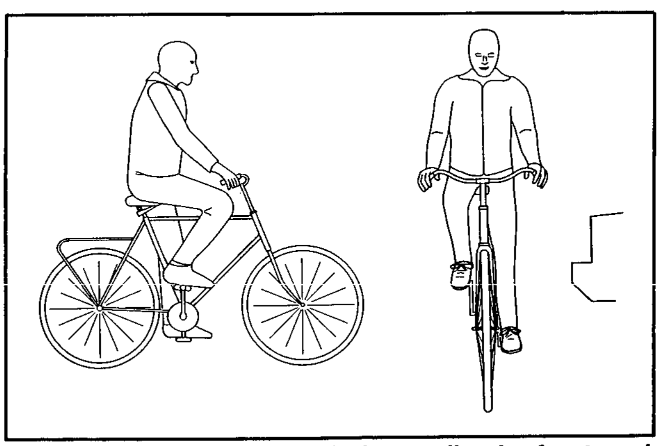

Figure 2. Initial position of the cyclist in front and side views

Dummy and human cadavers

The 50th percentile Standard Part 572 dummy (Hybrid II) was chosen for the dummy tests. It is a well-defined dummy and often used as a research dummy in spite of some disadvantages, such as limited lateral flexibility. Moreover, other, more realistic dummies were not available. To enable this dummy to be used as a cyclist dummy, a number of

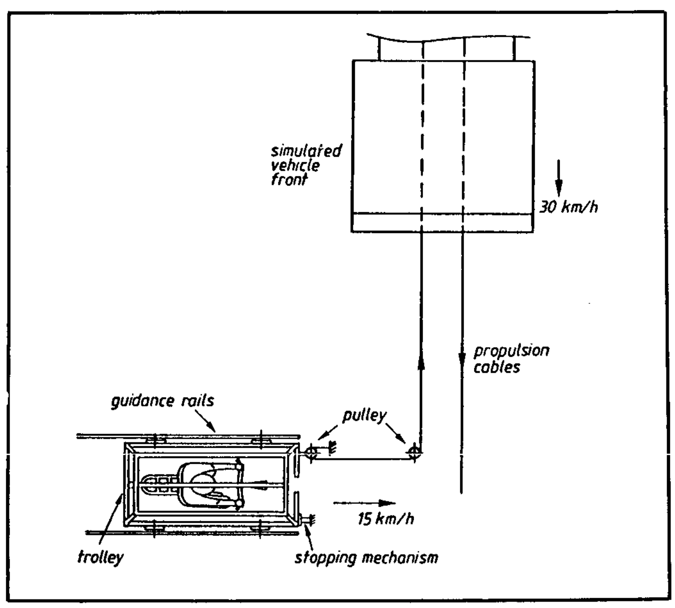

Figure 3. Test set-up of moving-bicycle test (top view)

Table 2. Human cadaver data.

Test Number	Cadaver Number	Sex	Age (yrs)	Body height (cm) Standing	On bicycle	Body weight (kg)
2R01	SA5	male	76	170	184	53
2R02	SA7	male	79	179	190	53
2R03	SA6	male	77	178	195	62
2R04	SA10	male	57	183	197	66
2R05	SA9	male	80	163	180	58

modifications are necessary. The range of motion in the hip joint is increased by cutting away some buttocks flesh. The knees, lower legs, and feet of the Standard Part 572 dummy are replaced by special pedestrian body parts, supplied by Humanoid Systems. These body parts are reinforced and easily replaceable. Furthermore, the ankles have a lateral flexion possibility and accelerometers can be mounted into the knees and feet.

The cadaver tests are conducted with fresh, unembalmed human cadavers. Table 2 summarizes some important cadaver data. The cadavers are all relatively old males. The body weight of the cadavers is lower than that of the dummy (75 kg) and the body height of the cadavers approximates the dummy's height (about 172 cm). However, the sitting height of the cadavers on the bicycle exceeds that of the dummy (approximately 176 cm). The lungs and vessels of the cadavers are pressurized during the tests.

Instrumentation and measurements

The dummy was instrumented with triaxial accelerometers in the centres of gravity of the head, thorax, and pelvis. Additionally, triaxial accelerometers were mounted into the knees and uniaxial accelerometers into the feet (lateral direction).

The cadavers were also instrumented with triaxial accelerometers in the head (mouth), thorax (sternum) and pelvis (sacrum). Furthermore, the inside of the impacted leg was equipped with three uniaxial accelerometers; two on the femur and tibia condyles and one on the ankle.

Two uniaxial accelerometers were mounted on the vehicle to measure the braking and impact deceleration.

Table 3. Results of cyclist tests.

Test Number	Head/Hood Impact Velocity (km/h)	Head/Hood Impact Location (cm) Longitudinal*	Lateral*	Remarks
8303	37	114	- 9	reference test
8306	39	112	-10	reference test
8415	32	125	-	Audi 100
8416	25	123	-	Audi 100
8501	37	112**	54**	moving bicycle
8502	35	106**	59**	moving bicycle
2R01	22	140	-	cadaver test
2R02	26	146	-	cadaver test
2R03	32	135	-	cadaver test
2R04	34	140	-	cadaver test
2R05	34	110	-	cadaver test

* see Figure 4 for definition.
** impact on upper arm.

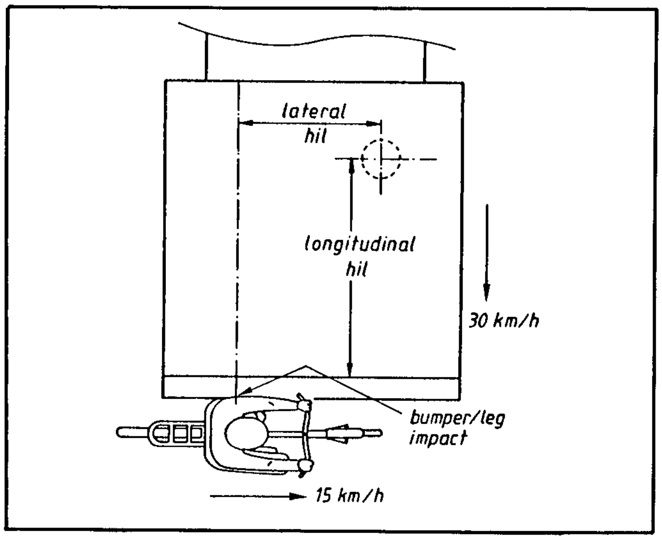

Figure 4. Definition of longitudinal and lateral head impact location

The throwing distance of the centre of gravity of the cyclist and bicycle were measured, as well as the braking distance of the vehicle and the permanent deformations of vehicle components. Eight high-speed film cameras were used to measure the motions of the cyclist and bicycle during the impact.

Test Results

Introduction

The results of the moving-bicycle tests, the real passenger car tests and the cadaver tests will be compared with those of the reference dummy tests (test number 8303/8306). The head impact on the hood will be analysed in detail, in particular the location and velocity of the head impact. Table 3 summarizes some important test results in this respect. The head acceleration versus time histories will also be presented in this section.

Moving-bicycle tests

The influence of the initial bicycle speed was analysed in an earlier study by means of mathematical model simulations[2]. A simulation was conducted with an initial velocity for the bicycle-cyclist combination of 15 km/h perpendicular to the vehicle velocity of 30 km/h. The results of these simulations indicated a shift of 67 cm to the lateral side of the vehicle in respect of the head impact location on the hood, whereas the longitudinal head impact location was hardly affected. Figure 5 illustrates the kinematics of the cyclist with and without initial velocity. One of the goals of the experimental moving-bicycle tests is to validate the mathematical model simulations.

The motion of the dummy in the stationary-bicycle tests (8303/8306) is an almost purely lateral one, without the dummy rotating around its longitudinal

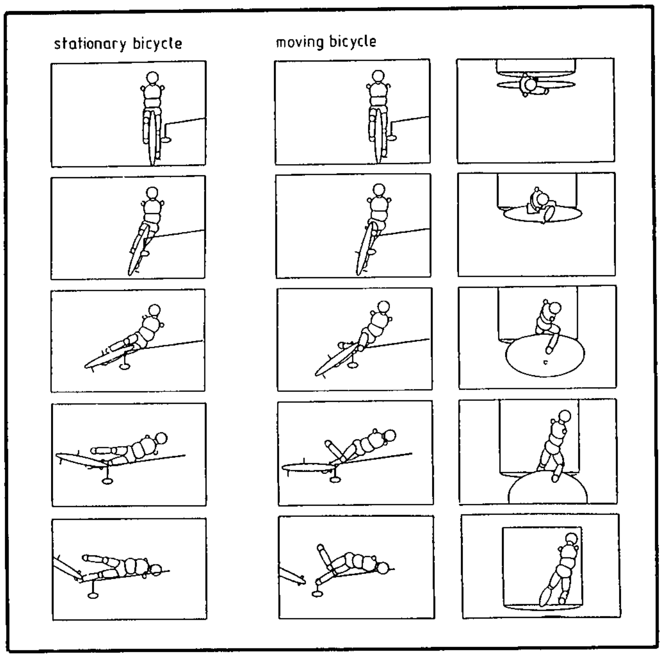

Figure 5. Kinematics of a stationary-bicycle and a moving-bicycle test obtained from mathematical model simulations with the MADYMO CVS package. Time frames of 50 ms are shown

body axis. In the moving-bicycle tests (8501/8502) the lateral motion of the dummy towards the hood is combined with the motion caused by the initial speed of the dummy; the dummy rotates around its longitudinal body axis with its face towards the vehicle. This is in agreement with the results obtained from the mathematical simulations (see Figure 5).

In the stationary-bicycle tests the lateral motion of the dummy resulted in an impact of the left shoulder and the left upper side of the head on the hood. In the moving-bicycle tests a more frontal impact of the head occurs on the upper arm, which is trapped between head and hood. The head impact locations are presented in Table 3. The longitudinal head impact locations in the moving-bicycle tests are similar or just slightly closer to the hood-edge than those in the stationary-bicycle tests. In lateral direction a shift of 63 to 69 cm could be observed. These results show that the predictions made by the mathematical model were correct.

The impact velocities of the head on the hood (or on the arm) are also included in Table 3; the head impact velocities in the stationary-bicycle and moving-bicycle tests are more or less identical.

Figure 6 shows the head acceleration versus time histories obtained from the stationary-bicycle and moving-bicycle tests. The head contact times are more or less identical. However, the shape of the curve differs owing to the impact on the (stiff) arm in the moving-bicycle tests instead of on the (soft foam)

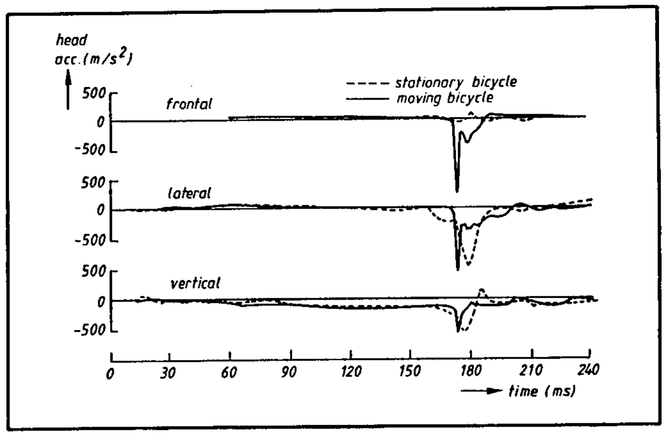

Figure 6. Head acceleration versus time histories of the cyclist-dummy obtained from a stationary-bicycle test and a moving-bicycle test, respectively

hood. Furthermore, in the moving-bicycle tests an acceleration in anterior-posterior direction of the head is observed, which is caused by the more frontal head impact. This acceleration was absent in the stationary-bicycle test, where the head impacted the hood laterally.

Real passenger car

The gross motion of the dummy laterally impacted by the simulated vehicle front (test numbers 8303 and 8306) and of the dummy laterally impacted by the Audi 100 (test number 8416) are similar. Figure 7 shows the trajectories of the head, chest and ankles of the dummy in both test types. However, the head impact location is approximately 10 cm further on the hood in the test with the real passenger car (see also Table 3). This could be caused by small geometrical (and stiffness) differences between the simulated vehicle front and the front of the real passenger car.

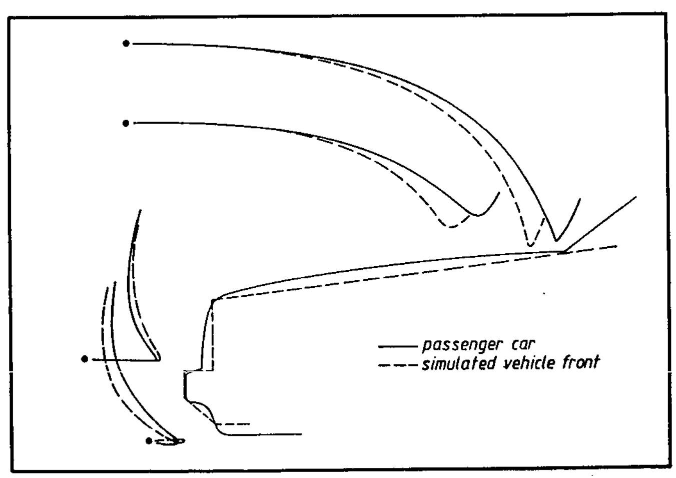

Figure 7. Trajectories of the cyclist-dummy's head, chest and ankle relative to the vehicle in a simulated vehicle front test and a real passenger car (Audi 100) test, respectively

(Mathematical model simulations showed that the difference in friction between simulated vehicle front and passenger car appears to be hardly of any influence on the head impact location).

The head impact location on the hood in the test with the Audi in braking position (test number 8415) is somewhat closer to the windshield, while the head impact velocity is higher than in the test with the Audi in driving position (see Table 3).

The head impact velocities in both real passenger car tests are lower than in the tests with the simulated vehicle front. Figure 8 shows the differences in head contact time and in slope of the acceleration curve at head contact. The tests with the real passenger car shows a sharp peak, probably caused by the inertia effects of the hood. The acceleration versus time histories are influenced by the local stiffness of the vehicle at the impact location. Nevertheless, the test with the simulated vehicle front and the tests with the real passenger car show surprisingly great similarity as to acceleration results.

The permanent deformations of the foam front and of the real passenger car, as well as the throwing distance of the dummy and bicycle appear to be quite similar.

Human cadaver tests

The overall kinematics as well as the acceleration versus time histories of the cadavers seem to reproduce surprisingly well. The motion of the cadaver is a purely lateral one without rotation around the longitudinal body axis, which is in agreement with the dummy reference tests. Figure 9 illustrates the kinematics of a cadaver laterally impacted by a simulated vehicle front. The upper body of the cadaver remains longer in an upright position than the upper body of the dummy; the lower body of the cadaver is more wrapped around the front. Furthermore, the lower body of the cadaver is thrown more upwards by the impact, and between approximately 125 and 225 ms the whole body flies over the hood without contacting the vehicle. Figure 10 shows the head trajectories of the cadavers and the dummy. The cadaver head impacts the hood 20 to 35 cm further towards the rear-end of the vehicle than the dummy head, while the sitting height of the cadavers exceeds that of the dummy by only 8 to 21 cm (except for the relatively stout cadaver SA9 in test number 2R05). The head impact velocity of the cadavers is lower than that of the dummy. This difference in kinematical behaviour between dummy and cadavers is also illustrated by the head acceleration versus time histories shown in Figure 11. In the cadaver tests the head impact on the hood occurs much later in time.

Injuries sustained by the cadavers are summarized in Table 4. In addition to these injuries all cadavers

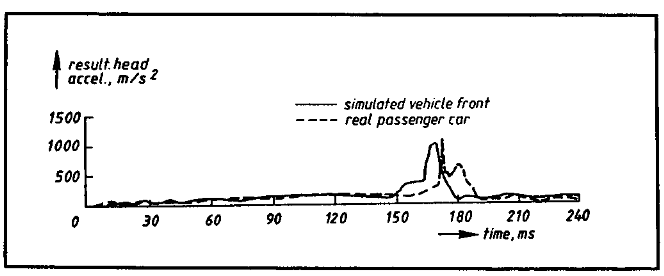

Figure 8. Head acceleration versus time histories of the cyclist-dummy obtained from a test with a simulated vehicle front and a real passenger car (Audi 100) test, respectively

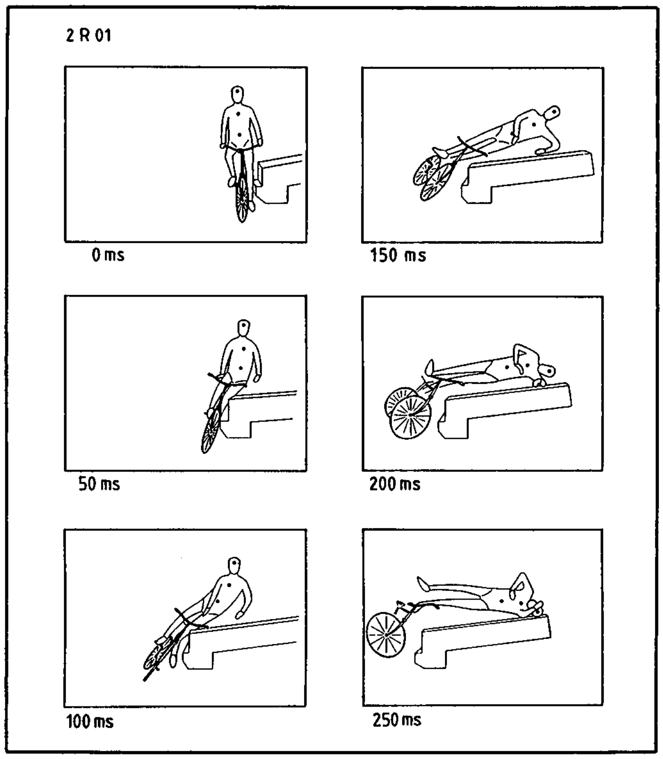

Figure 9. Kinematics of a cyclist-cadaver in a 30 km/h impact with a simulated vehicle front

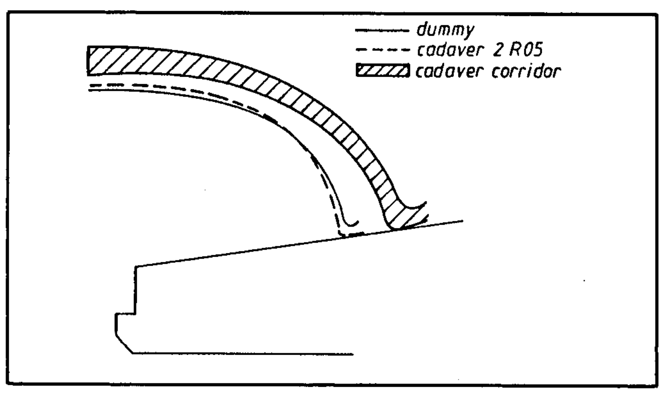

Figure 10. Head trajectories of the cyclist relative to the simulated vehicle front obtained from dummy tests and human cadaver tests, respectively

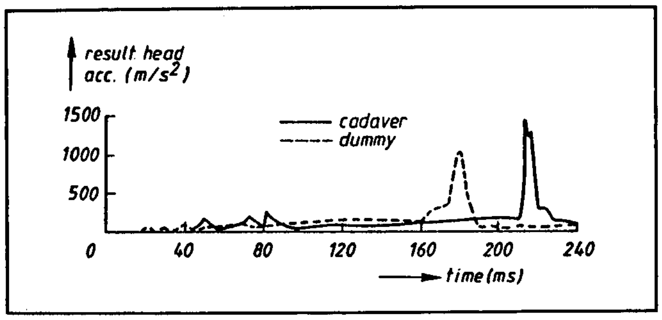

Figure 11. Head acceleration versus time histories of the cyclist obtained from a dummy test and a human cadaver test, respectively

Table 4. Injuries sustained by cadavers in cyclist tests.

Body region	Test number 2R01	2R02	2R03	2R04	2R05
Head	-	-	-	-	-
Neck	fracture-luxation (AIS 2)	-	-	fracture C5/C6 (AIS 3)	fracture C3/C4 (AIS 3)
Thorax	-	1 rib fracture (AIS 1)	-	3 ribs fractured (AIS 3)	-
Abdomen	-	-	-	-	-
Pelvis	-	-	-	-	-
Extremities	fractures lower leg and ankle (AIS 3)	fracture lower leg (AIS 3)	fractures lower leg (AIS 3)	fractures lower leg (AIS 3)	fractures lower leg (AIS 2)
External *	(AIS 1)	(AIS 1)	(AIS 1)	(AIS 1)	(AIS 1)
MAIS	3	3	3	3	3
ISS	14	11	10	28	14

* See text for description of injuries.

showed external contusions, haematomas, lacerations or abrasions on several body parts. None of the cadavers sustained a skull fracture or brain injury (the HIC values are all below 1000). Two cadavers showed one or more rib fractures, while cervical spine fractures were found in three cadavers. None of the cadavers showed a pelvis injury. Of all five cadavers the lower leg on the impact side was broken. No injuries were found in the knee ligaments.

Discussion and Conclusions

Until now 45 full-scale dummy and cadaver tests have been performed within the framework of the TNO's research project on pedestrian and cyclist safety. Almost all tests were conducted with simulated vehicle fronts made of poly-urethane foam in order to change stiffness and geometry in an inexpensive and controllable manner, and to improve the repeatability of the tests.

In previous studies[2,3] the influence of the vehicle speed and geometry on the cyclist and pedestrian kinematics was analysed. Comparison of the simulated vehicle front tests with real passenger car tests in the present study showed that dummy kinematics and accelerations are quite similar. So, the results obtained in the tests with the simulated vehicle front appear to be realistic in this respect.

Previous mathematical model simulations with the MADYMO CVS package showed that an initial bicycle speed (perpendicular to the vehicle speed) appears to have only a slight influence on the head impact location in longitudinal vehicle direction; the shift in lateral vehicle direction appeared to be equal to the initial bicycle speed multiplied by the head/hood impact time[2]. Moreover, the head impact velocity was hardly affected by the bicycle speed. The present experimental simulations show that the predictions made by the mathematical model are correct. These previous simulations with MADYMO also showed that the peak bumper load is hardly influenced by the initial bicycle speed. Furthermore, similar peak bumper loads were obtained in cyclist and pedestrian impacts. The bicycle mass probably compensates for the difference in mass distribution relative to the vehicle and for the difference in leg position between cyclist and pedestrian.

The dummy tests in our research programme are conducted with a (modified) Hybrid II dummy. It appears from the present study that human cadavers are laterally more flexible than this dummy; the lower body of the cadaver is wrapped more around the vehicle front. Furthermore, the upper body of the cadaver remains longer in an upright position, while the dummy rotates more or less as a single element around the hood-edge. This results in considerably different head impact locations for the cadavers and the dummy, even when the results are compensated for differences in sitting height. It appears that the kinematics of relatively heavily built cadavers are more 'dummy-like', probably because they are less flexible than the long and thin cadavers. The human cadavers did not sustain brain injuries or skull fractures, while rib and cervical spine fractures, as well as lower leg fractures were observed frequently. The maximum AIS was 3 for all cadavers.

To enable a more detailed analysis of the differences between dummies and cadavers, the research programme has meanwhile been extended with mathematical cadaver-model simulations. The dimensions and masses, as well as the mass-distribution and moments of inertia of the model elements are based on the real cadavers as well as on anthropometric studies. The legs are provided with a breakage mechanism and the hip joints are simulated by a flexible connection rather than by a ball-and-socket joint. Moreover, the characteristics of the joints (particularly in the spine) have been changed with respect to the dummy-model. The results of this analysis will be presented in a future paper. It can already be concluded that this type of simulation is more complicated than the previous dummy simulations and that a realistic description of the vehicle-leg contact is very important. Further research work will be concentrating on this aspect.

Real-life pedestrian and cyclist accidents show a large variability, in particular with respect to vehicle dimensions and anthropometry of the victim. This variability usually results in a wide variation in the impact locations of the vehicle front. Earlier TNO experiments, which served to compare pedestrian and cyclist accidents, indicated that the major difference is in the head impact location: a cyclist could be treated as a large pedestrian[2]. It seems impossible to determine the injury reduction potential of a vehicle front by one single standard test. It would take at least a series of realistic pedestrian dummies with different anthropometry to provide sufficient data for the evaluation of vehicle fronts with respect to both pedestrian and cyclist safety. However, such a 'dummy-family' is not yet available and will not be easy to design. As an alternative, a test method could be developed in which the vehicle front is evaluated by body segment impactor tests. The initial conditions of such tests could be determined by mathematical models. Results of mathematical simulations obtained in our research programme have so far indicated that these simulations can indeed provide quite reliable predictions for this purpose.

Acknowledgements

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A Bumper for Both Pedestrian and Vehicle Body Protection; A Contradiction in Terms or a Soluble Conflict?

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Abstract

In normal operation, passenger car bumpers are meant to protect the vehicle body from damages during minor collisions. In most cases, when colliding with a pedestrian, the bumper is the first vehicle component to be hit causing injuries to the legs. The induced injury severity depends on the bumper configuration and stiffness. As a result of biomechanical investigations a maximum force level has been recommended which should not be exceeded in order to minimize the injury severity.

This recommendation conflicts with the demands on the bumper as a body part: The bumper system must be able to transmit high forces to meet current regulations, such as Part 581 for example, on the one hand, but on the other it has to be stiff enough to sustain normal operating conditions without failing throughout the vehicle's service life.

The present paper examines the possibilities for an integrated bumper system in consideration of biomechanical findings, current regulations, and manufacturers' demands using various series vehicles and test components.

Introduction

The road users participating in today's passenger car traffic differ from each other in several aspects. This difference is most pronounced between passenger cars and pedestrians. The most essential distinctive features are: road safety drill (drivers, pedestrians), physical constitution (of human beings), stiffness (of the vehicle structure), mass, speed. Experience has shown that it is not always possible to ensure problem-free coexistence between these road users. During collisions, the different physical, mechanical and anthropometric conditions result in accident aftereffects and, as in the present case, in personal injuries which should be minimized.

The first step is to define the category of road users for whose sake measures have to be taken of which measurable benefit may be expected. These investigations are based on accident statistics and on results of detailed accident analysis. The 1985 accident statistics of the FRG present the following figures/1/: 53.5% of all persons involved in a traffic accident were passenger car occupants, while 32.6% were motorcycle and bicycle riders. Pedestrians account for 10.5% of all road users injured.

As far as the injury severity is concerned, however, it is found that 21.3% of all persons killed and 15.6% of the severely injured were pedestrians. Based on these figures it seems to be useful to deal more intensely with the problem of pedestrian accidents.

There are three basic approaches to the attenuation of the accident severity:

- traffic safety education for both drivers and pedestrians to sharpen the danger awareness and to reduce the willingness to run risks;
- taking the edge off accident-prone road sections by changing the traffic facilities for example;
- modifications to the vehicle to minimize the injury severity of the hit pedestrian.

The first two approaches are beyond the automotive manufacturer's direct influence. Therefore, the following chapters are focussed on the technical improvements to the vehicle only.

Prior to any improvement it is necessary to obtain thorough information on actual accident conditions and on the parameters influencing the hit pedestrian's kinematics and injury severity. These questions have been investigated during recent years and the basic correlations during this type of collision could be determined/2/,/3/,/4/,/5/,/6/,/7/. The following parameters have been found to essentially influence the injury severity during a primary impact:

- age
- weight
- height
- constitution of the pedestrian
- speed (level and direction)
- impact height
- contour
- stiffness
- bumper height
- hood edge height
  of the impacting vehicle
- hood length
- driving speed
  (level and direction)

In an earlier publication/2/, the advantages of a long shaped vehicle front end have been presented. The present investigation takes into consideration the same geometrical conditions and analyses various bumper concepts and stiffnesses. It is aimed at harmonizing the diverging demands made on a bumper system.

Demands on Bumper Systems

When conceiving a vehicle front end the automotive manufacturer must make it comply with a multitude of requirements. The most important features to be observed are:

- safety regulations
- protection during accidents
- operational and functional reliability
- customer demands
- styling and aerodynamics
- production and repair costs
- component weight

The following specifies the demands on bumper systems assuming a protective function during vehicle/pedestrian collisions.

Safety Regulations

According to the country for which a car is destined the bumpers intended to protect the vehicle body from minor damages must meet the following regulations:

- 49 Code of Federal Regulations, Part 581
- ECE-42
- CMVSS 215

Protection During Accidents

When colliding with a pedestrian the bumper transmits forces to the pedestrian's lower extremities which, depending on the respective anthropometric conditions, may result in more or less severe injuries. From accident studies and biomechanical investigations it is known which vehicle front geometries favour the kinematics and which stiffness should not be exceeded/8/,/9/,/10/,/11/.

The vehicle front end design depends on which group of persons should be protected (adults, children). As it is impossible to set up objective priorities a compromise must be found covering both groups.

As biomechanical research has confirmed the loads applied to the lower extremities should not exceed 4,000 N. Therefore, during bumper stiffness layout, it must be confirmed by means of a defined test procedure that this measured value is not exceeded.

Operational and Functional Reliability

The bumper must be functionally reliable under all service conditions a vehicle undergoes during normal operation:

- temperature of e.g. -20°C to +50°C
- high atmospheric humidity
- ultra violet effects
- vertical vibration stresses due to road surface unevenness

These conditions are expected to last throughout the vehicle's entire service life.

These conditions stipulate the position of and the performance criteria to be fulfilled by the bumpers. At present, the most severe demands in view of body protection must be met in 8 km/h (5 mph) pendulum and collision tests where no damage to the body and lamps is allowed (CMVSS 215). Porsche's actual bumpers do fulfill these requirements.

Customer Demands

The customer expects a bumper system to reliably protect the vehicle body from damage during minor parking collisions and to display a high surface quality which will not change throughout the vehicle's service life. In general, there is no customer demand for an integrated pedestrian protection. Thus, it is the designer's task to provide for a functionally reliable and optically superior system which takes into account the customer's demands.

Styling and Aerodynamics

The front bumper is an important feature in the design of a vehicle front end. Various highly distinctive bumper configurations are obtained by varying the bumper arrangement, geometry, colouring and material *(Figure 1)*.

Production and Repair Costs

The production costs should be low to allow a favourable vehicle sticker price. However, the vehicle must be designed in a way so as to be repairable at relatively little expense in case the bumper system is overloaded during an accident.

Component Weight

In the interest of a reduced overall vehicle weight and a low mass-related fuel consumption the bumper weight should be minimized.

Method

Actual Production Vehicles

According to the weight attributed to the individual demands different bumper systems have been developed. One of the main targets in automotive bumper development, however, is to provide the overall system with a high energy-absorption capacity.

Among today's energy-absorbing bumper elements number:

- hydraulic damping elements
- regenerating plastic structures
- PU-foams
- thermoplastic synthetics

Figure 2. The operativeness and usability of an energy-absorbing structure essentially depends on the impact force to be absorbed.

Energy-absorbing components alone are not well suited for direct force introduction. It is the force transmission between the point of impact and the energy-absorbing component which determines the efficiency of the bumper system.

Table 1 shows various bumper configurations and describes the respective outer panels, type of force-transmission and energy-absorbing structures used.

In the past, the vehicles had either a front end designed mainly for efficient pedestrian protection(/12/,/13/) or were fitted with a bumper which while meeting numerous other demands offered improved pedestrian protection, too(/14/,/15/).

An extensive theoretical and experimental analysis of the above-mentioned demands led to further investigations aimed at identifying design solutions for an integrated bumper system taking into account biomechanical findings, current regulations, and manufacturer demands.

Tests

Requirements

As accident investigations have shown, pedestrians suffer severe injuries when colliding with a vehicle at collision speeds of 5 m/sec and more. About 50% of all pedestrian accidents occur at collision speeds of more than 5 m/s.

For the layout of a "pedestrian-compatible" bumper the upper speed limit has been fixed at 40 km/h. Beyond this speed, the frequency of severe or fatal injuries increases. Pedestrians mostly die of head injuries. Lesions to the lower extremities are rarely rated more than AIS 3. Very often, however, the healing process takes long and severe permanent or late impairment may occur which is not taken into account in the AIS rating.

Several test methods have been defined to evaluate the protective effects of a bumper system during a vehicle/pedestrian collision.

Test Procedure

For the purpose of the present investigation, the tibia pendulum proposed in "Federal Register / Vol.

Table 1. Examples for different bumper configurations.

Outer skin	Force transmission	Energy-absorbing structure
PUR-RIM	cross member	hydraulic shock absorbers
Modified polypropylene	cross member	mainly plastic-integrated
modified polypropylene and EA-foam	cross member	plastics and EA-foam
PUR-RIM	through connenction between PUR-RIM/EA-foam	EA-foam
rubber	cross member	regenerating plastic cells (e.g. polytheylene honeycombs)

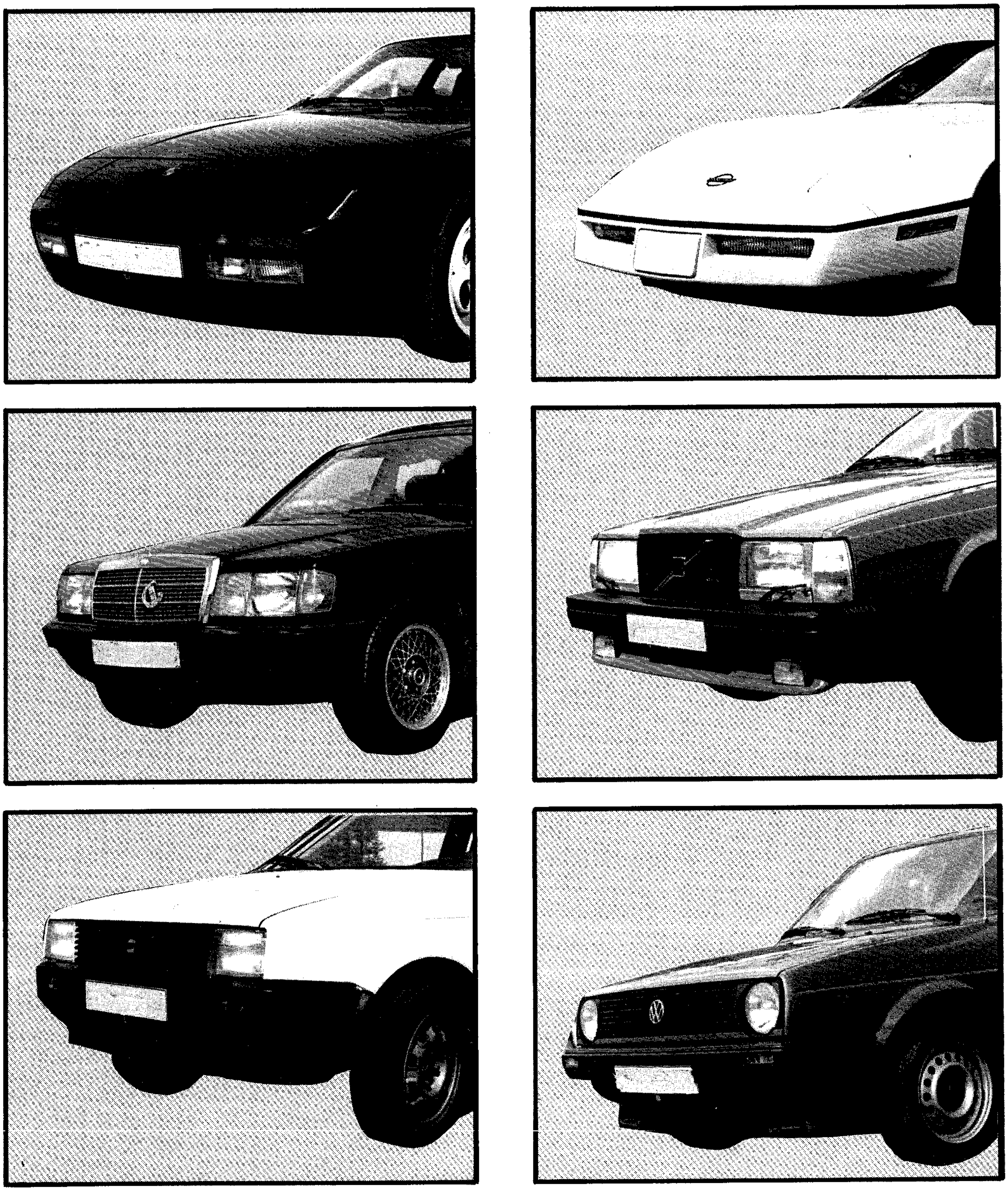

Figure 1. Examples for different bumper designs in today's production cars

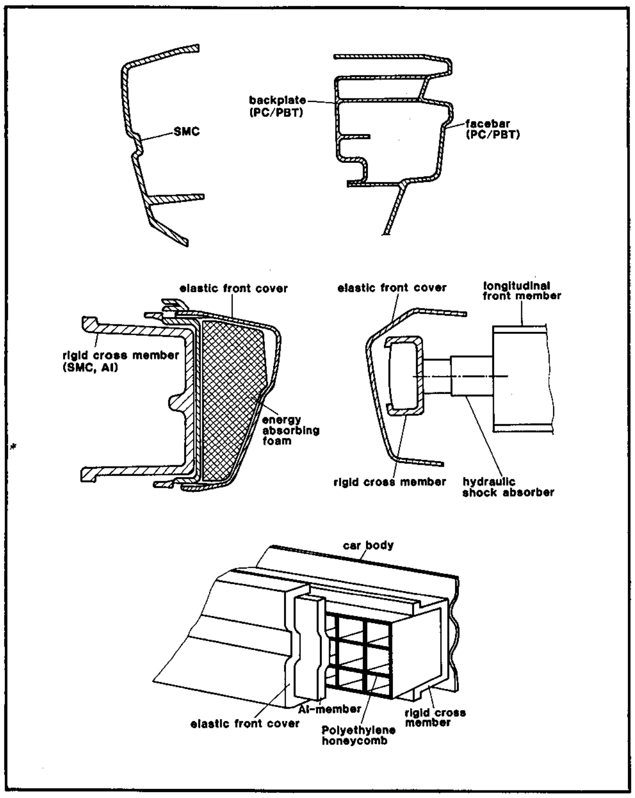

Figure 2. Examples for different bumper design concepts

46, No. 14 / 49 CFR Part 571; Proposed Rule" was chosen (see *Figure 3*). The wooden impactor was connected to the 4.36 kg pendulum above an aluminum plate.

For bumper testing with the tibia pendulum, speeds of 5, 7 and 8.6 m/s respectively were chosen. The acceleration was measured at the impactor center whereas the deformation was determined at the point of impact by means of a potentiometer.

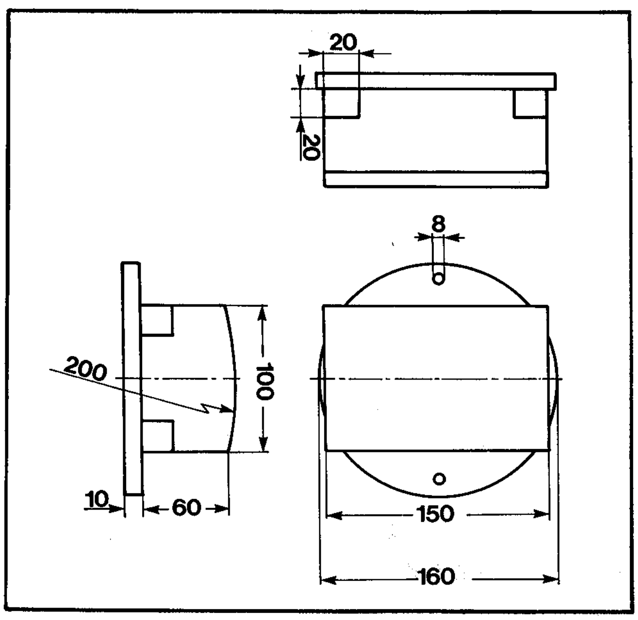

Figure 3. Dimensions of impactor

Systems Tested

First, the bumper characteristics of actual production vehicles were determined, using the following bumper variants:

Table 2. Bumper variants examined.

Bumper variant	"No damage" Impact speed
Vehicle A (cross member, shock absorbers)	8 km/h
Vehicle B (EA foam)	8 km/h
Vehicle C (EA foam, plastic cross member)	4 km/h
Vehicle D (foam prototype without skin, dimensions: W = 750 mm, T = 60 mm, H = 135 mm)	8 km/h

The bumpers meet FMVSS, Part 581, at the given impact speeds. Bumper variant D was a prototype made of energy-absorbing PU-foam (PU-foam No. 1 according to *Table 3*) without skin.

In addition, a PUR-RIM front-end panel without cross member has been examined.

For the tibia pendulum test the bumpers and front-end panels had been duly fastened to the vehicle.

For testing, the foams were made into square bumper prototypes without elastic outer skin.

The test parameters are listed in *Table 4*. *Table 5* gives a survey of the tibia pendulum tests performed with the bumpers and front-end panels. The test matrix for the foam tests is given in *Table 6* and *Table 7*.

Results

Evaluation of Measurements

Contacts are qualified as either "hard" or "soft". Hard contacts produce high maximum forces and relatively low 3-ms-forces. The softer the contact the smaller the difference between the 3-ms and the maximum contact forces. With soft contacts the maximum contact force is lower than with hard ones.

Table 3. Material properties of the energy-absorbing PU-foams.

PU-foam		1	2	3	4
density	kg/m**3	90-95	85	85	85-100
crush resist. 50% deform.	kPa	500	*	180	165
tensile strength	kPa	580	500	290	350
breaking elongation	%	30	25	25	30

* value not available

Table 4. Test parameters and filtering.

Mass of Impactor	:	4.366 kg
Impact Speeds	:	5, 7 and 8.3 m/sec
Impact Energy	:	55, 107 and 150 Nm
Filter Frequency	:	1000 Hz

Table 5. Vehicle test matrix.

Impact Energy	55 Nm	107 Nm	150 Nm
Car Type A	3*	3	-
Car Type B	3	3	-
Car Type C	8	7	-
PUR-RIM Skin	12	13	-
EA Foam Bumper Prototype	-	4	2

*Number of tests performed

Figure 4 shows the results of a tibia pendulum test performed with a bumper integrated in the outer body panels and an energy-absorbing PU-foam system.

The impact tests carried out with the various bumpers and EA-foams have been evaluated for maximum force levels and deformation. The force applying during the 3-ms period has been neglected as it is the maximum force which accounts for the pedestrian's injury severity. Due to the vibrations of the pendulum it has been impossible to evaluate the pendulum rebound velocities from deceleration/time histories.

Table 6. EA-foam test matrix.

Foam Nr.		Thickness (mm)	Height (mm)	Impact Energy 55 Nm	107 Nm	150 Nm
1	FC7	93	135	-	4*	2
1		60	135	-	2	-
1		45	135	-	2	-
1		30	135	-	2	-
1		15	135	-	2	-
2	FC1	45	135	2	2	-
2		30	135	2	2	-
2		15	135	-	2	-
2		60	90	3	3	-
2		45	90	2	3	-
2		30	90	3	2	-
3	FC3	45	135	2	2	-
3		30	135	2	2	-
3		15	135	-	2	-
4		45	135	4	3	-
4		30	135	4	4	-

* Number of tests performed

Table 7. Test matrix of different EA-Foam combinations.

Foam Nr.		Thickness (mm)	Height (mm)	Impact Energy 55 Nm	107 Nm	150 Nm
2		45	135			
1	FC2	60	135	3*	3	-
3		45	135			
1	FC4	60	135	3	3	-
2(foam layer 1)		15	135			
3(foam layer 2)		30	135			
1	FC5	60	135	2	-	-
3(foam layer 1)		15	135			
2(foam layer 2)		30	135			
1	FC6	60	135	3	3	-

* Number of tests performed

Foam Nr.1 is used as base foam in all foam combinations.

Vehicle Type A Bumper System

At impact energies of 55 Nm and 107 Nm deformations of 13.5 to 16 mm occur. Maximum forces range between 8.8 and 9.7 kN for 55 Nm and between 17.7 and 20.9 kN for 107 Nm *(Figure 5)*. Due to the integrated lamps the bumper stiffness varies. The efficiency of this bumper system with hydraulic impact absorbers and bending members depends on the stiffness in the front area of the bumper. The bending member is to ensure good force transmission between the point of impact and the impact absorbers during a vehicle/vehicle or a vehicle/wall impact. The impact energy which the tibia pendulum introduces into the bumper is not high enough to cause impact absorber deformation.

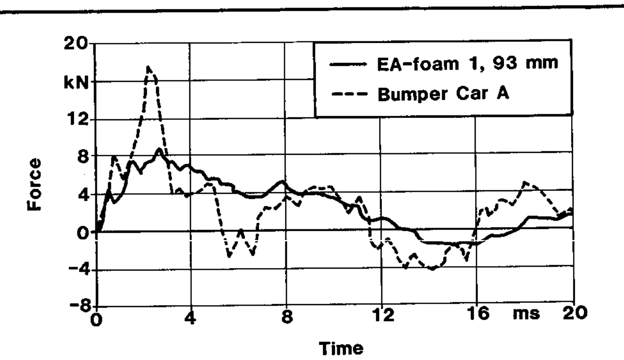

Figure 4. Force/time histories of "hard" (dotted line) and "soft" impacts

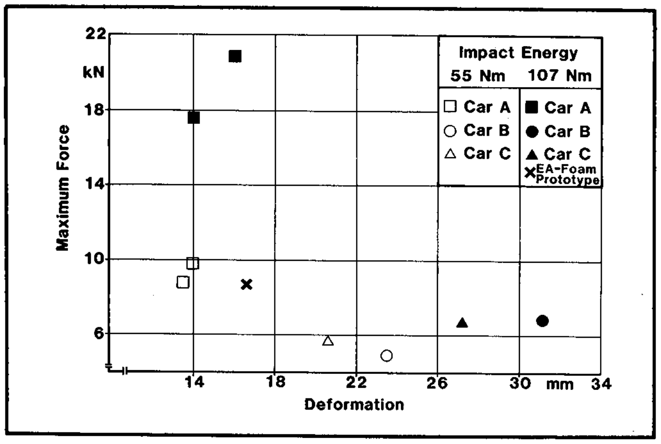

Figure 5. Force/deformation relations of different bumper systems

PUR-RIM Front End Panels Without Cross Member

The deformations measured at the PUR-RIM front-end panels without supporting member range between 44.2 and 56.3 mm at an impact energy of 107 Nm. With 55Nm, deformations of 27.7 to 43.0 mm are measured.

Maximum forces are 6.8 to 8.8 kN for 107 Nm and 5.8 to 6.0 kN for 55 Nm.

Vehicle Type B Bumper System

With this bumper system the deformation is 31.3 mm for an impact energy of 107 Nm and 23.5 mm for 55 Nm. Maximum forces are 6.9 kN and 5.0 kN respectively *(Figure 5)*.

During deformation of the front cover the impact energy is absorbed by the underlying foam layer. There are no stiff elements that would cause a force increase.

Vehicle Type C Bumper System

This bumper deforms by 27.2 mm and 20.6 mm at impact energy of 107 Nm and 55 Nm respectively. The maximum force level is 6.7 kN for high impact energies and 5.6 kN for the lower ones *(Figure 5)*.

This bumper is provided with an energy-absorbing foam layer inserted between two plastic components supported by a fibre-reinforced cross member. Most of the tibia-impact energy is absorbed by the foam layer and the bumper suspension elements.

EA-Foam Bumper Systems

This bumper prototype corresponds to bumper variant D in *Table 2*. The tibia pendulum test has been performed with skinless foam parts. With the minimum dimensions shown in *Table 2* a 16.3 mm deformation has been measured for an impact energy of 107 Nm while the maximum force level is 8.8 kN. The force level could not be reduced by increasing the foam thickness.

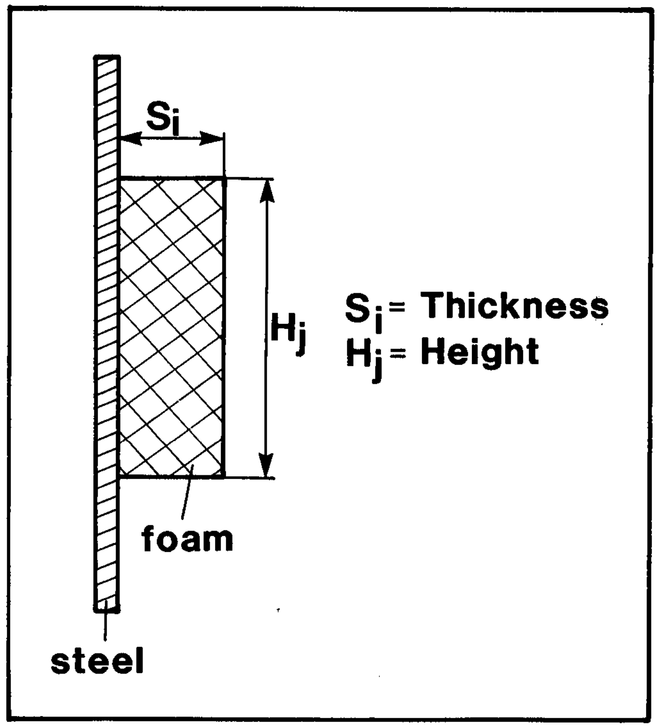

Figure 6. Test set up of steel-backed EA-Foams

Basic Tests with Energy-Absorbing Foams

Additional component tests have been performed to find out whether it is possible to remain under the target value by using different energy-absorbing foams. It must be mentioned that these foams do not meet the demands of CMSVSS 215.

Testing of Steel-Backed EA-Foams of Various Thicknesses

The energy-absorbing foams had a height of 135 mm. The foam thicknesses tested were 15 to 93 mm for foam 1 and 15 to 45 mm for foams 2 to 4. Additional tests with 30 to 60 mm thick foams (Foam height: 90mm) were performed with foam 2. The foam specimen were square-shaped *(Figure 6)*.

With an initial thickness of 93 mm foam 1 does not deform much more at an impact energy of 150 Nm (energy according to 49 CFR, Part 571) than at 107 Nm. The PU-foam thickness can be reduced to 30 mm without changing the force level reached with an impact energy of 107 Nm. Maximum forces range between 8.8 and 10.2 kN. When the foam thickness is further reduced, the force level strongly increases *(Figure 7)*. At the same time, the relative foam deformation compared to the initial thickness continuously intensifies *(Figure 8* and *9)*.

As foams not meeting the requirements of CMVSS 215 may only be used as bumper material when combined with foam 1 which fulfills these demands, the energy-absorbing foams intended for pedestrian

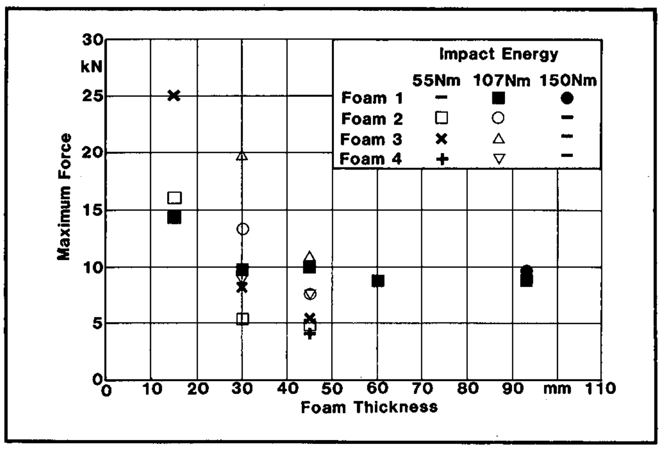

Figure 7. Force/thickness relations of different foams

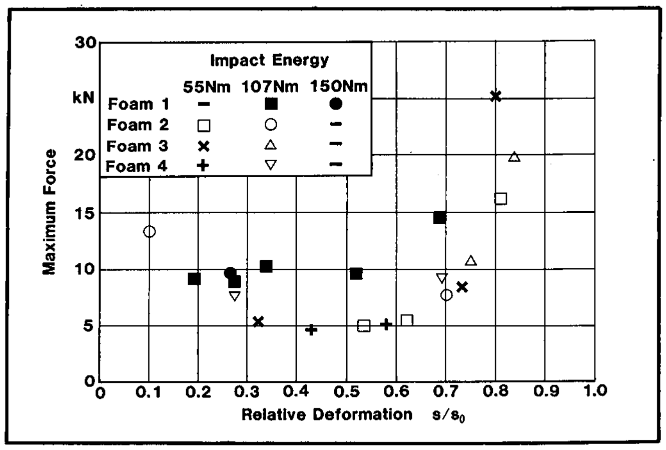

Figure 9. Force/deformation relations of different foams

protection were examined up to thicknesses of 45 mm. This dimension has been chosen to comply with the demand for a lowest possible space requirement for such EA-foam systems.

At a thickness of 45 mm, foam 2 performs better than foam 1. Maximum forces are 7.6 kN and 4.9 kN at impact energies of 107 Nm and 55 Nm respectively. At a foam thickness of 30 mm distinctly higher maximum forces are measured *(Figures 7, 8,* and *9).*

The third foam tested had the lowest density of all foams examined. At an impact energy of 107 Nm high force levels were measured for all thicknesses, because the energy absorbing capacity was exceeded. At 45 mm already it was as high as 10.7 kN. At an impact force of 55 Nm maximum force at a foam thickness of 45 mm was 5.2 kN. A similar value had been obtained with foam 2. The force level clearly increased also as the foam height was lowered *(figures 7, 8,* and *9).*

As far as foam 4 is concerned, a constant maximum force of 5.7 to 5.9 kN was measured for an impact energy of 55 Nm and foam thickness of 30 to 45 mm. At 107 Nm, maximum force increases to 7.6 kN for a foam thickness of 45 mm and reaches 9.1 kN at a thickness of 30 mm.

It is generally acknowledged, that the achievable relative deformation (related to the initial foam thickness) increases with decreasing foam thickness. EA-foam 1 used in the bumper prototype represents the lower limit deformation.

The analysis of these impact tests with various energy-absorbing foams shows that it is not possible to lower the force level to the desired value for an impact energy of 107 Nm. As far as the 55 Nm impact energy is concerned a force reduction to about 4.9 to 5.7 kN can be obtained with a foam thickness of 45 mm. The results of this specific test series show that a foam material suited for pedestrian protection should have a minimum thickness of 45 mm.

Testing of Steel-Backed EA-Foams of Various Heights

Till now, only the foam thickness has been varied even though it is basically possible to change the foam height, too. Taking the example of foam 2, the

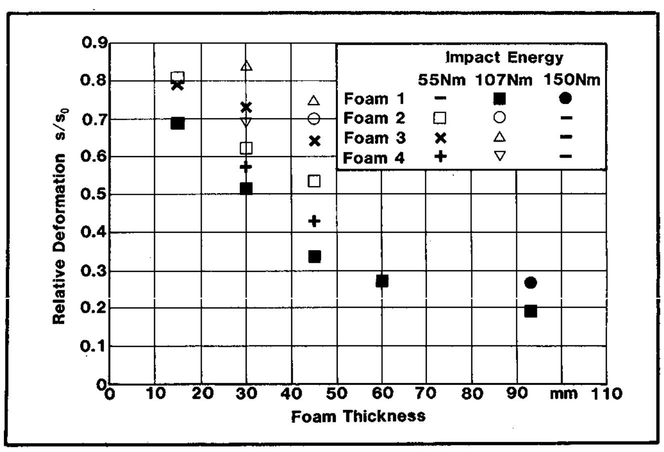

Figure 8. Deformation/thickness relations of different foams

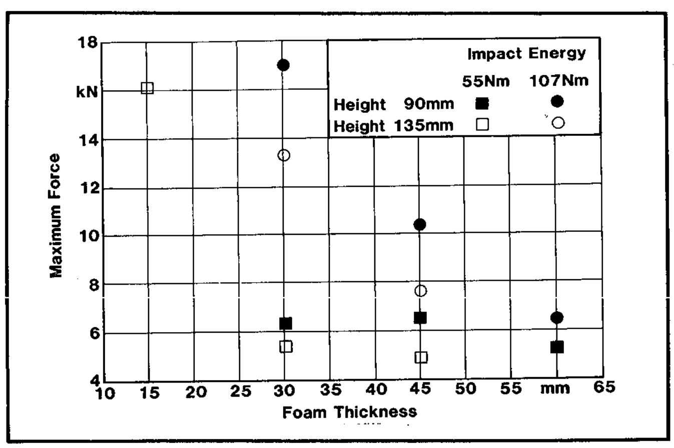

Figure 10. Force/thickness relations at different foam heights

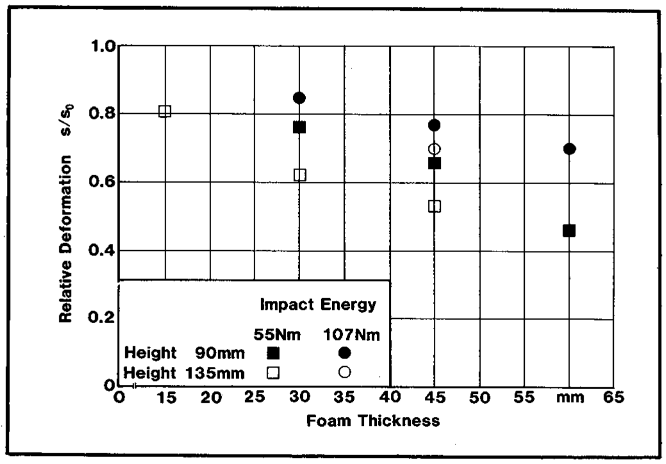

Figure 11. Deformation/thickness relations at different foam heights

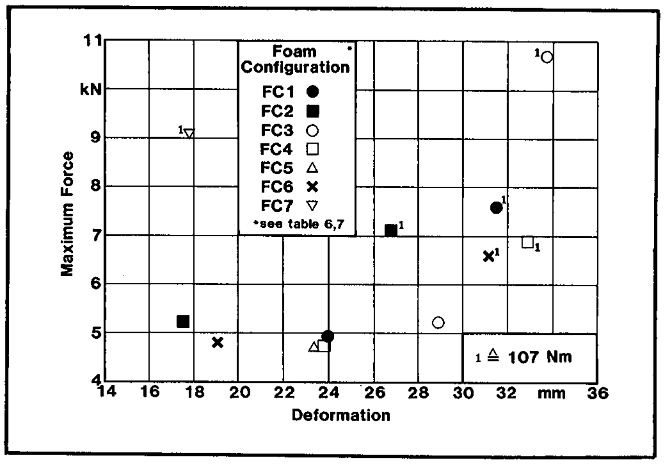

Figure 12. Force/deformation relations of different foam combinations

influence of a foam height reduction to 2/3 of the original 135 mm has been investigated. The lower foam height increases the deformation at impact energies of both 55 Nm and 107 Nm. The thickness of the lower foam must be increased by about 15 mm in order to achieve the same relative deformation.

At a 55 N energy the maximum forces of the lowered energy-absorbing foam are 1 to 2 kN higher than those measured with the original foam height *(Figures 10* and *11).* This tendency is even more pronounced with the higher impact energy of 107 Nm.

According to the results of this investigation, reducing the foam height entails essentially higher forces and relative deformations. Increasing of the foam height, on the contrary, does not result in reduced force and deformation values since the impactor is 147 mm high and thus cannot deform any additional foam.

Testing of EA-Foam Combinations

Analogous to the testing of the steel-backed foam, foams 2 and 3 were provided with a 60 mm foam-1 backing. When comparing the steel and foam-backed materials it is found that there is no force level change with foam 2 *(Figure 12)* neither at 107 Nm nor at 55 Nm. Foam-backed foam 3, on the contrary, shows a maximum force reduction to 6.9 at an impact energy of 107 Nm. At 55 Nm, the force reduction is very small. Foams 2 and 3 were also combined with foam 1 and tested.

At 55 and 107 Nm, maximum forces did not change much as compared with the 45 mm foam-backed foams.

Combined foams producing a stepped characteristic are more useful for pedestrian protection than foam 1 which only fulfills the demands of vehicle protection. But as the basic investigations have shown, the forces acting on the pedestrian can be minimized only for a defined impact energy. However, the target value of 4,000 N is always exceeded. Even though the maximum force at an impact energy of 107 Nm can be lowered by stepping the bumper curve, it is still clearly above 6.5 kN.

Influence of the Front-End Panelling

The above-mentioned test results have been obtained with unpanelled PU-foams. When testing a front end of a production car it was found that suitable front-end panelling will provide the bumper with a not increasing stiffness only under most favourable conditions. Reference /16/ reports on a stiffness increase.

Conclusions

The bumper designs used in today's production cars clearly differ in view of impact forces and deformation when submitted to a simulated pedestrian leg impact test. Bumpers with energy-absorbing foam systems offer basic advantages but require more space than hydraulic shock absorbers with cross members to ensure vehicle body protection up to impact or pendulum speeds of 8 km/h.

None of the standard bumpers has been able to reduce the load to less than 4 kN when tested with a tibia pendulum at speeds of up to 8.6 m/s. The impact forces could not be reduced either by using an elastic front end without the shock-absorber-equipped cross members required for vehicle protection. This is due to the bending and torsional stiffness and tensile strength levels required to meet the demands on the optical quality of the outer panelling, on handling during assembly and on a uniform superior production quality.

With none of the EA-foam systems the impact forces could be reduced to less than 4 kN. One of the foam systems examined, however, only slightly exceeded this value and therefore seems to be best suited for a "pedestrian-protecting" bumper layout.

The conflicting objectives of pedestrian and vehicle protection, while meeting other essential requirements such as safety regulations, demands on styling, quality, functional reliability and durability, cannot be solved by means of the bumper concepts and materials investigated in the present study.

In order to realize an appropriate compromise between the two main objectives of vehicle and pedestrian protection it must be allowed to make certain restrictions with a view to the biomechanical requirements and the demands by both manufacturers and customers.

Summary

The present study investigates the possibilities of realizing a bumper concept ensuring both good pedestrian and efficient vehicle body protection. In addition, the concept has to meet current regulations and to comply with design, production, durability and quality demands.

To determine the actual state of engineering, bumper systems of various production cars have been examined in 52 pendulum tests. The bumper systems were of basically different design. In addition, 85 pendulum tests have been performed with 4 different energy-absorbing PU-foams and their combinations. To complete the investigations, a PUR-RIM front end without the usual supporting elements (cross member with shock absorbers) has been examined to assess the influence of elastic outer panelling elements on the deformation forces.

As the analysis has shown, none of today's standard bumper systems and none of the PU-foams especially designed for low deformation forces, allows to remain under the impact force target of 4 kN at a test impact speed of 7 m/s. The conflict between the objectives of pedestrian safety and body protection can be solved only if restrictions are allowed in view of the biomechanical requirements on the one hand and manufacturer as well as customer demands on the other.

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Recognition of Pedestrians During Nighttime Traffic (Written only paper)

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Abstract

Based on the European and US low-beam situation, tests were made about the recognition distance for pedestrians. The following parameters were investigated

- shape of the headlamps
- mounting height of the headlamps
- inclination of the headlamps
- reflectance factor of pedestrians.

Introduction

Starting from the measured contrast of pedestrians in the streets during nighttime, in dynamic experiments in the street the recognition distance for pedestrians illuminated by low-beam headlamps was measured. During the tests several parameters were changed.

Pedestrians in the Street

In Figure 1 schematically a two-lane road is shown as seen by a car driver. The pedestrians are at a distance of d = 50 m in front of the car, illuminating the street by means of low-beam headlamps. C shows the cut-off-line of the two headlamps, correctly aimed with a mounting height of h = 65 cm.

Z describes the border lines to zone III in the European regulations. A is the area of an overhead sign, B of a shoulder mounted sign at 50 m distance. The Figure shows that the pedestrians are mainly illuminated by the light in-between "C" and "Z" (dotted area). This area is without legal requirements.

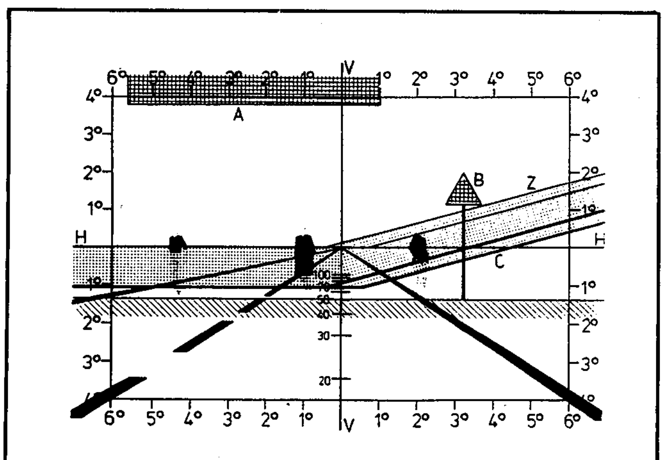

Figure 1. Perspective drawing of a two-lane road
A: overhead sign
B: shoulder mounted sign
Z: border to the illumination-zone III
C: cut-off-lines of two headlamps
Pedestrians in 50 m distance

Light-Distribution of Headlamps

In Figure 2 and 3 the vertical illumination distribution for two different types of headlamps are shown. The distribution is measured through the points 75R and B50L of the European regulations.

In Figure 2 the light-distribution for a typical European low-beam headlamp is shown, in Figure 3 for a SB-headlamp.

Reflectance of Pedestrians and Recognition Distance

In the first test the influence of the reflectance factor of the clothing of pedestrians on the recognition distance was investigated. The results are plotted in Figure 4.

The difference in recognition distance for the object-positions left-hand/right-hand side of the street is 12 . . . 15 m. The difference in distance v for black, grey, white object is 25 m. The difference between the object (white, right) and the object (black, left) is more than the factor 4.

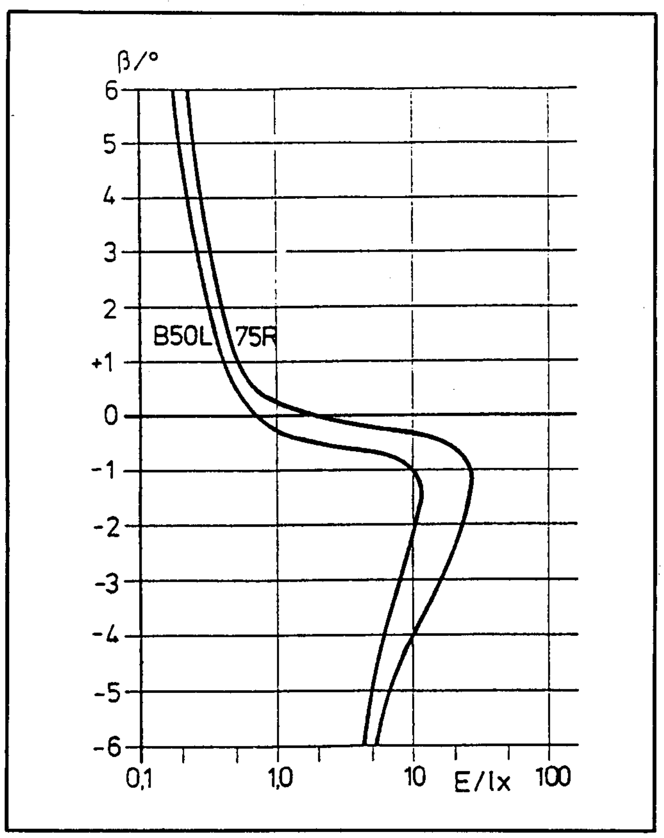

Figure 2. Vertical distribution of the illumination E for a European headlamp measured vertical through the points 75R and B50L

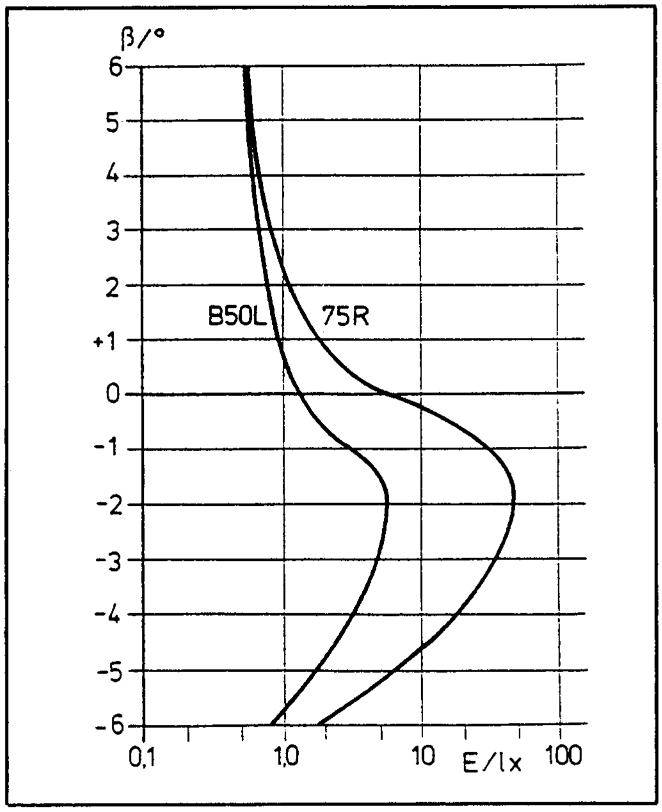

Figure 3. Vertical distribution of the illuminated E for a SB-headlamp measured vertical through the points 75R and B50L

Type of Headlamp and Recognition Distance

In Figure 5 the influence of the type of the headlamp, fulfilling the same regulation on the recognition distance is shown.

For the worst situation (object black, left) the difference in distance v between the 3 headlamps is v

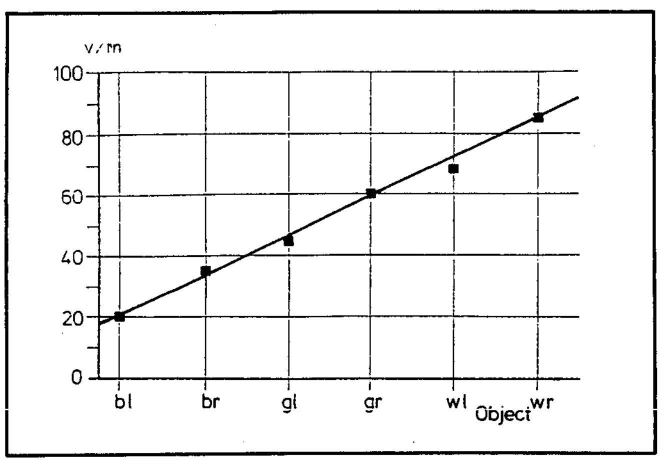

Figure 4. Recognition distance v for different objects
b: black object
g: grey object
w: white object
l, r: left, right position of the object in the street

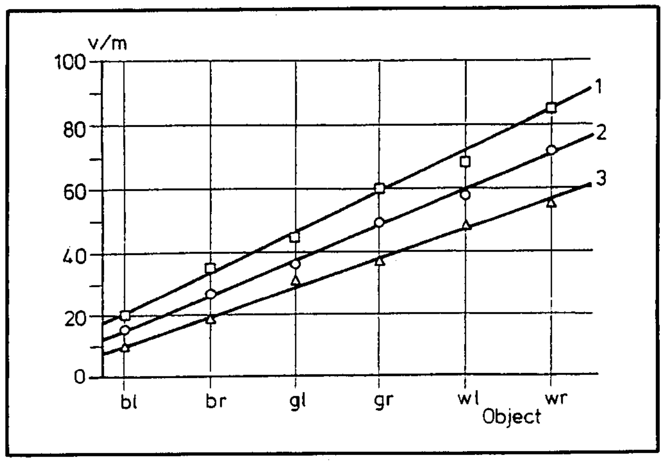

Figure 5. Recognition distance v for object and different headlamps
b : black object
g : grey object
w : white object
l, r: left, right position of the object in the street
1: rectangular headlamp
2: 7" headlamp
3: 5 3/4" headlamp

≈ 10 m. The value increases to roughly 30 m for the object (white, right).

Mounting Height of Headlamps and Recognition Distance

In Figure 6 the influence of the mounting height h of headlamps on cars on the recognition distance v is shown.

The dotted area is the mounting height used on several cars. The increase in the height h of 10 cm gives a gain of recognition distance of d≈4.5 m.

Inclination of Headlamps and Recognition Distance

The influence of the inclination of a headlamp on the recognition distance can be shown in Figure 7.

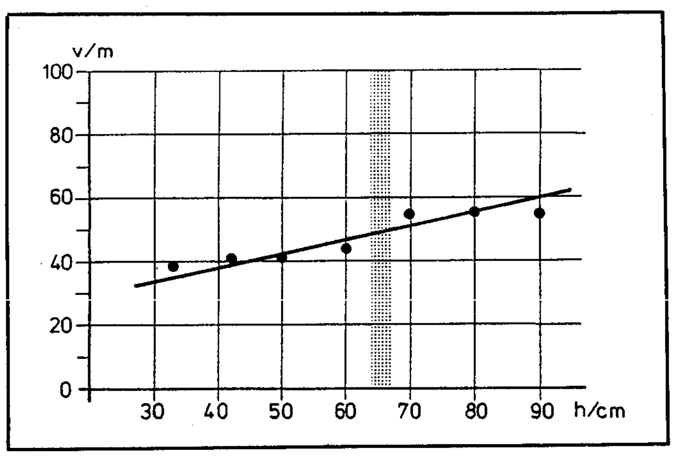

Figure 6. Recognition distance v and mounting height h of headlamps

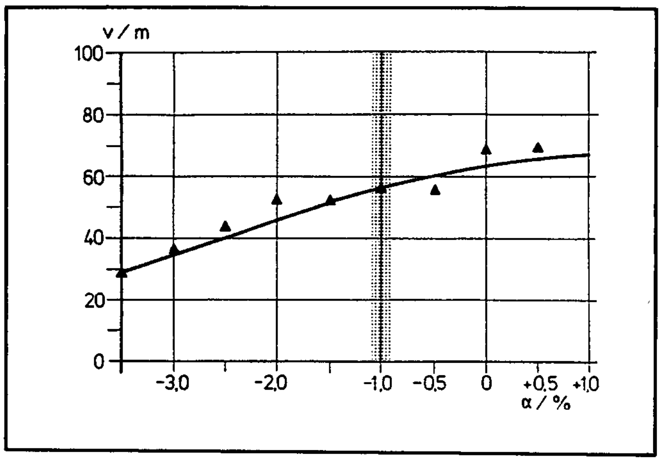

Figure 7. Recognition distance v and inclination of the headlamp

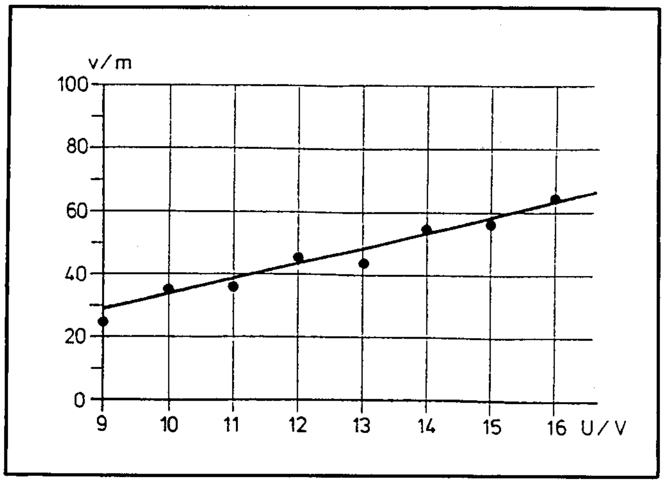

Figure 8. Recognition distance v and voltage at the headlamps

The dotted area describes the normal inclination of headlamps. With increasing inclination of the headlamp the recognition distance decreases. A change of inclination of ±1 for the normal inclination gives a change of the recognition distance v≈ ± 10 m.

Voltage of the Headlamp and Recognition Distance

A very often underestimated effect is the influence of the voltage of a headlamp on the recognition distance. In Figure 8 the influence of the voltage on the recognition distance is shown.

Out of these results a quotient of 5 m/1 V can be derived; that means a change in the voltage of 1 V gives a change in recognition distance of 5 m.

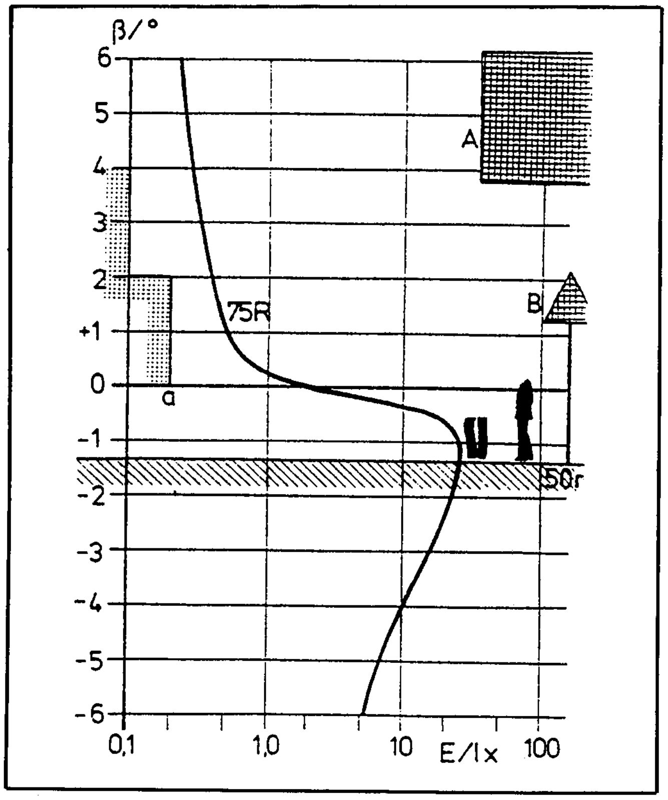

Figure 9. Vertical distribution of the illumination E for a European headlamp measured through 75R
A: overhead sign
B: shoulder mounted sign
a: limits for minimum "glare" values
Pedestrians in 50 m distance

Conclusions

The recognition distance for pedestrians illuminated by low-beam headlamps can be improved by several means,

- optimisation of the geometry and optics of headlamps
- improvement of the reflectance of pedestrians
- optimisation of the mounting height of the headlamp
- introduction of headlamp-leveling devices
- improvement in the power supply.

The possible improvement in the light distribution can be shown with Figure 9.

The pedestrians are standing at a distance of 50 m in front of the car. By introducing a controlled amount of light above the horizontal line, pedestrians closer to the car are better recognized. As a sample the dotted boundary a, as proposed in Europe, is plotted in the Figure. For better illumination beyond the horizontal line, the gradient in the cut-off must be increased; this is possible for example with elliptical headlamps.

Technical Session Eight

Motorcycle Safety

Chairman: Yoshio Nakamura, Japan

Development of a Safety Concept for Motorcycles—Results From Accident Analysis and Crash Tests

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Abstract

The high proportion of seriously injured persons and fatalities in traffic accidents involving motorcycles makes it clear that the protection of the motorcyclist is still not satisfactory. Since the middle of the Seventies it has therefore been an objective of accident research to reduce the injury risk of the motorcycle driver. By way of introduction, a survey is given of the current studies in Germany and abroad for improving motorcycle safety and of the different concepts.

The accident analyses, crash tests and mathematical simulations, carried out by the HUK-Verband, showed that the most important objective in reducing the severity of injuries is the optimization of the path of movement of a motorcyclist involved in an accident. Findings with regard to single vehicle accidents and with regard to the different collision types, if a motorcycle collides with another vehicle, are presented.

It can be seen that in both the accident in which no other vehicle was involved and in the different motorcycle/car-collision types, the reduced accident severity of the motorcyclist resulted from the controlled separation from his machine. This can be influenced by constructive measures, such as, for example, knee-pads, optimized tank and handle-bar designs, additional construction elements and possibly by an airbag in front of the motorcyclist.

Proposals for such safety designs are presented and their possible effects are estimated and discussed. On the basis of these scientific results questions relating to the practical application are dealt with.

Introduction

"Motorcycling has got safer," the motorcycle magazines in the Federal Republic of Germany announce, referring to the decline in the accident figures that can be observed in the last few years. A look at these figures, even in relation to the registrations, confirms this statement /1/.

Since 1982, the year with the largest number of motorcycle accidents and the largest number of motorcyclists killed (Figure 1), the clear downward trend has continued, although the increase in registrations still continues.

This pleasing development is due to various parameters, which, however, as they are related to active safety, will only be mentioned briefly.

The most important points which led to this development are improved training, the increasing number of places of further training, better traffic education for all road-users and technical improvement with regard to the driving stability and the brake systems /2,3,4,5,6,7,8/.

Unfortunately, this trend cannot be observed in the area of passive safety, i.e. measures aimed at reducing the severity of injuries, or, generally, the consequences of accidents.

The number of motorcyclists killed related to 1,000 traffic accidents shows (Figure 2), in the last few years, a very constant trend, i.e. over the last five years we can say that the risk has remained almost unchanged for an accident with a motorcycle.

Passive safety has therefore not yet been realised in the area of two-wheeler accidents.

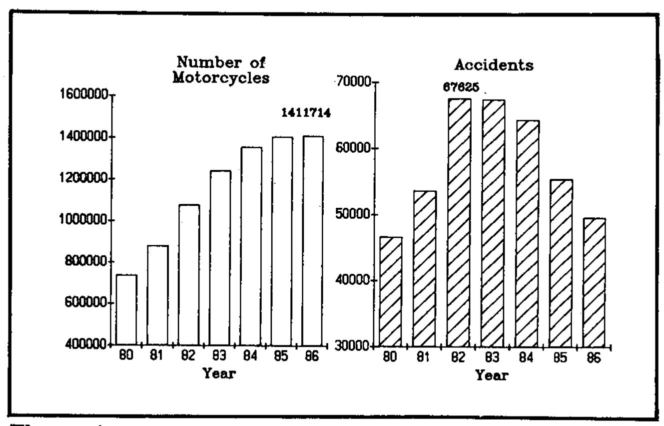


Figure 1

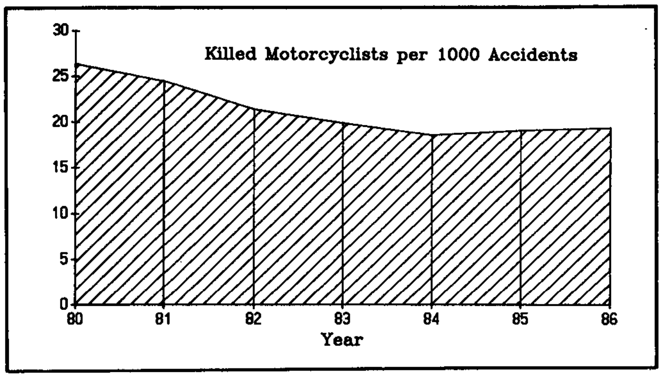


Figure 2

Man-Vehicle-Environment

A safety concept for motorcycles must cover all areas of the road traffic (Figure 3). This means that passive safety must also be related to the areas "Man" and "Environment" and must not deal solely with the "Vehicle".

Safety elements in the area "Man" relate mainly to the protective clothing and the crash helmet.

The most recent developments in this field show a clear increase in safety compared to the protective suits of only 5 to 10 years /9,10,11/.

One of the new materials, a special foam substance, only mentioned here as one example, is worked into the suit and thus increases the protective effect in the event of the direct application of force /13/. At the same time, however, movement is not impaired, so that these additional protective plates do not have a negative effect on active safety.

In the area "Environment" passive safety must concentrate, first and foremost, on neutralising aggressive obstacles at the side of the road /14,15/, but further improvements still have to be worked out for the other road-users /16/.

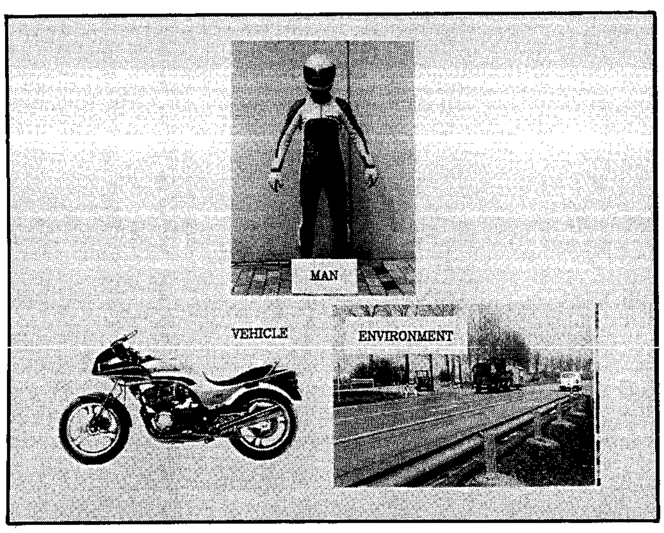


Figure 3

The most recent work in this field shows a construction with which the posts of guardrails can be enclosed. This reduces the impact against the guardrail posts, which often results in critical injuries and amputations.

Arranging the rest of the road demarcations with the motorcyclist in mind and neutralising the accident opponent's vehicle contours are also elements of importance in this area.

There remains the vehicle itself, which in the last 30 years has undergone hardly any changes as far as passive safety is concerned.

But the prerequisite for the development of a safety concept for motorcycles is the knowledge of how accidents happen, i.e. the motorcyclist's extremely different movement sequences as they occur in motorcycle accidents and the chances of injury which these involve have first to be analysed /17/.

Accident Systematic and Injury Distribution

A classification system which should be capable of describing *all* the possible motorcycle accidents does not work because of the large number of possible accident sequences and/or kinds of movements. For this reason it has proved practicable to divide up the ways in which motorised two-wheelers are involved in accidents into four main accident groups (Figure 4) which make a fundamental distinction in the ways in which the injuries arise but allow variations within the group /18/. These accident groups are:

- side collision (the motorcycle is hit)
- frontal collision (the motorcycle hits another vehicle with its front)
- grazing collision
- single accident

The first division is into single accidents and collisions. The collision groups themselves must be

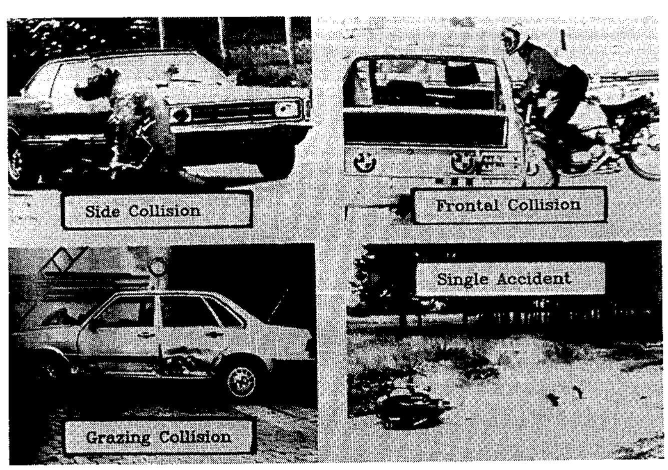


Figure 4

subdivided into three different movement sequences, because a combined assessment of these accidents tell us nothing clearly about how the injuries arise.

The frequency of these accident groups (Figure 5) shows that the collision of a motorcycle with another road-user dominates.

A large number of undetected cases must be expected in the proportion of single accidents, since many of these accidents are not recorded by the police or by an insurance company. This is the only possible explanation for the fact that about one-third of all motorcyclists killed—this figure has remained constant over the last few years—lost their lives in single accidents, although the proportion of single accidents never exceeds about 20% in all the material /1/.

In the distribution of the injuries related to the two collision types "frontal- and side collision" (Figure 6) it can be seen that the proportion of injuries to the lower extremities come first and are nearly equal. The difference of these two accident groups is shown by the injuries of the head. For the frontal collision it is therefore the main aim to reduce the risk of head injuries.

Generally you can say, that the injury risk for a motorcycle-user is lower when he is engaged in a side-collision. In a frontal-collision, in nearly every body region the frequency of injury is higher.

Realisation of Safety Concepts

The frequency of the last-mentioned accident group, the severitiy of injury within this group and the possibility of constructively working out improvements for this accident group, were, at the beginning of the research work in the two-wheeler sector, the reasons for concentrating on this accident group alone /20/.

In the meantime improvements and new safety elements have also been developed for the remaining accident groups and they will be presented below. But in the practical application of the safety elements developed it is important that a safety element which works well in one accident group should by no means have negative effects on one of the other accident groups. On the one hand, this special search for safety elements and, on the other hand, comprehensive knowledge of the overall two-wheeler's accident picture has led to the development of a concept, some of the main features of which have already been taken up by the manufacturers.

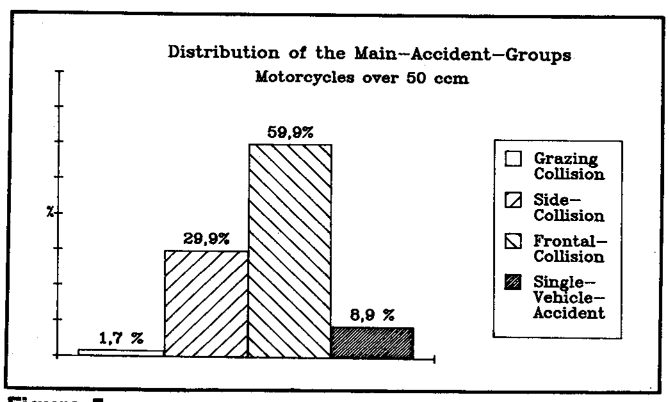


Figure 5

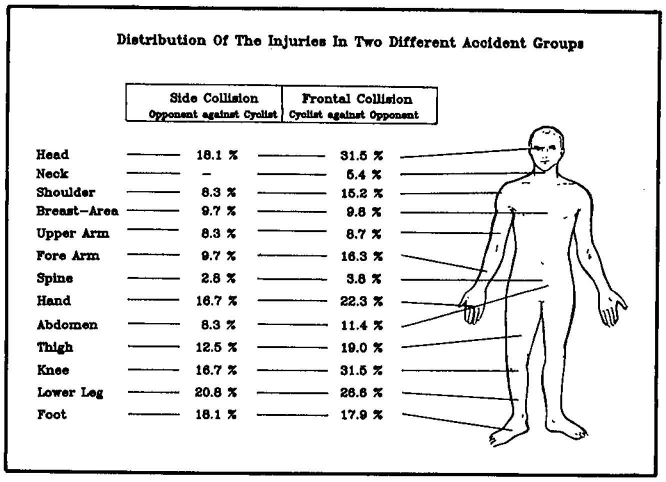


Figure 6

Directions of Passive Safety

So how can the motorcyclist's injury risk be reduced? In recent times two directions (Figure 7) have taken shape, both of which aim at reducing injuries but which go fundamentally different ways.

One of these directions is concerned with the possibility of influencing the movement path of the motorcyclist, so that dangerous contacts with his motorcycle and/or with the accident opponent can be avoided. This direction concentrates on avoiding the motorcyclist becoming entangled with his own machine and on deflecting the body when impact with an obstacle occurs.

In the second case, the motorcyclist should be held more firmly in position by creating a restraint system which, similar to the one used for car occupants, passes on a tolerable deceleration resulting from the design of the machine to the driver.

The practical solution provides for a safety belt or knee pads in front of the driver's chest /21,22,23/.

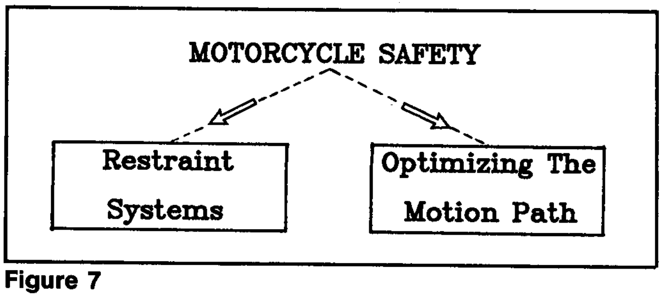


Figure 7

A prerequisite for this equipment to work is the controlled speed reduction by an appropriate design of a deformable zone. But this is not possible with a normal motorcycle, since the design of the motorcycle would have to be fundamentally changed if a restraint device of this kind is to fulfill its purpose.

An example (Figure 8) of what this may look like in reality was presented at the last ESV-Conference in the fringe programme two years ago and with this kind of vehicle a gain in safety by using a safety belt is within the bounds of possibility, but for the normal construction of a motorcycle the idea of a safety belt should not be pursued any further.

Influencing the flight path, that is the second possibility of realising a passive safety concept for the motorcycle, can be integrated into the present-day design of a motorcycle.

Frontal Collisions

Optimizing the motion path

The basis of this concept was the result of studies which showed that, in comparable accidents, i.e. collisions in which a motorcyle runs into an accident opponent, the motorcyclists were injured least severely when they were able to "fly over" their accident opponent /24,25,26/. This may take place completely or partially; in any event the direct impact of the head against the car must be avoided.

The evaluation of these accidents shows that the severity of injury in the case of a "fly-over" was, in every speed range, below the injury severity of the motorcyclist who impacted directly with the accident opponent.

A comparison of the injuries to the individual parts of the body (Figure 9), broken down according to "fly-over" and "impact", also makes it clear that the frequency of the serious head injuries decreases in the case of a "fly-over".

Distribution Of The Injuries In Different Motion Sequences

	" Impact " AIS 5 – 6	" Impact " AIS 3 – 4	" Impact " AIS 1 – 2	" Fly-Over " AIS 1 – 2	" Fly-Over " AIS 3 – 4	" Fly-Over " AIS 5 – 6
Head	14%	15%	79%	42%	3%	8%
Neck						
Shoulder		4%	21%	29%		
Breast-Area	2%	6%	6%	8%		
Upper Arm		4%	14%	7%	1%	
Fore Arm		10%	14%	12%	1%	
Spine			2%		1%	
Hand		2%	31%	29%	4%	
Abdomen	2%	8%	4%	13%	3%	
Thigh		6%	14%	29%	7%	
Knee			44%	36%		
Lower Leg		8%	19%	34%	12%	
Foot		2%	21%	20%		

Figure 9

With the lower extremities, the differences are not as clear. This is due to the fact that, even in the case of a "fly-over" the motorcyclist's legs can be caught on the handlebars or can graze the accident opponent if there is no knee-pad.

First proposal for a safer motorcycle

The findings of the analyses mentioned of real-life accidents were tested and confirmed by us by means of experimental simulation and mathematical calculations.

This first test serie (Figure 10) was already presented at the last ESV Conference, and, as a result of these experiments, a safety motorcycle /27/ was presented which possessed the following additional and/or altered characteristics.

By combining knee-pads in front of the legs of the driver with a touring handlebar it was possible, on the one hand, for the driver to avoid leg contact with his own handlebar and the accident opponent and, at the same time, to reduce a direct impact of the head against the edge of a car's roof by means of an

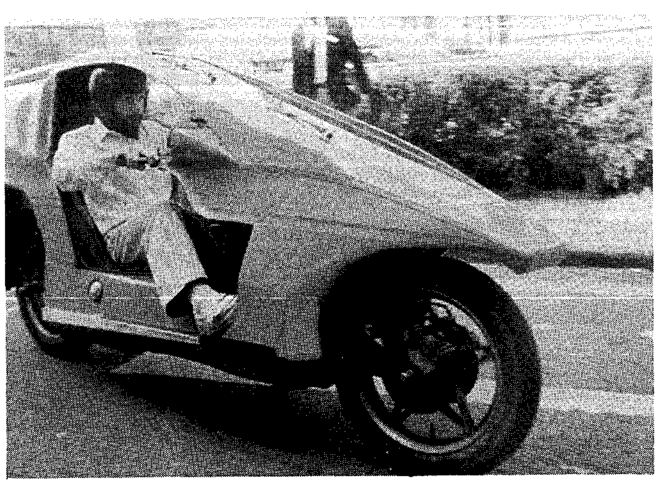


Figure 8

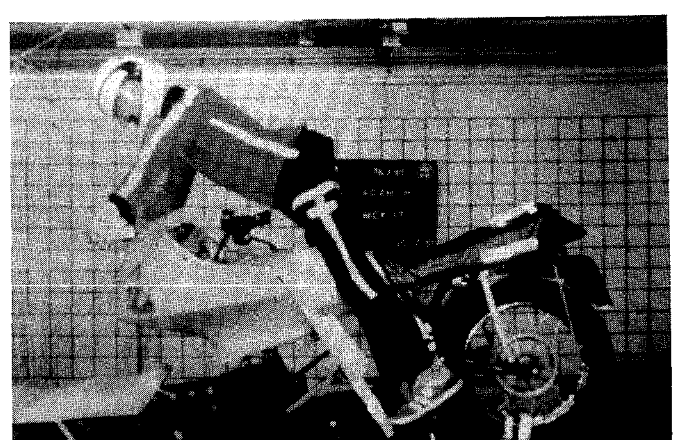


Figure 10

upright sitting position. On impact the motorcyclist did not become entangled in his machine and the prerequisite for a "fly-over" with reduced injury risk was created. This concept is supported by a ramp-like design of the tank, since it is the lower part of the driver's body which first strikes this part of the machine after a pause of about 30 ms. Raising the seat also results in further advantages, but it is in contradiction to considerations of driving dynamics, since the position of the centre of gravity has an effect on the handling of the motorcycle. A suggestion proposed by us at that time was to design the seat with a variable height, so that even drivers of different physiques would be able to handle one and the same motorcycle ideally.

A further, although very small, contribution, namely to take the head away from the accident opponent's danger zone, can be made by an anti-dive system, when, during an emergency braking, the dipping of the front wheel fork and thus the lowering of the body are avoided. Even the few centrimetres thus gained in height can be decisive for the severity of the injuries.

These measures that were proposed were the first step towards optimizing the driver's movement path; but it also became clear that a direct impact could not be completely avoided. For this reason the possibility of the motorcyclist actively inducing his flight path by bringing his body into the upright position shortly before the collision was also pointed out. That is, of course, not possible in all cases, and this is the reason why a closer study was made of another safety element which up to now has only made an appearance in the car sector, the airbag.

The airbag could, under circumstances (Figure 11), perform a function within the safety elements of the motorcycle as well. The most important prerequisite is, however, that the airbag is not conceived as a restraint system, but that its function is also extended to include the influencing of the flight path.

The first tests with airbags on the motorcycle go back 15 years and unfortunately nothing is known about a continuation of this work. It was not until 1985 at the ESV Conference in Oxford that two proposals for a "two-wheeler-airbag" was published /29,30/, one of the contributions being based on a test series by our office. This test series has, in the meantime, been continued and its findings can be summed up as follows.

Experiments were carried out with and without the airbag on a sledge facility, which allows the simulation of the simplified cause of a collision.

The motion sequence without an airbag (Figure 12) shows the direct impact against the simulated obstacle. The dummy's head smashes into the obstacle in the area of the visor's opening and at the same time its legs are caught up on the handlebar.

The same experience with an airbag and a knee-pad (Figure 13) clearly shows that no contact takes place between the head and the obstacle and that the motion path of the dummy is reflected upwards. This test series was made at a speed of 40 km/h, and at higher speeds it is to be assumed that the component of the dummy's upward direction of movement will be shifted further forward.

In the case of a frontal impact therefore a safety effect of the airbag can be observed. The problems which have still to be overcome thus concentrate on the economic feasibility, since the technical realisation of the triggering device still requires extensive work. In this connection the question of the all-mechanical airbag /31/ also has to be discussed, which in an amazingly simple way makes the ignition of the airbag a purely mechanical problem, without electrical sensors.

Nevertheless, there are still some problems to be solved, the following points being of paramount importance:

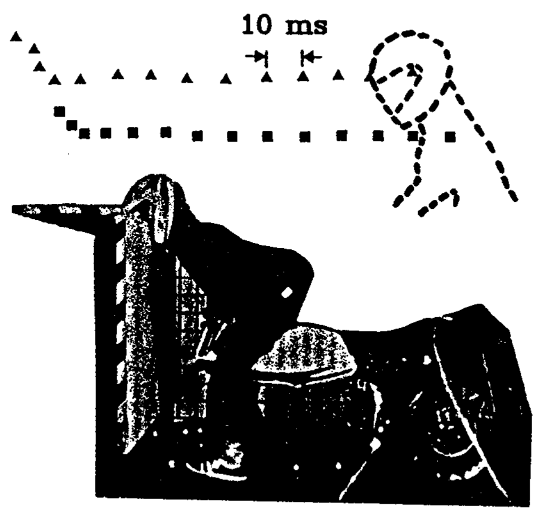
- triggering reliability
- sensor development
- economy.

Function Of A Motorcycle Airbag

Cushioning The Impact

Influence On The Flight Path

Figure 11



Dummy Motion without Airbag
40 km/h Collision Speed

Figure 12

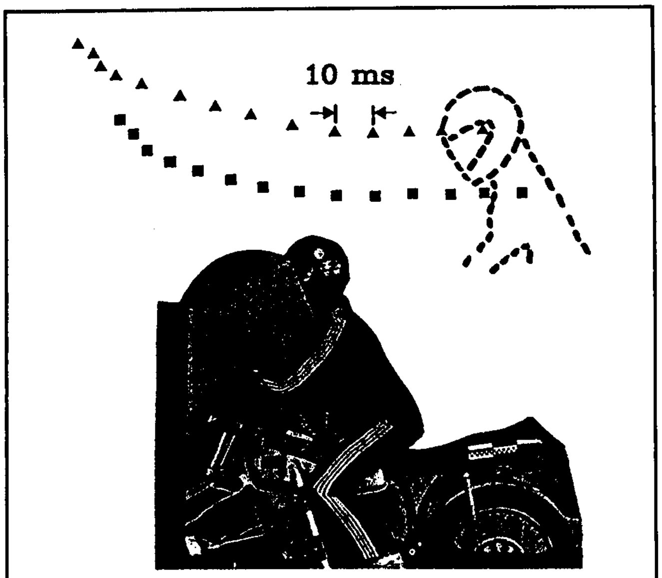


Figure 13

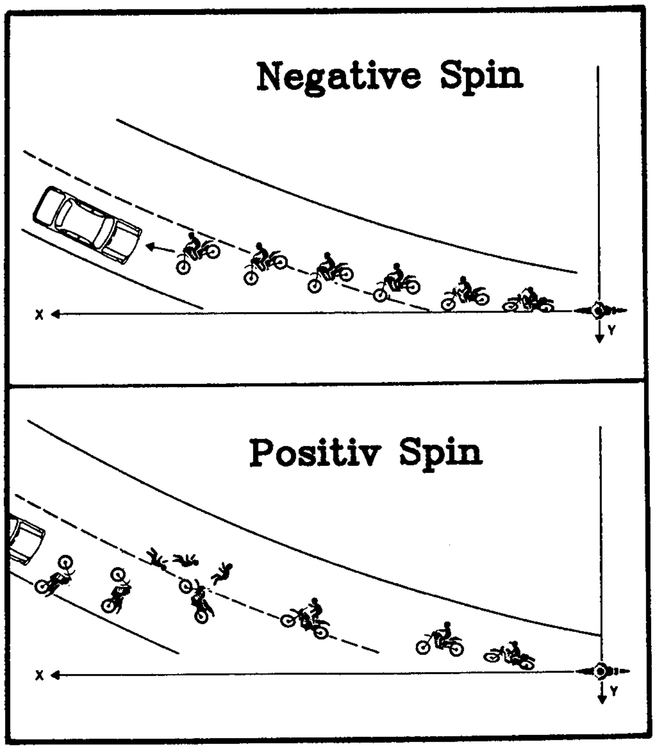


Figure 14

The usefulness of crash sensors on the basis of a special sound emission when the front wheel fork is bent is at present being examined by us. Combined with a normal deceleration element, this triggering mechanism could, under circumstances, meet the high demands required in the case of the motorcycle.

Single Accidents

For the remaining accident groups it has already been stated that the injury risk of the lower extremities comes first. At the same time it also applies to these accident groups that the injury risk of the whole body can be reduced if there is a separation of the motorcyclist from his machine. In the accident group termed "single accident" a new possibility of helping this separation to take place has emerged. By analysing 300 single accidents it was shown that a positive turning movement (positive spin) of the motorcycle (Figure 14) that has fallen over, i.e. when the motorcycle continues to turn in the curve through which it is driving, helps to bring about a separation of driver and machine /32/.

If a negative turn takes place (negative spin) and the motorcyclist cannot separate himself from his machine and he is deflected into the path of on-coming traffic together with the motorcycle and may thus become endangered by the on-coming traffic.

In a subsequent experimental and mathematical simulation it was possible to define special friction areas which have an effect on the turning movement of the skidding motorcycle.

Thus an area with the highest possible friction values in the region above the motorcycle's centre of gravity in combination with an area with a low friction value below the centre of gravity produce the best conditions for a positive turning movement. These friction areas could be integrated relatively unobtrusively into a motorcycle by means of the covering panels. In addition to this, however, the footrests, gearchange and brake would also have to be modified somewhat in their design, so that the anti-friction properties are improved.

At any rate, this development shows that by means of systematic analysis solutions can always be worked out which result in a further step towards passive safety.

Grazing Collision and Side Collision

With the last two accident groups it becomes clear how endangered the lower extremities of the motorcyclist are. In the case of these accident groups any contact is aimed at the area above the footrests up to about the level of the tank. While most grazing collisions lead to serious smashing of the knee or amputations, in the case of side collisions for the most part bruises and fractures can be observed. A safety concept for these groups has to overcome, first of all, the limited possibility of working out any design improvements.

Although the protective cages some people have proposed /33/ do help, they turn the motorcycle into a specialised vehicle once again. The only feasible way is to strengthen the covering panels, if a knee pad is already in place in front of the legs, in combination with an additional strengthening of the side bags. The protective space thus gained has, above a certain impact speed, no longer any function, but it certainly helps in all cases in which slight to medium injuries are nowadays still sustained /34,35/.

Summary

A safety concept for motorcycles must cover all areas of the traffic scene. Measures taken with the motorcyclist, such as improved protective suits and helmets, adapting the environment by neutralising dangerous road demarcations and aggressive accident opponents and optimizing the motorcycle itself result in a reduction of the injury risk. In this connection it is absolutely essential to divide up the different sequences in order to work out how the injuries arise.

A synthesis of the proposed safety elements of the individual accident groups, while observing the rule that no negative effects on the neighbouring groups can be observed, results in a safety concept for a motorcycle (Figure 15) which constitutes a clear gain in passive safety.

Depending on the accident groups, the chances of success in reducing the injury risk can be recognized. The greatest prospects of success will come about in the case of frontal collisions of motorcycles, since here the design elements can show the greatest effect. Making it possible for the motorcyclist and his machine to be separated and to overcome the obstacle—both of these actions involving a low injury risk—shows the greatest chances of success.

In the remaining accident groups it is always the impact energy of the accident opponent and/or the post-crash movement in the case of a fall which decides the final injury pattern. It is very difficult to influence these parameters by taking measures to change the usual design of the motorcycle.

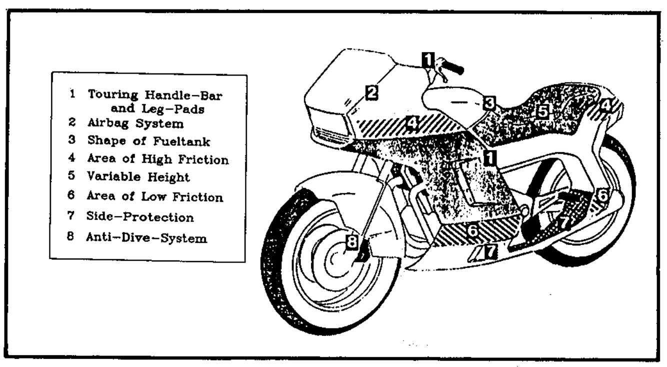


Figure 15

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A Study on Methods of Measuring Fields of View of Motorcycle Rearview Mirrors

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Abstract

This study involved 26 American motorcycle riders in an effort to establish measurement methods of rearward field of view of motorcycle rearview mirrors. A survey of rear-view mirror aiming, measurements of rider arm contour and computer simulations of rearward field of view were part of the study.

1. The survey of rear-view mirror aiming was carried out by checking the adjustments that the test riders had made on the test motorcycle with fairing mounted rear-view mirrors and on their own motorcycles. The results indicated that, to measure the rearward field of view, the rear-view mirrors should be adjusted at an angle where a light beam travelling from reference eye-point to mirror center was reflected in a horizontal and parallel manner to vehicle center line.
2. Arm contour was measured by taking photographs from the top and the side of the shoulder and arms of riders seated on two types of motorcycle mock-ups. Results showed that arm line was the significant factor in determining rider arm shadow.
3. Computer simulations of rearward field of view was conducted for three types of motorcycles. This resulted in a computer simulation program to examine the rearward field of view of motorcycle rear-view mirrors and measurement methods when using a three-dimensional manikin for motorcycles placed on motorcycle.

Introduction

A method for measuring the rearward field of view is a precondition for examining the design requirement (i.e. rear-view mirror curvature, size and mounting position) and the performance requirements (field of view reference) that determine the field of view of motorcycle rear-view mirrors.

In order to establish measurement method of rearward field of view of motorcycle rear-view mirror, it is necessary to determine the followng items.

1. Method of eye point determination as a reference point for field of view measurement.
2. Method of mirror aiming determination for selection of field of view.
3. Method of arm contour determination that prescribes inner field of view.

The method to establish eye point which is reference point for the field of view of rear-view mirror for motorcycle, and the development of the three-dimensional manikin for motorcycle which was necessary for the determination of eye point on the actual vehicle were reported based on the data of 155 American motorcycle riders at The 10th International Technical Conference on Experimental Safety Vehicles (Motoki & Asoh, 1985).

In order to use a manikin for measuring the rearward field of view of motorcycle rear-view mirrors and the method of determining arm contour had to be established.

Thus a survey was conducted to examine the rear-view mirror aiming methods of American motorcycle riders. Photographic measurements from the top and side were made of arm configurations of riders seated on a motorcycle mock-up to examine arm contour.

As the results of these studies, we propose the methods of measuring for field of view of rear-view mirror by using the three-dimensional manikin for motorcycle seated on the actual motorcycle and applications of computer simulations to examine the field of view of rear-view mirrors at the design stage.

Anthropometry

Purpose

Anthropometric measurements of a number of American motorcycle riders were made to obtain a variety of anthropometric data. These data were compared with eye-point measurement data[1] of 155 American motorcycle riders.

Method

Measuring Instruments. Anthropometers and a sliding caliper were used for the measurement.

Measuring Points. Before starting the measurements, measuring points indicated in Figure 1 that were difficult to locate visually (e.g. cervical, left and right acromions, right radiale, left and right trochanterions and sphyrion) were located manually and marked.

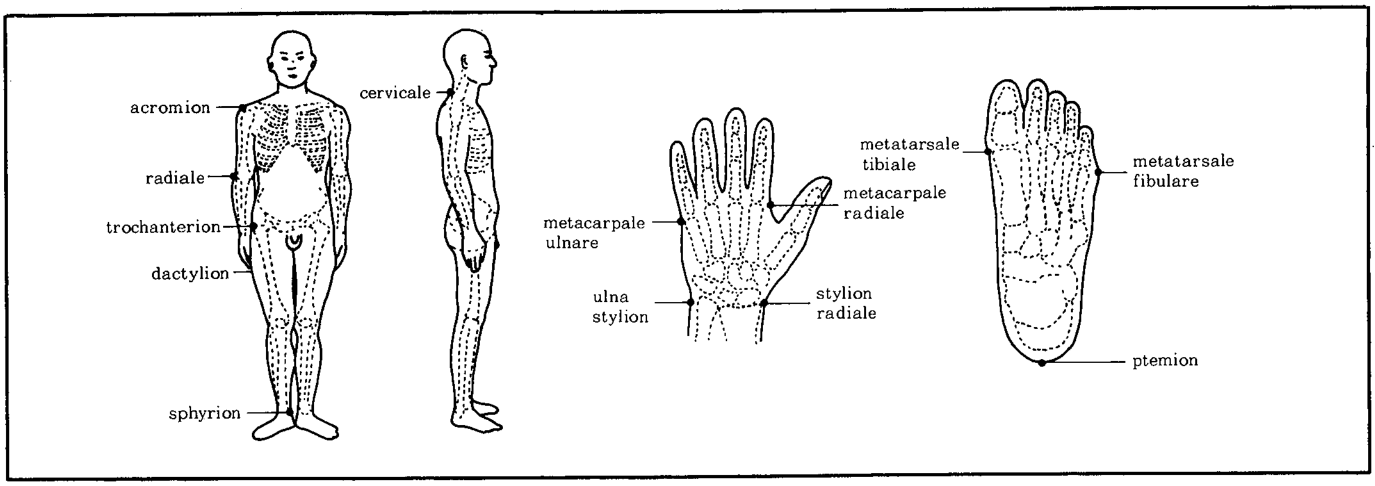


Figure 1. Measurement points

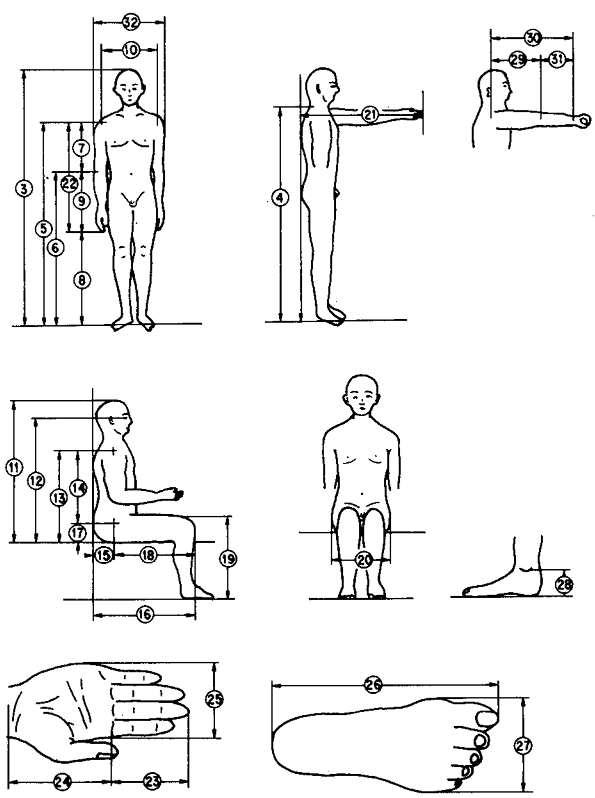
Anthropometry Items and Physical Members for Measurement. The anthropometry items and physical members for measurement are shown in Figure 2.

(1) Measurements in standing position

The subjects were placed in a natural standing position with the face forward, the right and left tragions and the right orbitale in the horizontal plane (i.e. ears and eyes horizontally). The arms were kept close to the body with the palms lightly touching thighs. (This position was not used for measurements 21, 29, 30 and 31).

(2) Measurements in seated position

The height of the chair was adjusted so that the subjects occupied a natural position with thighs nearly horizontal and the legs bent at a right angle at the knees. The head had the same position as in standing.



These 32 items are as follows:
1 (Age)
2 Body weight
3 Stature
4 Cervical height
5 Acromion height
6 Radiale height
7 (Acromion height) − (Radiale height)
8 Dactylion height
9 (Radiale height) − (Dactylion height)
10 (Biacromial breadth)
11 Sitting height
12 Eye to seated surface
13 Acromion to seated surface
14 (Acromion to seated surface) − (Trochanterion to seated surface)
15 Buttock−trochanter length, sitting
16 Buttock−knee length, sitting
17 Trochanter to seated surface
18 (Buttock−knee length, sitting) − (Trochanter to buttock, sitting)
19 Knee height, sitting
20 Trochanter breadth, sitting
21 Armreach from back
22 Arm length (Acromion height−Dactylion height)
23 Finger III length
24 Palm III length
25 Hand breadth at metacarpale
26 Foot length
27 Foot breadth
28 Medial malleolus height
29 Shoulder−elbow length (Acromion to radiale)
30 Shoulder−wrist length (Acromion to ulha stylion)
31 Elbow−wrist length (Radiale to ulna stylion)
32 Shoulder breadth (Maximum)

Figure 2. Anthropometry items and physical members for measurement

The subjects wore only jogging shorts while being measured.

Subjects. The subjects included 26 American motorcycle riders who used motorcycles as everyday affairs, with an average age of 29 years.

Results

Table 1 shows a comparison of the eye-point measurement data obtained in anthropometric measurements of 155 American motorcycle riders in 1985[1]) and the data gathered in measuring the 26 motorcycle riders who took part in the present study.

This comparison shows that the subjects used for the present study were an average 3 kg lighter and 21 mm lower in stature. However, anthropometric data closely related to riding position were about the same. Thus the differences were only 4 mm in biacromial breadth, 4 mm in eye to seated surface, 4 mm in acromion to seated surface and 1 mm in arm length.

Measurements in Rear-View Mirror Aiming

Purpose

The rearward field of view varies greatly with the angle of the mirror. The purpose was to measure the aiming of rear-view mirrors and examine actual conditions of mirror angle adjustments on a European type motorcycle adjusted by riders and on motorcycles owned by them.

Method

Coordinate System. Three-dimensional indications of data in this study employ an orthogonal coordinate system where R′-point (the point of intersection of a vertical line through R-point and seat surface; refer to JASO T 006[2]) is used as origin (see Figure 3). Vehicle reference points and riding position are represented as shown in Figure 4 (refer to JASO T 005[3]).

Test Motorcycles. One European type motorcycle (see Figures 5, 6 and Table 2) with fairing mounted rear-view mirrors and the test riders' own motorcycles were used in the test. Three of the riders' own motorcycles had fairing mounted rear-view mirrors and the other 23 motorcycles had handlebar mounted rear-view mirrors. The measurements were made with each rider seated on the test motorcycle and on his own motorcycle; thus two types of data were gathered for each rider.

Test Motorcycles and Screen Locations. Relative locations between test motorcycles and screen is shown in Figure 7. The screen was placed 10 meters behind eye-point and was orthogonal to vehicle center line. The screen was ruled into 300 mm squares and each square was provided with a code to facilitate identification.

Test Motorcycles and Camera Position. The camera was placed on the vehicle center line 8 meters in front of R′-point. The height of the camera was identical with the R′-point of European type test motorcycle.

Table 1. Comparison of anthropometric data among American riders (mean value).

Item \ Date	1985 N=155	1987 N=26	Difference 1987−1985	Item \ Date	1985 N=155	1987 N=26	Difference 1987−1985
1. (Age)	29 (7)	29 (8)	0	17. Trochanter to seated surface	89 (9)	84 (9)	−5
2. Body weight	80 (12)	77 (14)	−3	18. Knee to trochanter length	488 (24)	490 (25)	+2
3. Stature	1775 (67)	1754 (66)	−21	19. Knee height, sitting	555 (28)	554 (26)	−1
4. Cervical height	1533 (64)	1515 (64)	−18	20. Trochanter breadth, sitting	375 (26)	366 (32)	−9
5. Acromion height	1454 (63)	1439 (63)	−15	21. Armreach from back	895 (44)	882 (46)	−13
6. Radiale height	1122 (50)	1108 (52)	−14	22. Arm length	780 (34)	779 (36)	−1
7. Acromion height -Radiale height	333 (21)	332 (16)	−1	23. Finger III length	83 (5)	84 (5)	+1
8. Dactylion height	674 (38)	660 (40)	−14	24. Palm III length	109 (6)	111 (7)	+2
9. Radiale height−Dactylion height	447 (31)	448 (24)	+1	25. Hand breadth at metacarpale	89 (4)	87 (5)	−2
10. Biacromial breadth	416 (18)	412 (21)	−4	26. Foot length	268 (13)	271 (15)	+3
11. Sitting height	914 (45)	915 (38)	+1	27. Foot breadth	107 (5)	106 (8)	−1
12. Eye to seated surface	786 (33)	790 (35)	+4	28. Medial malleolus height	76 (7)	75 (6)	−1
13. Acromion to seated surface	601 (40)	597 (33)	−4	29. Shoulder−elbow length	—	320 (14)	—
14. Acromion to trochanterion height	512 (39)	513 (29)	+1	30. Shoulder−wrist length	—	573 (25)	—
15. Buttock−trochanter length, sitting	125 (12)	119 (9)	−6	31. Elbow−wrist length	—	253 (14)	—
16. Buttock−knee length, sitting	610 (30)	609 (30)	−4	32. Shoulder breadth	—	470 (29)	—

Note: ( ) S.D. Unit (Age: Year, Weight: kg, Dimension: mm)

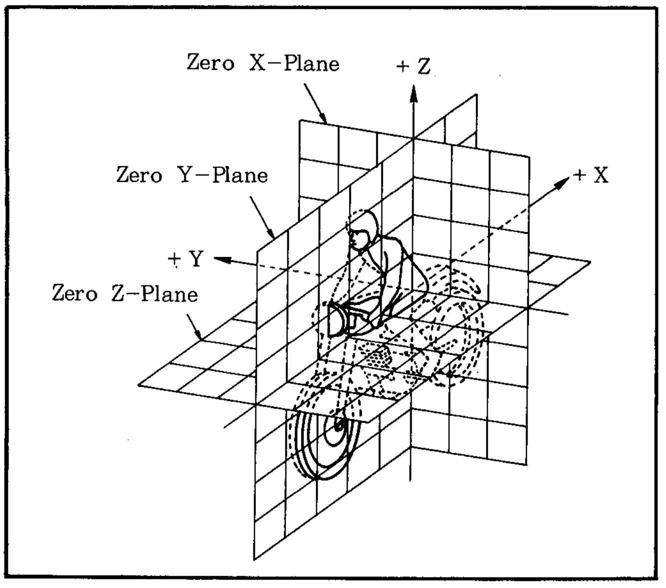


Figure 3. Three-dimensional reference system for motorcycle

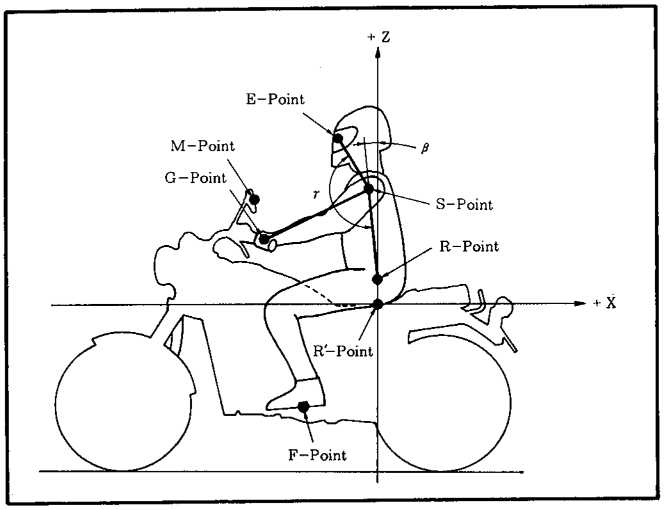


Figure 4. Marking method for riding position

Test Riders. 26 male American motorcycle riders participated in this test (as in Anthropometry).

Measuring Procedures.

(1) The riders rode the test vehicles over a 1 km straight course and adjusted the angle of the rear-view mirrors. The motorcycles brought in by the test riders were exempted from this part of the procedures.
(2) Motorcycle fixing jigs were used to fix the vehicles in an orthogonal position to the screen (see Figure 8).
(3) When the riders had taken a natural riding position on the motorcycles, it was checked that the vertical center plane of the vehicles were perpendicular. This was done with the help of the horizontal line on the focusing screen of the front camera.

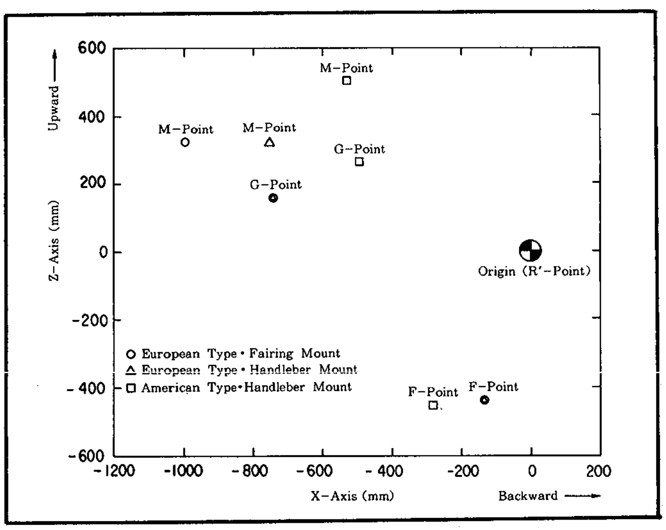


Figure 5. Locations of vehicle reference points (side-view)

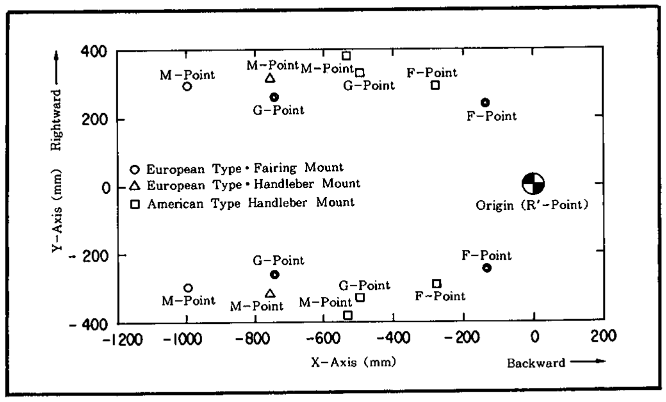


Figure 6. Locations of vehicle reference points (plan-view)

(4) The center of the right and left rear-view mirrors was marked and the distance between the two marks and their height were measured.
(5) The eye locations when a rider faced straight ahead and when he looked at the mirrors were photographed (Figures 9 and 10).
(6) The riders were asked to look at the mirrors and report which square of the screen could be seen at the center mark in the mirrors. The riders were made to look at the right mirror with the right eye and at the left mirror with the left eye.

Data Processing

(1) Eye location measurements

The 35 mm films taken during the test were read by a film analyzer with regard to the lateral difference (Y coordinate) between eye location when a rider faced forward and when he looked at the mirror.

Table 2. Locations of test vehicle reference points and rear-view mirror specification.

Type \ Item		Related Riding Position			Related Rear-View Mirror	
		R-Point	G-Point	F-Point	M-Point	Mirror Specification
1. European type • Fairing Mount (R′-Point Height from G/L: 800)	X	0	−743	−135	−996	Vertical Width: 104
	Y	0	±260	±243	±295	Lateral Width: 125
	Z	+60	+160	−440	+325	R: 1200
2. European Type • Handlebar Mount (R′-Point Height from G/L: 800)	X	0	−743	−135	−756	Ditto
	Y	0	±260	±243	±315	
	Z	+60	+160	−440	+320	
3. American Type • Handleber Mount (R′-Point Height from G/L: 820)	X	0	−495	−282	−532	Ditto
	Y	0	±330	±290	±380	
	Z	+60	+266	−453	−503	

Note: European type • fairing mount test vehicle was used in Mirror-aiming measurment, arm contour measurement and computer simulation. European type • fairing mount test vehicle and American type test vehicles were used in arm contour measurement and computer simulation. Unit (mm)

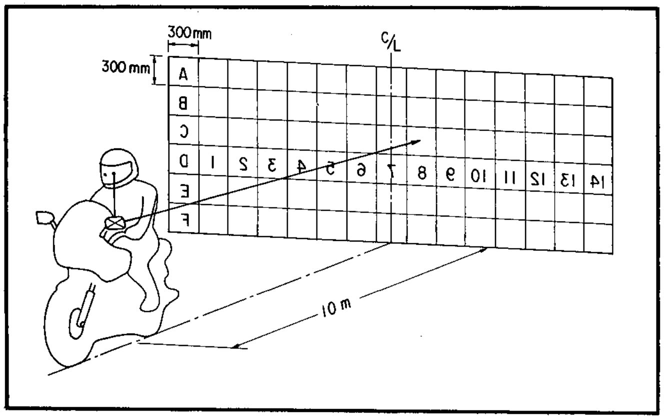


Figure 7. Measurement method of rear-view mirror aiming

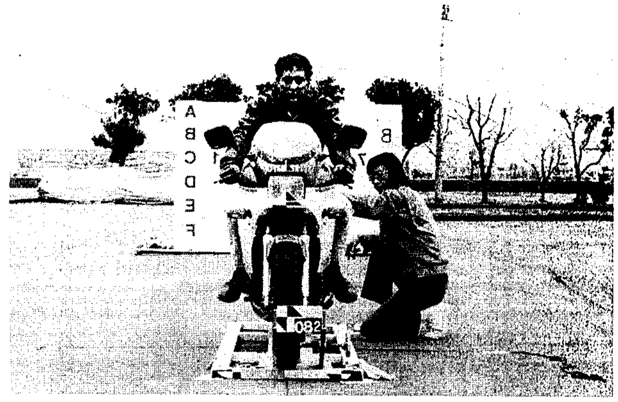


Figure 9. Measurement of eye locations (looking forward)

(2) Measurement of rear-view mirror center position

The average value for mirror center position of right and left rear-view mirrors adjusted by the riders are calculated.

(3) Measuring rear-view mirror aiming

The distribution of square positions on the screen as reported by the riders and the median of the squares was examined.

Results and Consideration

Eye Location Differences. The mean value of the lateral difference (Y coordinate) between eye location when a rider faced forward and when he looked at the mirror was established for the 26 riders. A comparison showed that the difference in case of European type test motorcycle with fairing mounted rear-view mirrors was 15 mm and 35 mm for the riders' own motorcycles. This discrepancy was due to the difference between the mirror locations of European type test motorcycle and the rider's own motorcycles.

Rear-View Mirror Center Position. The average value for mirror center position of the right and left rear-view mirrors adjusted by the riders are given below. On the European type motorcycle with fairing mounted rear-view mirrors the center position was 295 mm outside of vehicle center line and 1125 mm high. On the riders' own motorcycles it was 345 mm outside of vehicle center line and 1241 mm high.

Rear-View Mirror Aiming

(1) The distribution of rear-view mirror aiming and median value

The distribution of rear-view mirror aiming and median for the different test mirrors as reported by the riders are shown in Figures 11, 12, 13 and 14.

The distribution of rear-view aiming showed that the lateral deviation was greater than the vertical deviation, as the more than 3.6 meter wide distribution on the screen 10 meters to the rear testified.

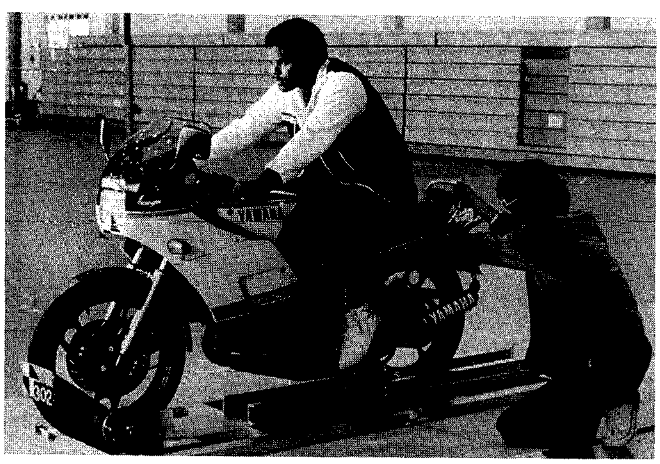


Figure 8. Test motorcycle (European type, fairing mount) with fixing jig

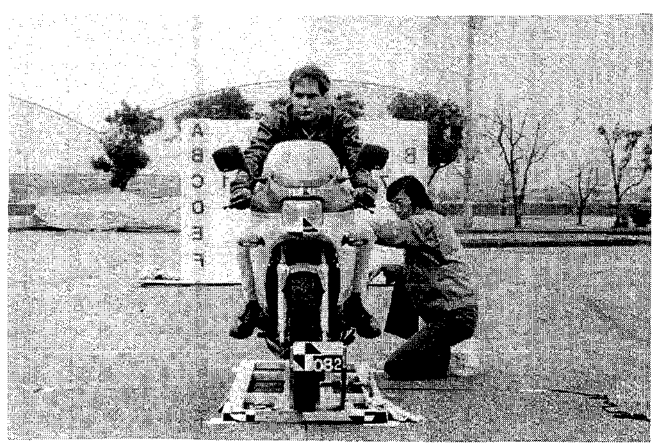


Figure 10. Measurement of eye locations (looking at right-side mirror)

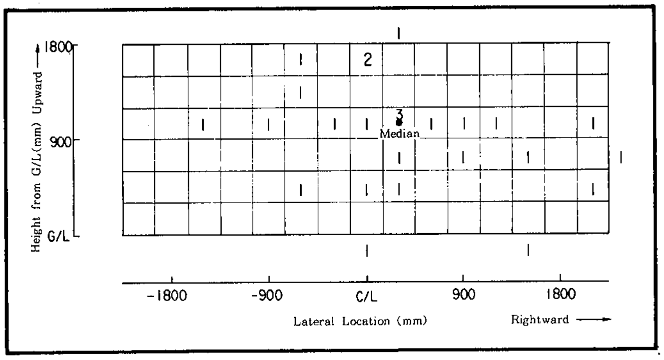


Figure 11. Distribution of rear-view mirror aiming (European type fairing mount, right-side mirror)

The medians of rear-view mirror aiming indicated that for a European type motorcycle with fairing mounted rear-view mirrors it was straight behind the rear-view mirror (in case of right mirror) and at a point between one straight behind the mirror and the vehicle center line (in case of left mirror). For the riders' own motorcycles, it was at a point between one straight behind the mirror and the vehicle center line (in case of right mirror), and in the vicinity of the vehicle center line (in the case of left mirror).

The above shows that the lateral distribution in aiming position of any type of rear-view mirror adjusted by a rider is quite great and the representative value (median) is straight behind the rear-view mirrors or in the vicinity of the vehicle center line.

(2) The variation in the rear-view mirror aiming depending on change in eye location

Fig. 15 indicates the variation in rear-view mirror aiming (median) depending on changes in eye-location.

This study shows actual measured values established from eye locations when looking at the rear-view mirror and an estimate of rear-view mirror aiming based on reference eyepoint (according to JASO 005 eye-points when facing forward) for European type motorcycles with fairing mounted rear-view mirrors.

Since the lateral difference between eye locations when the rider is looking straight ahead and when he is looking at the mirror is 15 mm, the rear-view aiming from reference eye-point varies by 210 mm towards the outside even when the angle of the rear-view mirrors is the same. This result indicates that rear-view mirror aiming position from reference eye-point lies straight behind the rear-view mirrors or slightly to the outside for European type motorcycles with fairing mounted rear-view mirrors (see Figure 15).

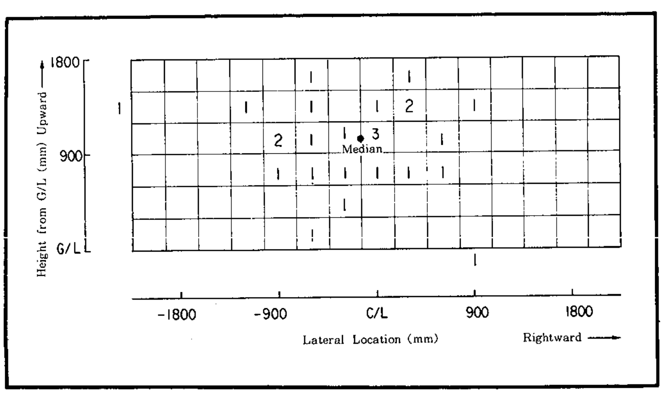


Figure 12. Distribution of rear-view mirror aiming (European type fairing mount, left-side mirror)

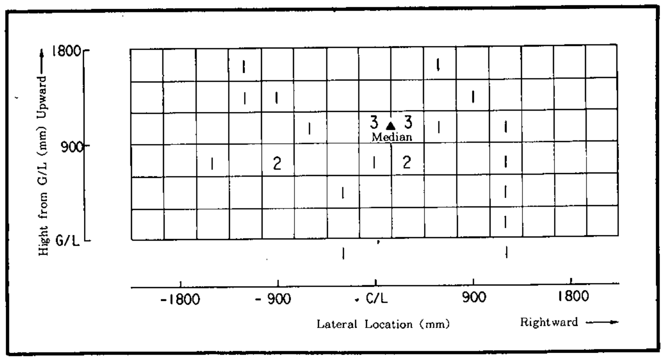


Figure 13. Distribution of rear-view mirror aiming (rider's own motorcycle, right-side mirror)

Eye-Point Selection and Adjustment of Rear-View Mirror Aiming in Measuring Rearward Field of View. It found that it was easier and more useful to rely on reference eye-point as a reference point for measuring rearward field of view than to use rider eye location when looking at the mirror—a location that varied with rear-view mirror mounting position.

Consequently, the following methods of selecting eye-point and adjusting rear-view mirror aiming for measuring rearward field of view was regarded as suitable.

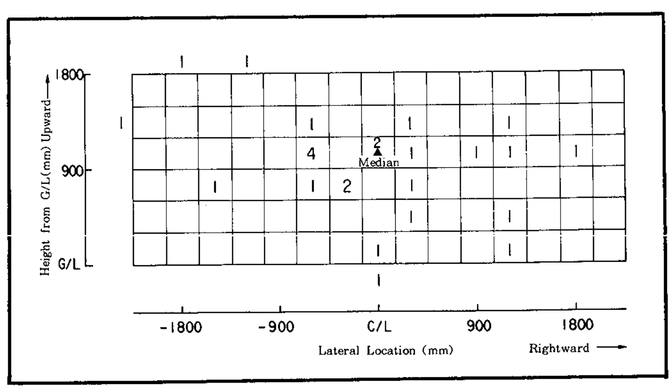


Figure 14. Distribution of rear-view mirror aiming (rider's own motorcycle, left-side mirror)

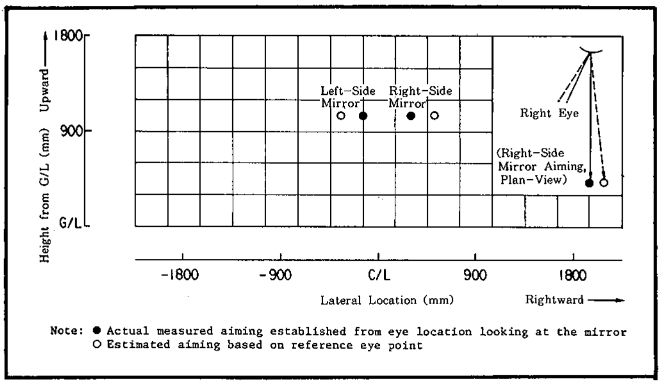


Figure 15. Variation in rear-view mirror aiming depending on eye location change

(1) The reference eye-points for the binocular condition prescribed by JASO T 005[3] was used.

(2) In case of rear-view mirror aiming, the angle of the mirrors was adjusted so that a light beam from reference eye-point towards the center of the mirror would be reflected in a parallel and horizontal manner to vehicle center line. Adjustments were made by using a light beam for left eye point for the left mirror and one coming from the right for the right mirror.

Measurement of Arm Contours of American Riders

Purpose

To obtain arm contours of American riders seated on a motorcycle mock-up by measuring their shoulder configuration.

Arm line, a straight line formed by connecting the extreme outer positions of the shoulder and the wrist positions obtained in anthropometric measurements was compared with actual arm contour and checked for agreement. Arm line was a factor introduced to simplify the test methods.

Method

Motorcycle Mock-Up. The hand grips on the motorcycle mock-up used permitted both lateral and longitudinal adjustments. Both European and American type motorcycle mock-ups were employed (see Table 2). Measurements on the European mock-up were performed under standard conditions and also with the position of the hand grips modified 35 mm inwards, outwards, forwards and rearwards.

Mock-Up and Camera Position. The camera was positioned on a line (right side) going through R'-point which was orthogonal to the vehicle center plane and a vertical line (top) going through R'-point. The distance between R'-point and the camera was 8 meters.

Test Rider. 26 male American were used as test riders (as in anthropometric measurement).

Measurement Procedures. After the anthropometric measurements, the riders were seated in a position close to actual riding position (seated and both arms stretched forwards horizontally) and the right acromion, the right radiale and the right ulna stylion were located manually and marked. Photographs were taken of the riders seated on the mock-up in a position that corresponded to a natural riding position with the side camera and front camera synchronized (see Figures 16, 17, 18 and 19).

Data Processing. The 35 mm films taken during the test were analyzed for the following data. On the side view film (X and Z coordinates) the positions of the marks could be directly analyzed and on the plan view

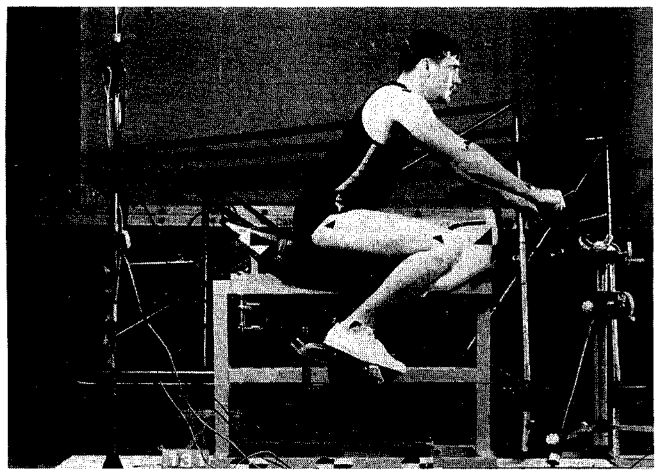


Figure 16. Motorcycle mock-up (European type) and riding position (side view)

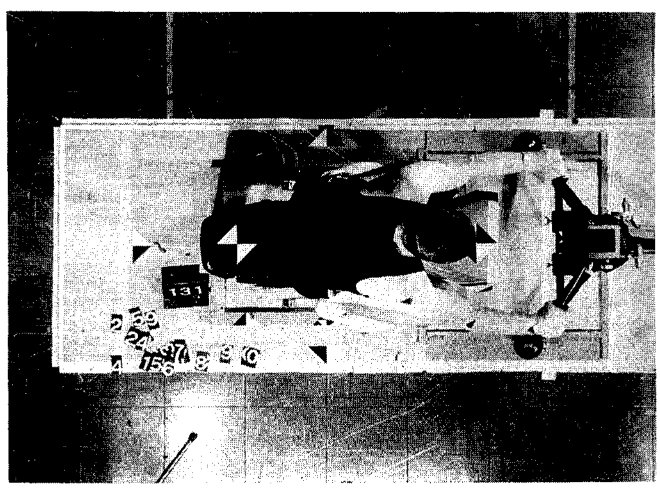


Figure 17. Motorcycle mock-up (European type) and riding position (plan view)

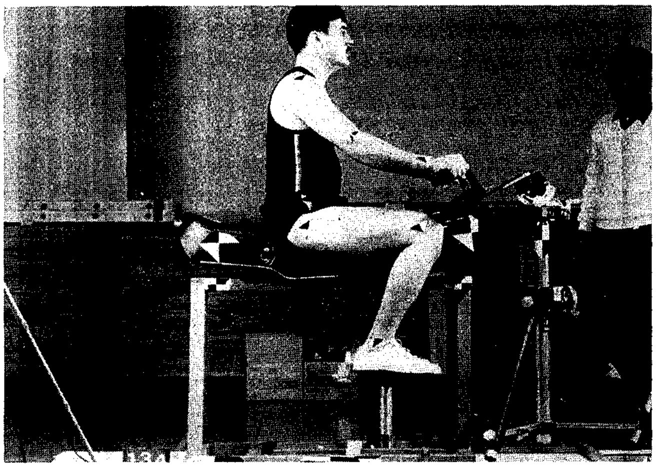


Figure 18. Motorcycle mock-up (American type) and riding position (side view)

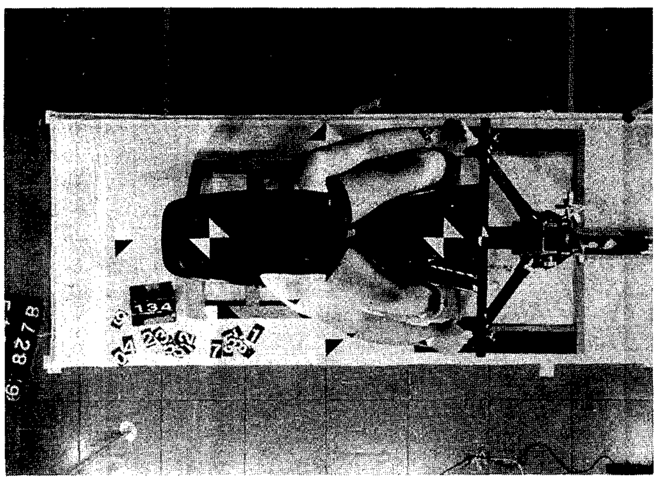


Figure 19. Motorcycle mock-up (American type) and riding position (plan view)

film (X and Y coordinates) the position of the points to be analyzed (the intersection of the outer line of the arm and straight lines parallel to the Y-axis that went through the positions of the mark; see Figure 20) could be found.

(1) Side view film
 Position of mark on right shoulder (X and Z coordinates)
 Position of mark on right elbow (X and Z coordinates)
 Position of mark on right wrist (X and Z coordinates
 Position of mark on test vehicie (X and Z coordinates)

(2) Plan view film
 Position to be analyzed on right shoulder (Y coordinate)
 Position to be analyzed on right elbow (Y coordinate)

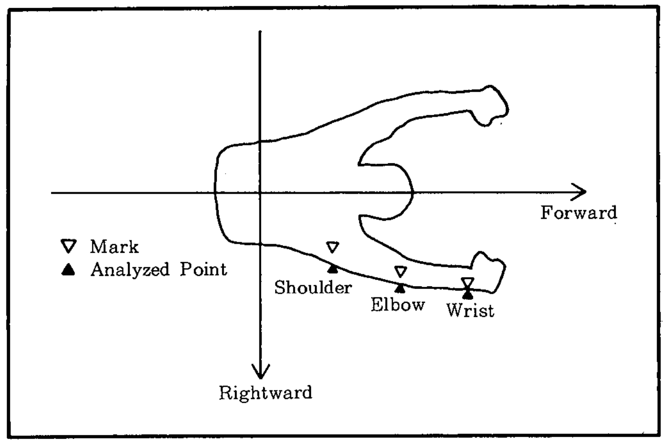


Figure 20. Marks and analyzed points

 Position to be analyzed on right wrist (Y coordinate)
 Position of marks on test vehicle (Y coordinate)

These data were fed into a computer to obtain the extreme outer positions of shoulders, elbows and wrists.

Results and Consideration

The line connecting the extreme outer position of the shoulder, the elbow and the wrist as actually measured was regarded as arm contour and the straight line connecting S′-point and G′-point (positions estimated in anthropometric measurements) was regarded as arm line (see Figure 21). Both sets of points were compared and checked for agreement.

Positions in longitudinal direction and in vertical direction (X and Z coordinates) of S′-point and G′-point correspond to shoulder point (S-point) and center of effective upper hand grip (G-point) as regulated by JASO T 005[3]. The distance between right and left S′-point was 500 mm and the lateral position of G′-point was 45 mm outside of G-point. These distances are larger than the mean value of rider biacromial width (412 mm) and half of the mean value for palm width (see Table 1). The extreme outer positions of the shoulder, the elbow and the wrist obtained with the riders seated on the test motorcycles are shown in Table 3.

European Type Motorcycle. Rider arm contour on a European type motorcycle with hand grips in standard location form an almost straight line. Compared with arm line the position of the elbow is slightly lower (see Figure 22) from the side view and the position of the shoulder from the plan and front views is slightly towards the inside (Figures 23 and 24), but other positions correspond quite closely.

Even when the position of the hand grips were modified laterally or longitudinally, rider arm contour

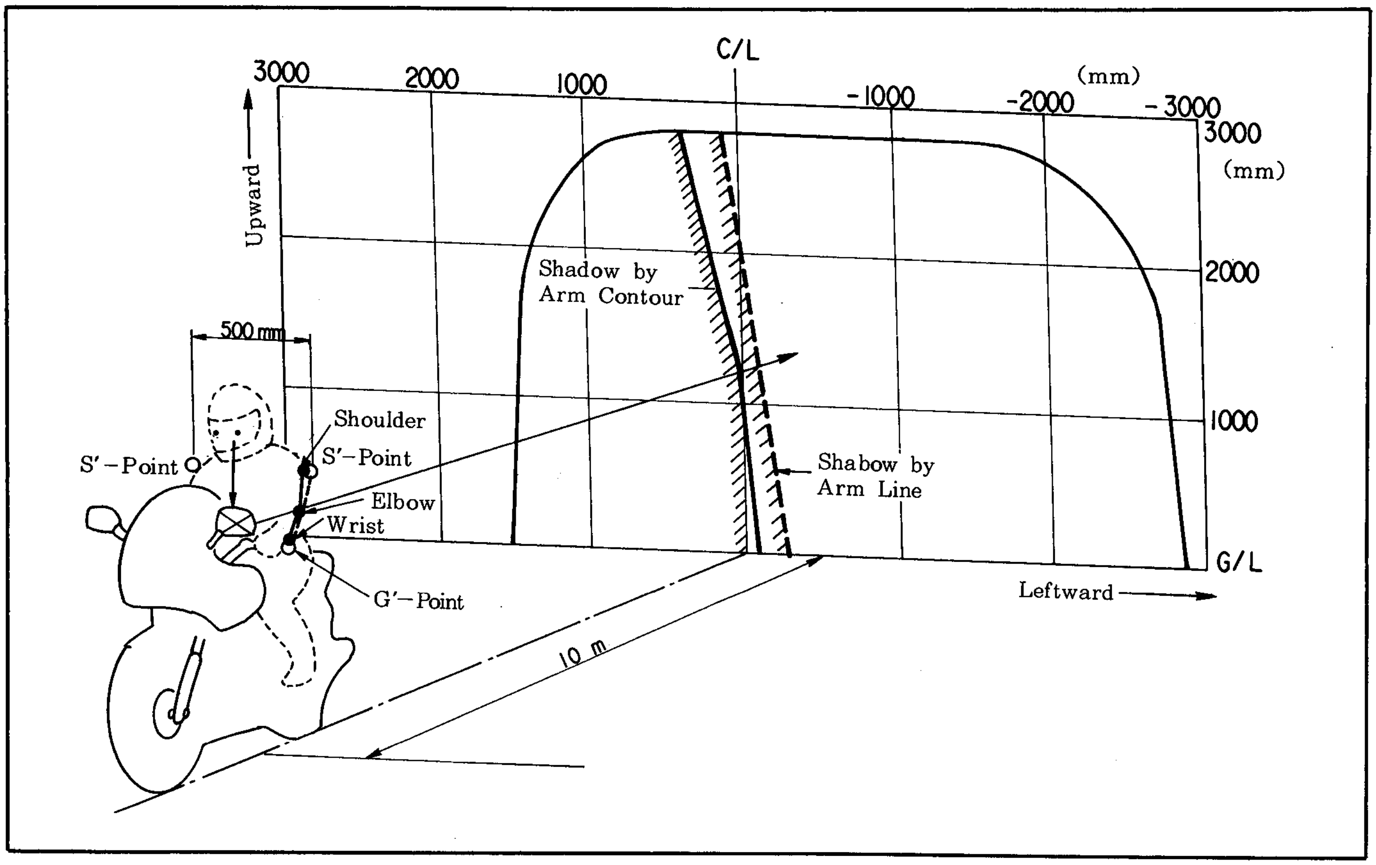


Figure 21. Evaluation method of field of view of rear-view mirror

still formed an almost straight line and compared with arm line the position of the shoulder was slightly inwards, but other positions corresponded quite closely (Figures 25, 26, 27 and 28).

Table 3. Arm contour actual measured (extreme outer positions of shoulder, elbow and wrist).

Test Vehicle / Hand Grip Location / Measured Item		European Type					American Type
		Standard	Outward 35mm	Inward 35mm	Forward 35mm	Rearward 35mm	Standard
Shoulder	X	− 240	− 245 (− 5)	− 236 (+ 4)	− 268 (− 28)	− 216 (+ 24)	26
	Y	224	226 (+ 2)	222 (− 2)	226 (+ 2)	223 (− 1)	217
	Z	530	528 (− 2)	527 (− 3)	517 (− 13)	539 (+ 9)	556
Elbow	X	− 461	− 461 (0)	− 457 (+ 4)	− 491 (− 30)	− 435 (+ 26)	− 178
	Y	278	295 (+ 17)	262 (− 16)	280 (+ 2)	276 (− 2)	315
	Z	311	313 (+ 2)	304 (− 7)	304 (− 7)	316 (+ 5)	339
Wrist	X	− 680	− 679 (+ 1)	− 681 (− 1)	− 715 (− 35)	− 653 (+ 27)	− 414
	Y	294	326 (+ 32)	264 (− 30)	295 (+ 1)	295 (+ 1)	351
	Z	189	192 (+ 3)	188 (− 1)	190 (+ 1)	192 (+ 3)	277

Note : Numerals in thick lined frames mean standard location of hand grip. Unit (mm)
Numerals in ( ) show deviation from standard location of hand grip.
On X−axis, "+" means rearward deviation
On Y−axis, "+" means outward deviation
On Z−axis, "+" means upward deviation

The positions of the shoulder, the elbow and the wrist when the hand grips had been modified laterally or longitudinally and the difference between these positions and standard positions are shown in brackets in Table 3.

When the hand grip position was modified 35 mm to the right or to the left, shoulder position hardly changed, whereas elbow position changed 16-17 mm

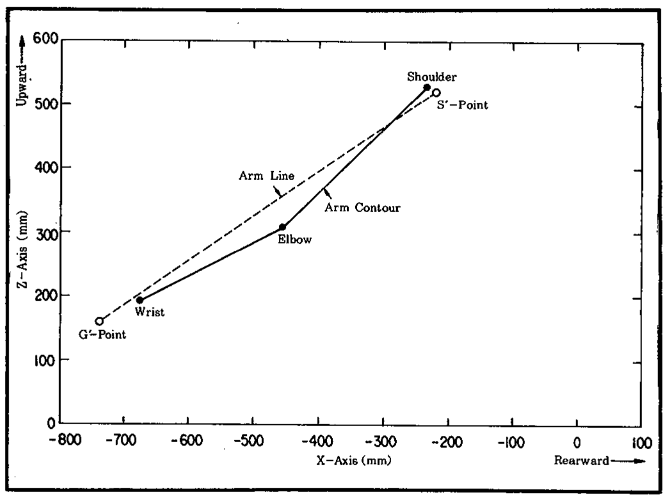


Figure 22. Arm contour and arm line (European type, side view)

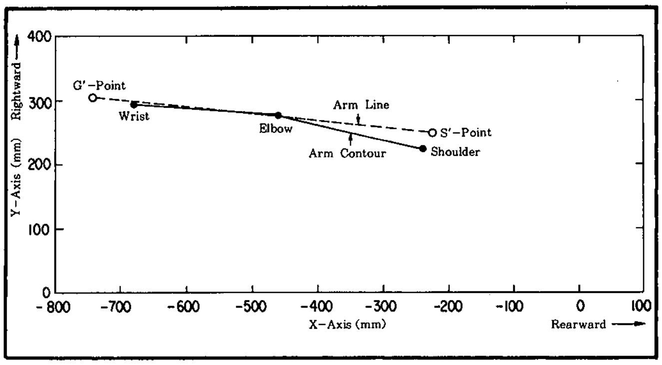


Figure 23. Arm contour and arm line (European type, plan view)

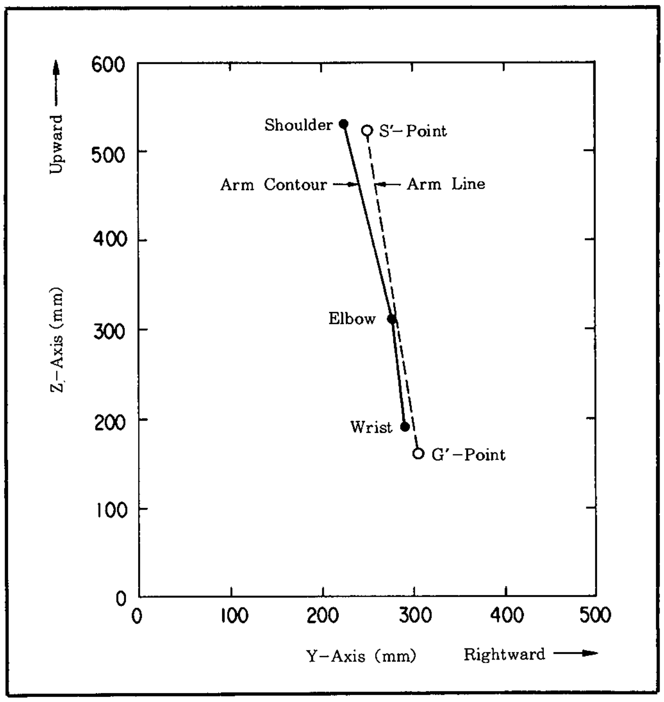


Figure 24. Arm contour and arm line (European type, front view)

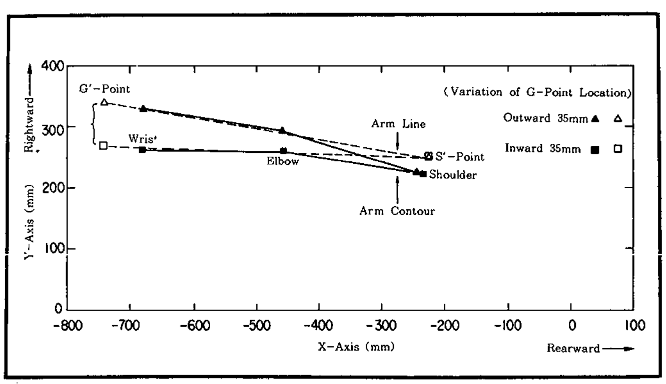


Figure 25. Arm contour and arm line variation according to variation in hand grip location in lateral direction (plan view)

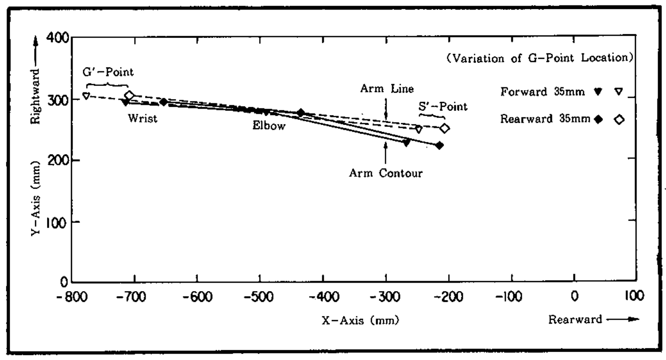


Figure 26. Arm contour and arm line variation according to variation in hand grip location in longitudinal direction (plan view)

and wrist position changed 30-32 mm to the right or left. When the hand grip position was modified 35 mm to the rear or front, shoulder, elbow and wrist positions changed 24-28 mm, 26-30 mm and 27-35 mm respectively to the rear or the front.

American Type Motorcycle. Rider arm contour on an American type motorcycle with hand grips in standard location form a slightly bended line from the side view and plan view, but is an almost straight line from the front view. Compared with arm line elbow and wrist positions are slightly lower (Figure 29) from the side view and shoulder position from the plan view and front views is towards the inside (Figures 30 and 31). Elbow position from the plan view is towards the outside, but is towards the inside from the front

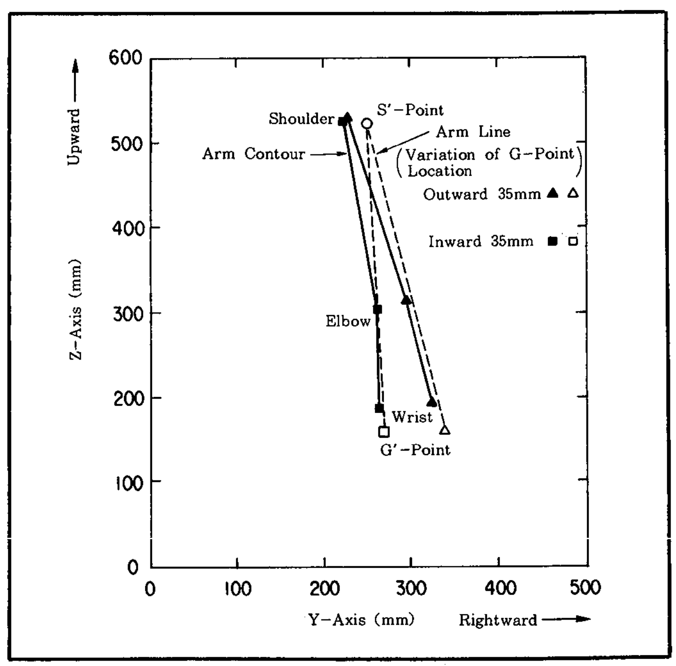


Figure 27. Arm contour and arm line variation according to variation in hand grip location in lateral direction (front view)

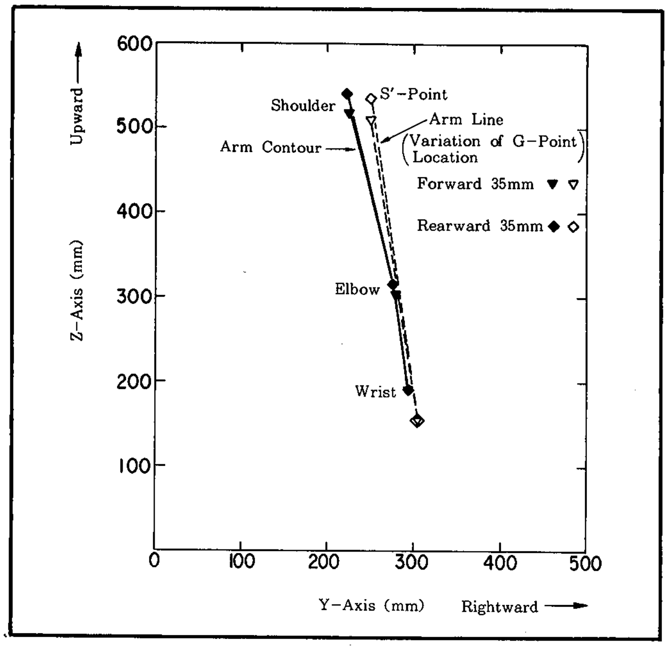


Figure 28. Arm contour and arm line variation according to variation in hand grip location in longitudinal direction (front view)

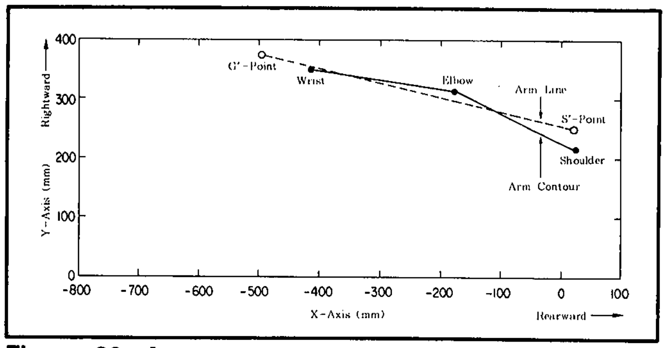


Figure 30. Arm contour and arm line (American type, plan view)

view. This is because vertical data from the plan view and longitudinal data from the front view are not included.

Consequently, in order to properly evaluate the effect of rider arm contour and arm line on the rearward field of view, it is necessary to simulate or measure the rearward field of view using three-dimensional positions for arm contour and arm line.

Computer Simulation of Motorcycle Rearward Field of View

Purpose

Three-dimensional positions of actual measurements of arm contour and estimated values of arm line were used in a simulation of rearward field of view to compare arm contour shadow and arm line shadow. The arm line factor was used to simplify test methods.

The relationship between the rear-view mirror aiming, eye-point and rearward field of view was also examined.

Method

A CAD based simulated program was used with three types of test motorcycles with rear-view mirrors mounted on the fairing or the handlebars. This corresponds to the method of evaluating motorcycle rearward field of view indicated in Figure 21.

Computation Flow Chart. The computation flow chart (items executed and items input) is shown in Figure 32. Arm contour and arm line was evaluated by the shadow (especially the value at ground level 10 meters to the rear) they projected on a screen placed 10 meters behind eye-point.

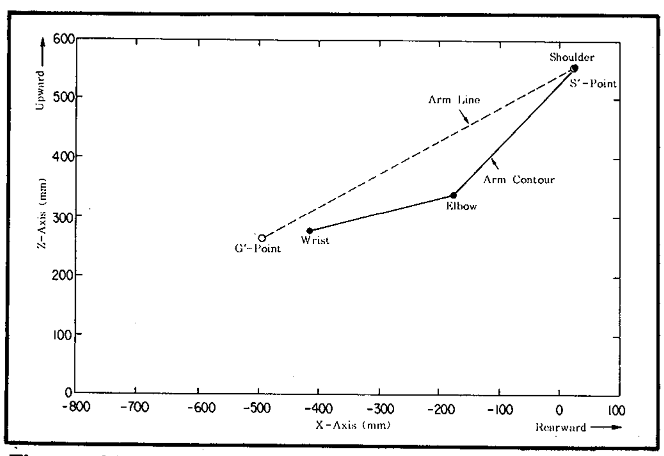


Figure 29. Arm contour and arm line (American type, side view)

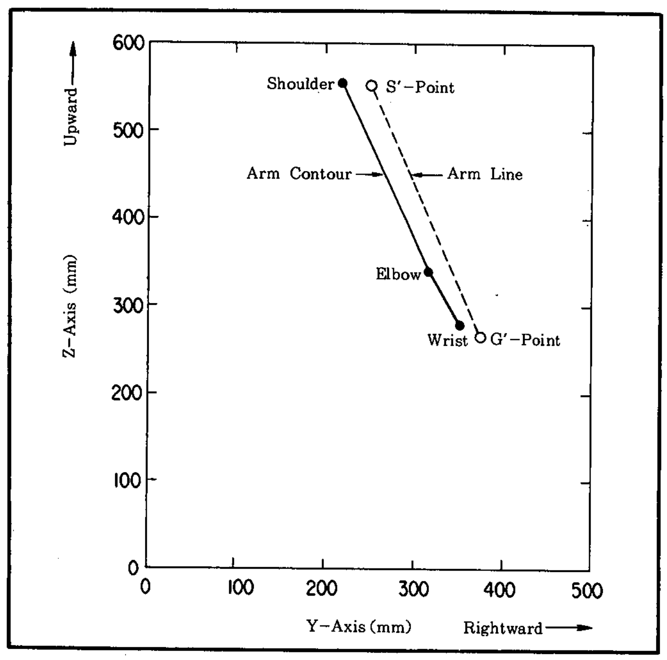


Figure 31. Arm contour and arm line (American type, front view)

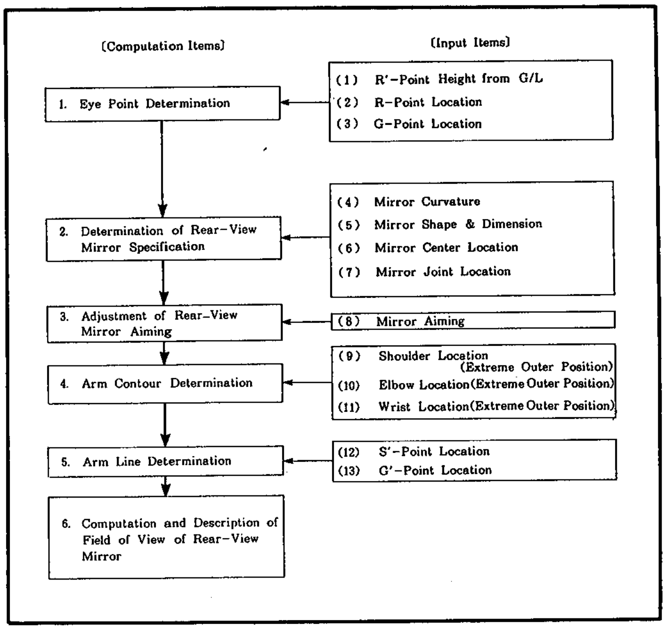


Figure 32. **Computation procedure of field of view of rear-view mirror for motorcycle**

Eye-Point Selection. The reference eye-points for the binocular condition prescribed by JASO T 005[3] have been used. Although eye position shifts 15-35 mm towards the mirror when a rider looks in the rear-view mirror, procedures were simplified by employing a reference eye-points which did not account for head and eye movements.

Rear-View Mirror Aiming Methods. The adjustment method employed corresponded to measurement results of rear-view aiming position. In case of reference eye-point of binocular conditions the angle of the mirrors was adjusted so that a light beam towards the center of the mirror (M-point) would be reflected in a parallel and horizontal manner to vehicle center. Adjustments were made by using a light beam from left eye position for the left mirror and one coming from the right for the right mirror.

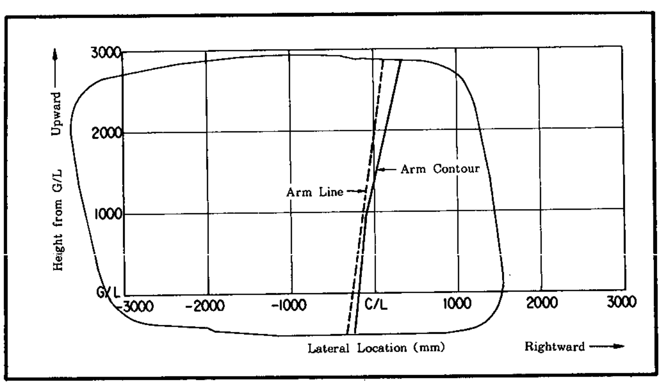


Figure 33. **Field of view of left rear-view mirror mounted on European type motorcycle fairing (G-point: standard location)**

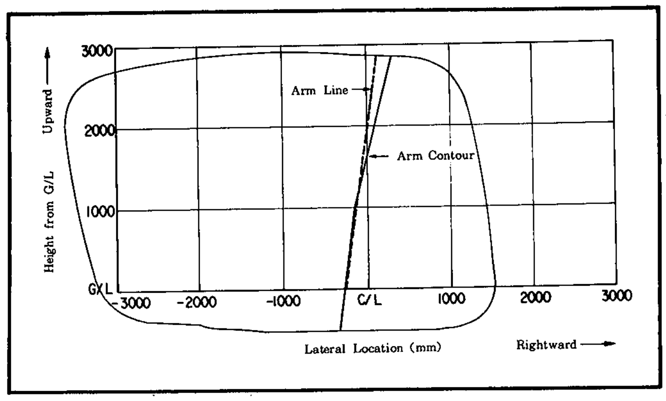


Figure 34. **Field of view of left side rear-view mirror mounted on European type motorcycle fairing (G-point: standard location, wearing a 5 mm thick motorcycle suit)**

Test Motorcycles. Three test motorcycles were used: two types of European motorcycles (one with fairing mounted and the other with handlebar mounted rear-view mirrors) and one American type motorcycle (see Table 2).

Test Conditions. The rear-view mirrors and the hand grips of the European type motorcycle with handlebar mounted rear-view mirrors and the American type motorcycle were in standard location (Table 2) during the test. The European type motorcycle with fairing mounted rear-view mirrors was tested with rear-view mirrors and hand grips in standard locations. It was also tested with the position of the hand grips modified 35 mm towards the outside, the inside, to the front and rear. When the position of the hand grips was modified, the position of the rear-view mirrors was also changed so that the relative position of the rear-view mirrors and the hand grips remained the same. Thus when the hand grips were moved 35 mm outwards the mirrors were also moved 35 mm outwards.

Results and Consideration

The left rearward field of view (binocular condition) from reference eye-points (binocular condition)

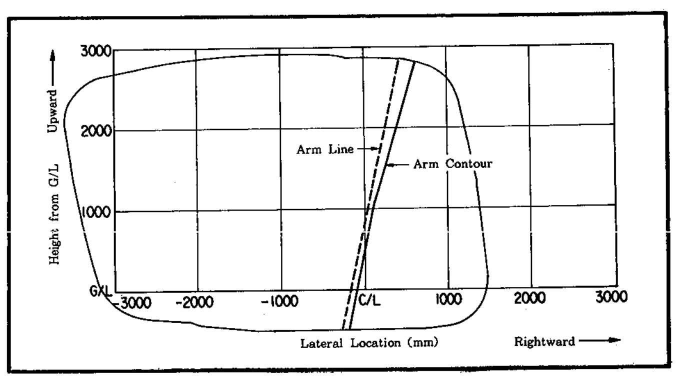


Figure 35. **Field of view of left side rear-view mirror (G-point: outward 35 mm)**

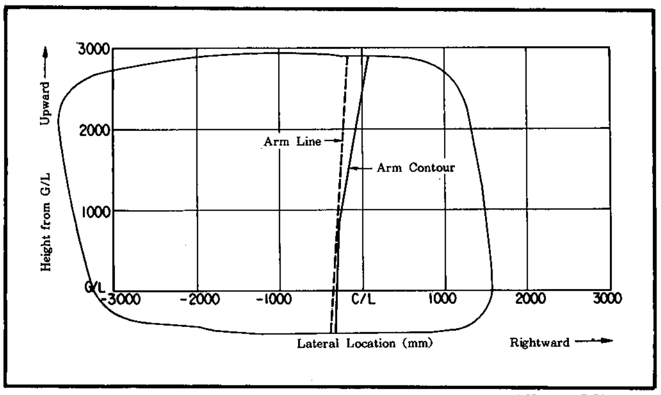


Figure 36. Field of view of left side rear-view mirror (G-point: inward 35 mm)

for the respective conditions is shown in Figures 33-42. The Figures show projections of the left rearward field of view (binocular condition) 10 meters behind reference eye-point. The shadow caused by arm contour and arm line is the shadow seen by the left eye.

Comparisons of Shadows Caused by Arm Contour and Arm Line.

(1) European type motorcycle with fairing mounted rear-view mirrors

When the hand grips were in standard location on the European motorcycle with fairing mounted rear-view mirrors, the arm contour shadow was slightly inside the arm line shadow (about 70 mm at ground surface 10 meters to the rear) (Figure 33). When the arm contour (shoulder, elbow and wrist position) was modified 5 mm (e.g. by wearing a 5 mm thick motorcycle suit), the shadow from the elbow and downwards, corresponded to arm line (Figure 34). When the hand grip position was modified laterally or longitudinally, arm contour was slightly inside arm line (40-70 mm at ground surface 10 meters to the rear) (see Figures 35, 36, 37 and 38).

(2) European type motorcycle with handle bar mounted rear-view mirrors

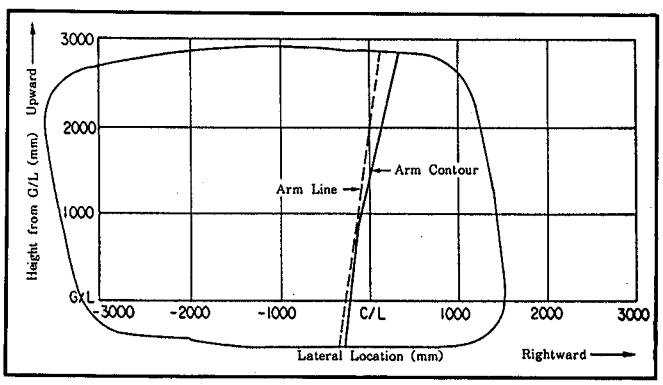


Figure 37. Field of view of left side rear-view mirror (G-point: forward 35 mm)

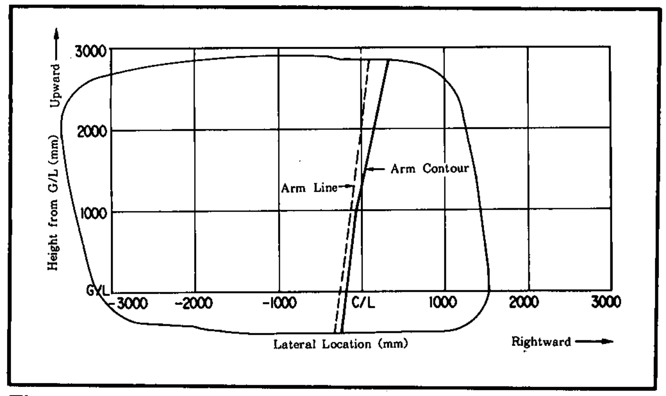


Figure 38. Field of view of left side rear-view mirror (G-point: rearward 35 mm)

Arm contour for European motorcycles with handle bar mounted rear-view mirrors was also slightly inside arm line (60 mm at ground level 10 meters to the rear, Figure 39). The Shadow area caused by arm contour and arm line was small and at ground surface 10 meters to the rear the vehicle center line could easily be seen.

(3) American type motorcycles

On American type motorcycles, the arm contour is rather to the inside of arm line (there is a 270 mm difference at ground surface 10 meters to the rear, see Figure 40). However, the shadow area produced by arm contour and arm line is negligible. At ground surface 10 meters to the rear it is 1 meter inside vehicle center line.

Arm Line Used as a Method of Measuring Rearward Field of View. Riders on European type motorcycles with handlebar mounted rear-view mirrors and American type motorcycles could easily see the ground surface vehicle center line 10 meters to the rear. Thus practically there was no reason to take the influence of rider arm shadow into account.

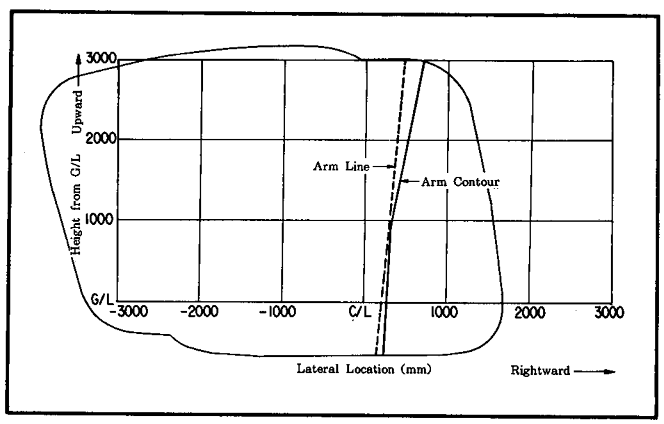


Figure 39. Field of view of left side rear-view mirror mounted on European type motorcycle handle bar

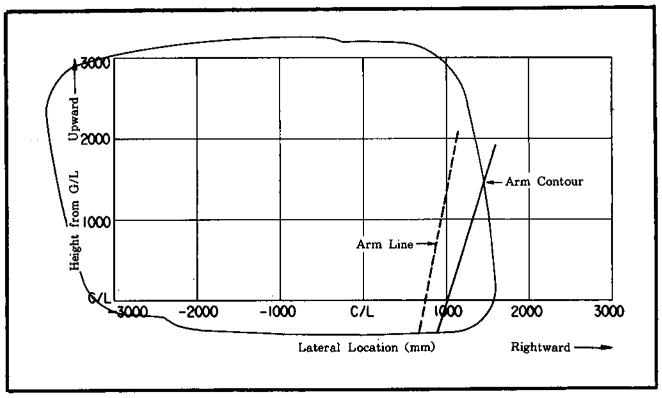


Figure 40. Field of view of left side rear-view mirror mounted on American type motorcycle handle bar

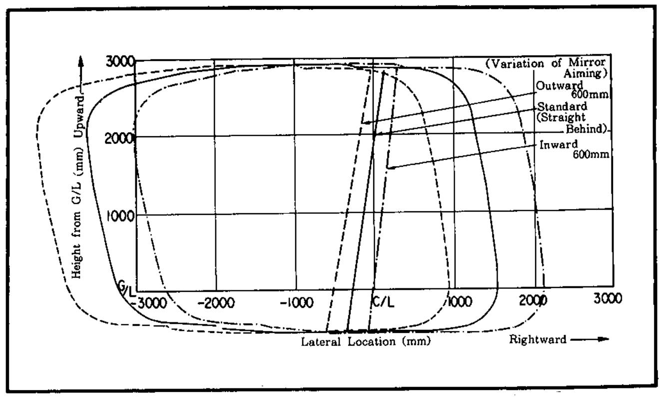


Figure 41. Field of view of left side rear-view mirror variation according to variation in mirror aiming

In case of European type motorcycles with fairing mounted rear-view mirrors where rider arm shadow had to be considered, rider arm shadow (wearing a motorcycle suit) and arm line was regarded as identical.

Consequently, to evaluate rider arm shadow in rearward field of view measurements, it is advisable to make a simplification by using the line formed by connecting S′-point and G′-point (arm line).

Variations in Rearward Field of View as Dependent on Variations in Rear-View Mirror Aiming and Eye-Point. The variations (arm line influence) in rearward field of view for European motorcycles with fairing mounted rear-view mirrors are shown in Figures 41 and 42.

(1) Variations in rearward field of view as dependent on variations in rearview mirror aiming (Figure 41)

When the rear-view mirrors were aimed 600 mm outwards, the shadow of the arm line moved 240 mm outwards as compared to when they were aimed straight behind (standard condition). When they were aimed 600 mm inwards, the arm shadow moved 240 mm inwards.

(2) Variation in rearward field of view as dependent on variations in eye-point location (Figure 42)

The following relationship prevailed between reference eye-point (the eye-point when the rider looks straight forward; as determined by JASO T 005[3]) and rearward field of view. When eye-point location shifted outwards, the rear-view aiming position (point of projection of M-point) also shifted inwards. Even when the mirror angle remained the same, a 10 mm shift in eye-point position produced a 140 mm shift in aiming position. The 15 mm eye movement required to look into the rear-view mirrors of this type of motorcycle produces a 210 mm shift in aiming position.

When eye-point location is shifted outwards, the shadow caused by arm line is also moved inwards. A 10 mm shift in eye-point location produced a 65 mm change in arm line shadow. The 15 mm eye movement required to look into the rear-view mirrors of this type of motorcycle produces a 95 mm shift in arm line shadow. The inside field of view from the actual eye position of a rider looking at the rear-view mirrors appeared 95 mm inwards as compared with the inside field of view from reference eye-point.

Consequently, an evaluation of inside field of view based on reference eye-point and arm line indicates the inside field of view actually seen by riders wearing 12 mm thick motorcycle suits. (The arm contour margin for arm line is 70 mm and the margin for eye-point variation is 95 mm i.e. a total of 165 mm; this corresponds to the effect a 12 mm variation in arm line towards the outside has on the inside field of view).

Summary

In this study to establish measurement method of rearward field of view, we measured rear-view mirror

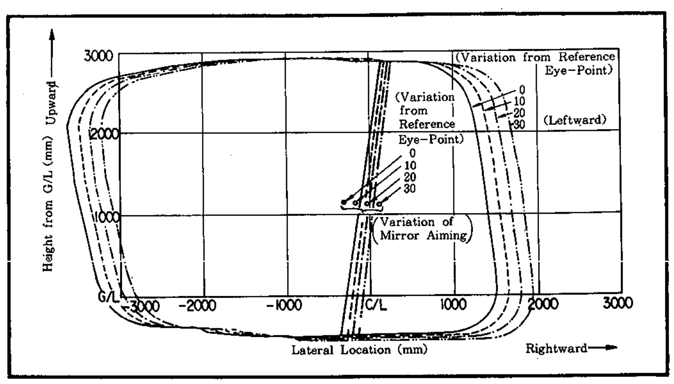


Figure 42. Field of view of left side rear-view mirror variation according to variation in eye location (leftward)

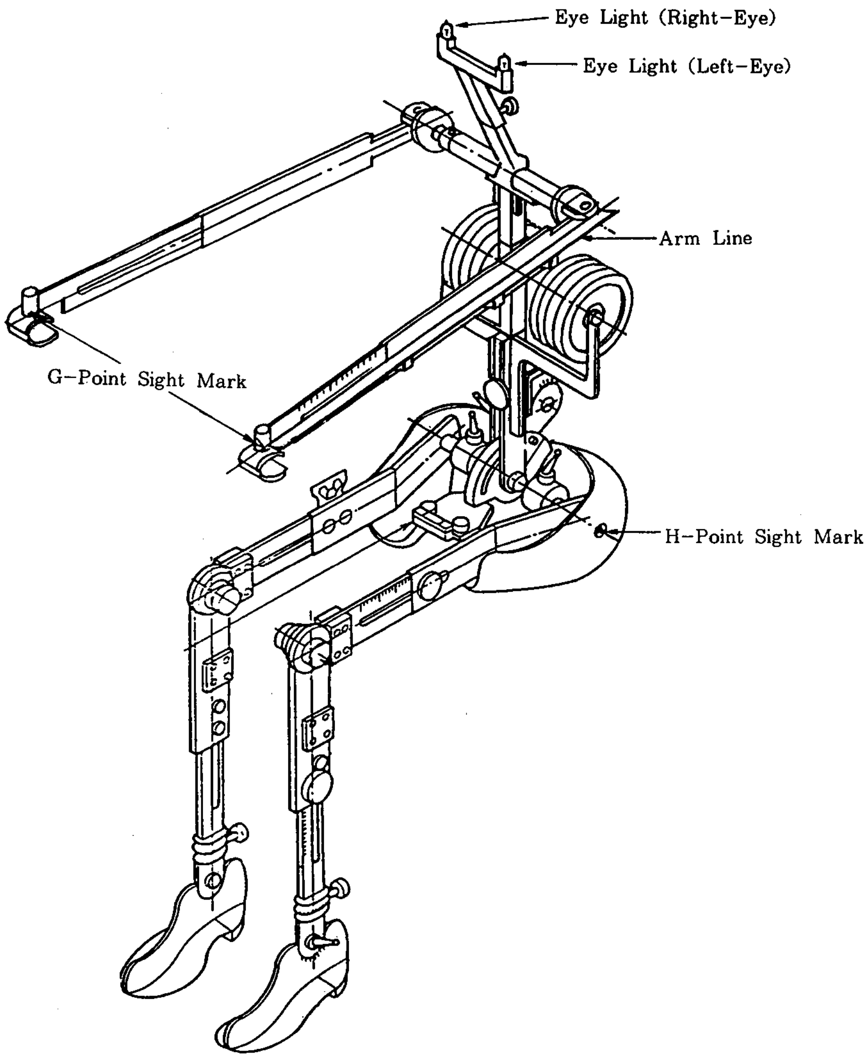


Figure 43. Three-dimensional manikin for motorcycle (with eye light and arm line)

aiming and rider arm contour, and examined those results by means of computer simulation for evaluating the rearward field of view.

In order to sufficiently evaluate design and performance factors of motorcycle rearward field of view, a quantitative study of the required field of view of motorcycle rear-view mirrors will have to be made.

Eye-Point Selection

We found it suitable to use the eye-point value prescribed in JASO T 005[3] as reference eye-point in field of view measurements. Because it is easier and more useful to rely on reference eye point than to use rider eye location when looking at the mirror—a location that varies with rear-view mirror location, and the difference between the rearward field of view from reference eye point and that from rider eye location can be compensated by motorcycle suit thickness.

Rear-View Mirror Aim

It was established that the most suitable angle of the rear-view mirrors when measuring the rearward field of motorcycle rear-view mirrors was when a light beam travelling from reference eye-point (binocular condition) to the mirror center would be reflected horizontal and in a parallel manner in relation to vehicle center line. The left rear-view mirror was adjusted with the aid of a light beam from the left eye and the right rear-view mirror with one from the right eye.

Rider Arm Contour

It seemed most appropriate to simplify the test method by using arm line, a line created by connecting S′-point and G′-point, to evaluate the effect of rider arm shadow (Figure 21). In case of European type test motorcycles with fairing mounted rear-view mirrors where the effect of rider arm shadow had to be considered, rider arm shadow (when wearing a motorcycle suit) and arm line shadow was regarded as identical.

Recommended Methods of Measuring Motorcycle Rearward Field of View

The above results were used as a basis for establishing measuring methods of rearward field of view on a motorcycle when using a three-dimensional manikin for motorcycle and making computer simulation program to examine rearward field of view at the design stage.

Methods of Measuring Rearward Field of View on a Motorcycle

The rearward field of view was evaluated as the field of view projected on a screen 10 meters to the rear of eye-point (see Figure 21). The measurement procedures were as follows:

(1) A three-dimensional motorcycle manikin was placed (Figure 43) on a motorcycle, the binocular eye-point (as stipulated by JASO T 005) was determined and an eye light device was lit.

(2) The mirrors were adjusted so that in binocular condition a light beam travelling from reference eye-point to the mirror center (M-point) would be reflected horizontally and in a parallel manner in relation to vehicle center line. The left rear-view mirror was adjusted with the aid of a light beam from the left eye and the right rear-view mirror with one from the right eye.

(3) The rearward field of view (arm line shadow) projected on the screen was analyzed.

Computer Simulation of Motorcycle Rearward Field of View at Design Stage

A CAD system was also employed in conjunction with other methods to measure the rearward field of view on motorcycles (Figure 32). This was conducted as follows.

(1) Eye-point was determined (according to JASO T 005).
(2) Rear-view mirror data were determined.
(3) Rear-View mirror angle was determined.
(4) Arm line was determined.
(5) Rearward field of view (and arm line shadow) was calculated and displayed.

Acknowledgement

The authors wish to express their appreciation to Maj. L.D. Maahs, TSgt. C.F. Carson and Mr. Noboru Horikoshi, Safety Department Yokota Air Base, U.S. Air Force for their great help; to Mr. Shiro Tomimoto, Mr. Hironao Adachi and Mr. Kazunari Ohara, Yamaha Motor Co., Ltd. for conducting the computer simulation and to all the motorcycle riders for their willing cooperation.

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Motorcycle Impact Simulation and Practical Verification

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Abstract

Experimental impact tests of a motorcycle into a flat rigid barrier at 90° to direction of travel have shown cast front wheels to have a marked pitching effect on the machine whereas wire spoked wheels do not have this effect. A mathematical model of the motorcycle and rider has been developed which can reproduce the dynamics of a motorcycle. The dynamic stiffness of front wheels has been experimentally related to static stiffness and impact velocity, and included in the simulation. The rider dynamics require further development on that part of the model.

Introduction

Motorcycle impact testing is inherently expensive and its results are not always conclusive. There is usually an inability to alter one major parameter, from test to test, without the inevitability of changing others in the process. A generalised overview of motorcycle and rider dynamics in an impact situation would require a vast number of practical tests before any reliable conclusions could be drawn. An experimental process was therefore devised whereby selected practical tests could be performed to verify a mathematical simulation of the process. This simulation can highlight the important parameters affecting the kinematics of motorcycle and rider, with sufficient accuracy for practical purposes.

Motorcycle and rider impact simulations have not, to date, taken full account of front and energy absorption characteristics. Recent impact tests at TRRL have shown that these parameters have a marked effect on rider kinematics. These tests involved identically impacting two heavy-weight motorcycles "head-on" into a massive rigid barrier. One of the machines had a cast aluminum front wheel and the other a conventional steel wire spoked wheel.

Film analysis of these impacts has shown that for the cast wheeled motorcycle the peak deceleration of the centre of gravity was nearly twice that of the wire spoked version. The rider of the former motorcycle left contact with the machine at one and a quarter times the velocity of the latter. This increase was due to the cast wheel not crushing, thereby allowing more radial movement of the headstock (pitching) about the front axle before the headstock impacted the barrier. This radial acceleration for the cast wheeled machine was twice that of the one with a wire spoked front wheel, providing an increase in radial acceleration which would catapult the rider with one and a half times the kinetic energy of the wire spoked version on leaving the machine. The extent of crushing of the two types of wheels can be seen in Figs. 1 and 2.

Motorcycle and rider impact tests have repeatedly shown that the only items on the motorcycle that absorb energy during "head-on" impact are the front wheel and forks. The optimisation of these energy absorption characteristics would greatly aid the optimisation of other safety items, such as air-bags, to retard the rider at a safer level during impact thus reducing injuries in the process.

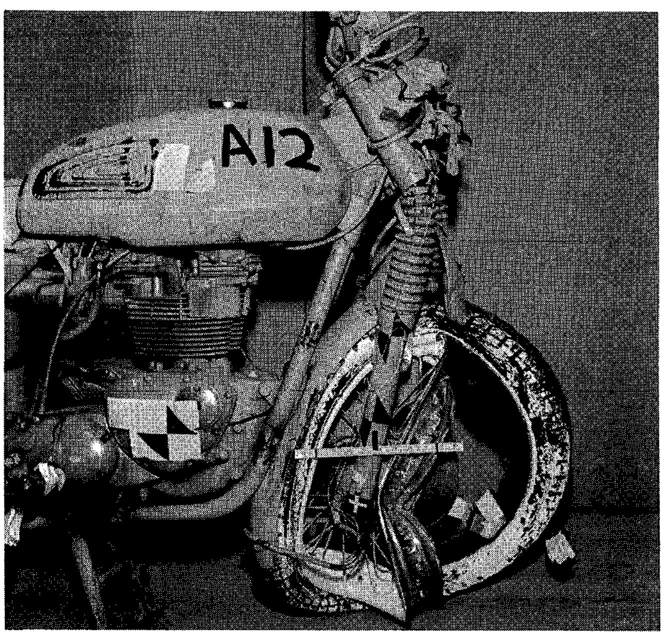

Figure 1. Heavyweight motorcycle wire spoked front wheel after a 13.4 m/s head-on impact

Figure 2. Heavyweight motorcycle cast front wheel after a 13.4 m/s head-on impact

Mathematical Simulation

Full Motorcycle

A mathematical simulation of the motorcycle and rider has been developed that is based on a three mass system, see Fig. 3. These masses are equivalent to the centres of gravity of the front wheel, the rest of the motorcycle and the rider's torso. The acceleration sustained by each of these masses was evaluated by the summation of the forces, acting in pertinent directions, in accordance with Newtonian mechanics.

The displacement of most points in the model was derived from absolute velocities, which were related to one of the main masses via translational and rota-

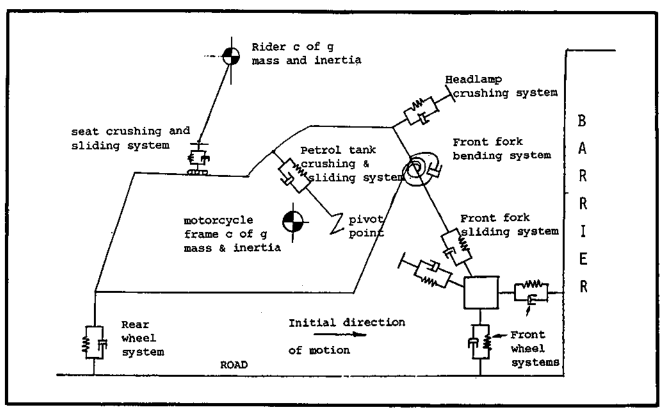

Figure 3. Diagrammatic view of mathematical simulation of a motorcycle and its rider

tional velocities. Angular velocities of two points relative to each other required triangulation via an auxiliary point, either dummy or real, and the use of the cosine rule. This method inherently provided velocities for damping forces and compression values for spring forces by integration.

The forces acting on the rider were from compression and friction against the seat and petrol tank as well as gravitational acceleration. The rider has initially been modelled as a point mass with rotational inertia at the centre of gravity of the torso with a mass-less beam connecting this mass to the rider's posterior.

The front forks were modelled as a sliding non-linear spring damper system, but also allowing for the forks to bend below the headstock. The bending of the front forks about the headstock is elastic, until the elastic limit is reached, after which the collapse is purely plastic. There is a small amount of structural damping applied to this motion.

The forces and moments acting on the main motorcycle centre of gravity were used to evaluate its translational and rotation motion. These forces were:

i. The gravitational force acting on the motorcycle.
ii. The front wheel when it impacts the lower front section of the frame.
iii. The front forks including the forces due to their rotation about the headstock.
iv. The action of the rear wheel on the road (the rear suspension system was initially considered to have a negligible affect on the global motion of the motorcycle).
v. The motion of the rider along the seat and petrol tank giving frictional and normal forces on the motorcycle. The seat and petrol tank were modelled as non-linear spring damper systems, with permanent deformation on the tank.

The inclusion of the front wheel and fork energy absorption characteristics in a simulation require an accurate knowledge of the dynamic load deflection properties of motorcycle wheels and front forks. Initially, the front forks have been modelled as an elastic-plastic system as suggested by Sherman(Ref. 1).

Front Wheel

The mathematical modelling of the front wheel was based on complete wheel spring stiffnesses obtained by static tests to destruction. An impact rig was used to obtain dynamic magnifying factors. The mathematical model is a non-linear spring with a damper in parallel. When the spring is unloaded there is an allowance for hysteresis and permanent deformation, based on the work of Fowler(Ref. 2). A more comprehensive third order system was not required as the relatively low tyre stiffness eliminated any initial velocity effects on impact, and damping levels were relatively low.

The forces acting on the front wheel mass during impact emanate from contact with rigid bodies, such as the road, the barrier or obstacle and the frame. The front forks bend at the headstock and trap the front wheel between the motorcycle frame and the barrier. The contact point of the wheel and frame will vary depending on the relative positions of the two bodies. It is assumed that the wheel stiffness characteristics are similar at all these contact points. The rotational motion of the front wheel was considered unimportant as it contained only a very small percentage of the total energy absorbed by the front wheel system but this motion may well have to be included in any three-dimensional motorcycle model.

Front Wheel Simulation Verification

A drop rig was built to test the behaviour under impact of motorcycle front wheels, front forks and possibly petrol tanks, at velocities up to 7 m/s, see Fig. 5. Characteristics for wheels have been extrapolated from a series of tests at differing velocities up to 7 m/s to give the required values for impact velocities up to 14 m/s. The rig results were in the form of energy/displacement characteristics, at selected velocities, for direct comparison with a mathematical simulation of the impact situation.

This simulation consisted of a mass falling a prescribed distance, under gravity, impacting a rigid mass, which was resting on top of, and independent of, a series pair of second order mass-spring-damper systems, see Fig. 5. The spring characteristics were

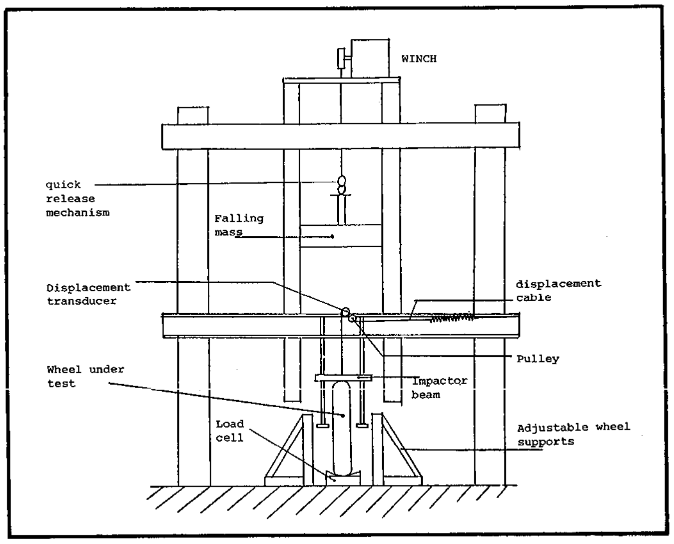

Figure 4. Diagrammatic view of the drop test rig used for dynamic analysis of motorcycle wheels

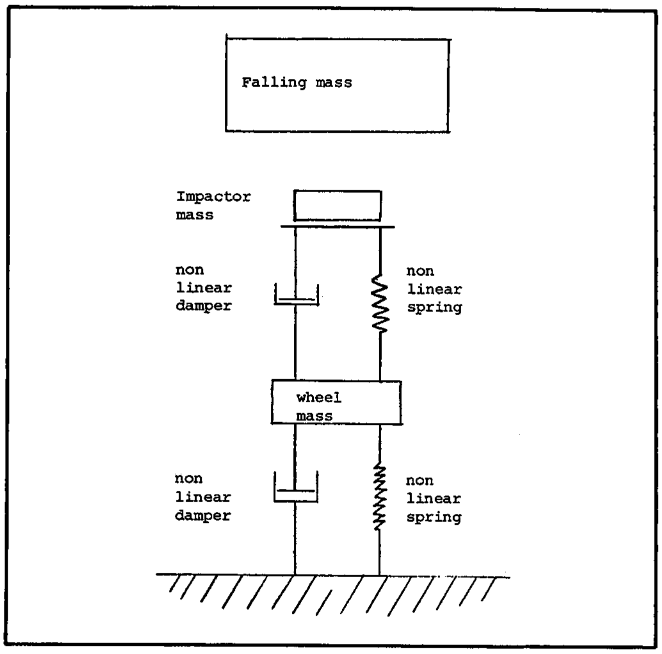

Figure 5. Diagram of mass-spring-damper model for front wheel drop test rig simulation

obtained by simple static crush tests with various velocity related factors, as on the full motorcycle model.

In practice, the struck rigid mass was used to obtain deflection/time characteristics. All the models have been written in A.C.S.L. which can be used for modelling continuous systems. The collision of the two rigid bodies was modelled by an instantaneous momentum balance relying on pre and post impact velocities. All models were designed to self-check using an energy based system.

The error in the experimental results was assessed by comparing the impact energy with the measured absorbed energy. The imbalance was used as a factor to weight the crush characteristics which were then averaged for each type of wheel and tabulated for use in the drop rig simulation. The velocity related factors in the simulation were adjusted to obtain an approximate fit to experimental results by achieving similar peak load and displacement values. A close fit was obtained by optimising the velocity factors to minimise the r.m.s. error between the simulation and experimental results. It has been found that damping significantly affects the system resonance and they relate non-linearly. This is particularly true for spoked wheels. Some drop rig tests have been performed on each type of wheel. When all the tests and simulations have been completed an impact on any wheel can be modelled up to an impact velocity of 15 m/s using only statically measured crush characteristics.

Drop tests at several different velocities need to be simulated for each type of wheel, but to date, only a few selected tests have been simulated. It can be seen

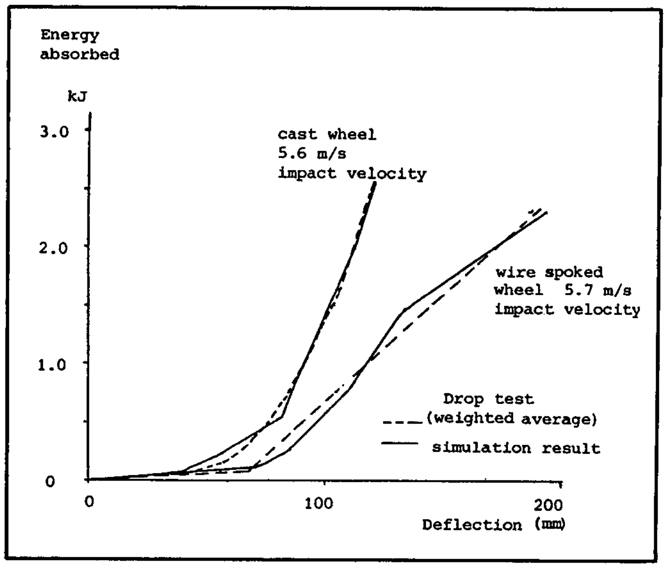

Figure 6. Comparison of results of drop rig mathematical simulation with weighted experimental results

from Fig. 6 that there is a good correlation for wire spoked and die cast wheels. The energy absorption can be separated into two sections, that absorbed by the tyre, which was never more than 10% of the total, and that absorbed by the wheel collapsing. Thus, there are distinct velocity related factors governing the behaviour of each section. Each half of the wheel model is simulated by a single non-linear spring and damper. When all the impacted wheels have been simulated then the variation of these velocity factors can be related to impact velocity and hence extrapolation can be performed to any impact velocity up to 15 m/s. A similar series of tests will be performed on front forks at a later date.

Wheel Test Results

Current production motorcycle wheels have been categorised, for test work, into three basic types: steel wire spoked, die-cast and a composite "comstar" type.

A comparison between the static and dynamic results highlights the following interesting phenomena. The wire spoked wheels showed a small increase in the peak load and a slight decrease in the energy absorbed for the dynamic tests compared with the static. During most of the impact, the dynamic load was significantly greater than the static load for both the cast and comstar wheels, see Figs. 7 and 8. However, the cast wheel failed prematurely in both tests and absorbed little energy although slightly more was absorbed in the static test. In contrast, the comstar wheel absorbed far more energy dynamically than statically, see Figs. 7 and 8. Whereas, the wire spoked and comstar wheels generally absorbed energy pro-

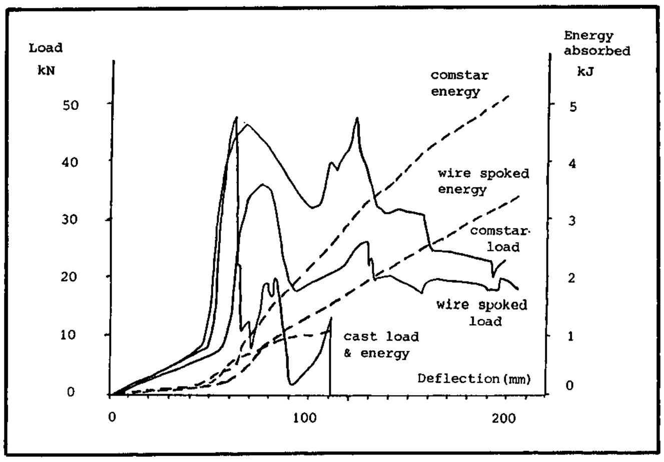

Figure 7. Comparison of static energy absorption characteristics of wire spoked, comstar and cast front wheels from medium weight motorcycles

gressively, the cast wheel did not. In Table 1 the energies absorbed and the peak loads are compared as a ratio using the cast wheel as a reference. This layout emphasises the phenomena mentioned above.

Results of Full Motorcycle Simulation

Two basic simulations of a medium-weight motorcycle and rider impacting a rigid barrier have been performed, one with a cast and the other with a wire spoked front wheel. The impact velocity has been restricted to 5.63 m/s (12.6 mile/h) because the wheel drop rig tests were not completed at the time thereby making extrapolation of velocity related factors difficult.

The cast wheel test induced a tendency for the motorcycle frame to pitch, once the front wheel has been trapped between the frame and barrier, see Fig.

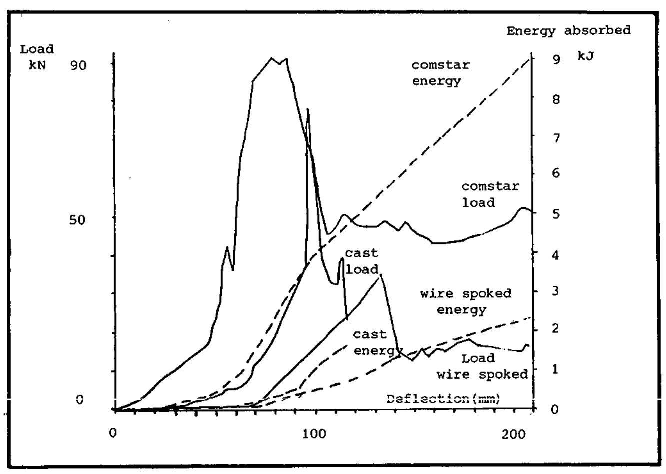

Figure 8. Comparison of dynamic energy absorption characteristics of wire spoked, comstar and cast front wheels from medium weight motorcycles. Average impact velocity was 5.41 m/s (12.1 mile/h)

Table 1. Table showing relative values for medium weight motorcycle front wheel types subjected to static and dynamic loading.

data type	factor considered	wheel type		
		cast	wire spoked	comstar
Static	energy absorbed for 112 mm displacement	1.0	0.71	1.65
	peak loads	1.0	1.32	1.33
Dynamic impact velocity 5.41 m/s	energy absorbed for 120 mm	1.0	0.7	3.86
	peak loads	1.0	2.26	2.65

9. The wire spoked wheel did not induce this tendency, see Fig. 10. Both these figures show the frame as a triangle with the front forks and the rider as a stick of 20 m/s intervals. The pitching tendency can be seen by following the rear portion of the motorcycle frame triangle.

Simulations have been tried at 13.4 m/s (30 mile/h) to gain an idea of the trend, without adjusting the velocity related factors from their value at 5.63 m/sec. The tendency for the cast front wheeled motorcycle frame to pitch was increased in these simulations. It is expected that inclusion of the correct velocity factors will exacerbate this tendency. This tendency to pitch was very evident from practical full scale impact tests on cast front wheeled motorcycles. Economy of computer use meant that the simulations were stopped at about 10 m/s after the rider had lost contact with the motorcycle.

The deceleration-time histories from mathematical simulations for both types of motorcycle are shown in Fig. 11. These curves show the peak deceleration of the cast wheeled machine to be twice that of the wire spoked version as in the practical tests. These curves also show the wire spoked machine to decelerate at a

Table 2. Table showing some results of mathematical simulations of medium weight motorcycle "head-on" impacts into a rigid massive barrier for differing types of front wheels.

	impact velocity 5.63 m/s (12.6 mile/h)		
	cast front wheel		wire spoked front wheel
Rider leaves motorcycle after	142 ms		175 ms
Period of rider contact with seat	93 ms		120 ms
Period of contact with petrol tank	52 ms		57 ms
Rider velocity on leaving petrol tank	forward	2.85 m/s	3.09 m/s
	vertical	2.11 m/s	1.55 m/s
	rotational	3.43 rad/s	3.14 rad/s
	Total translational	3.54 m/s	3.45 m/s

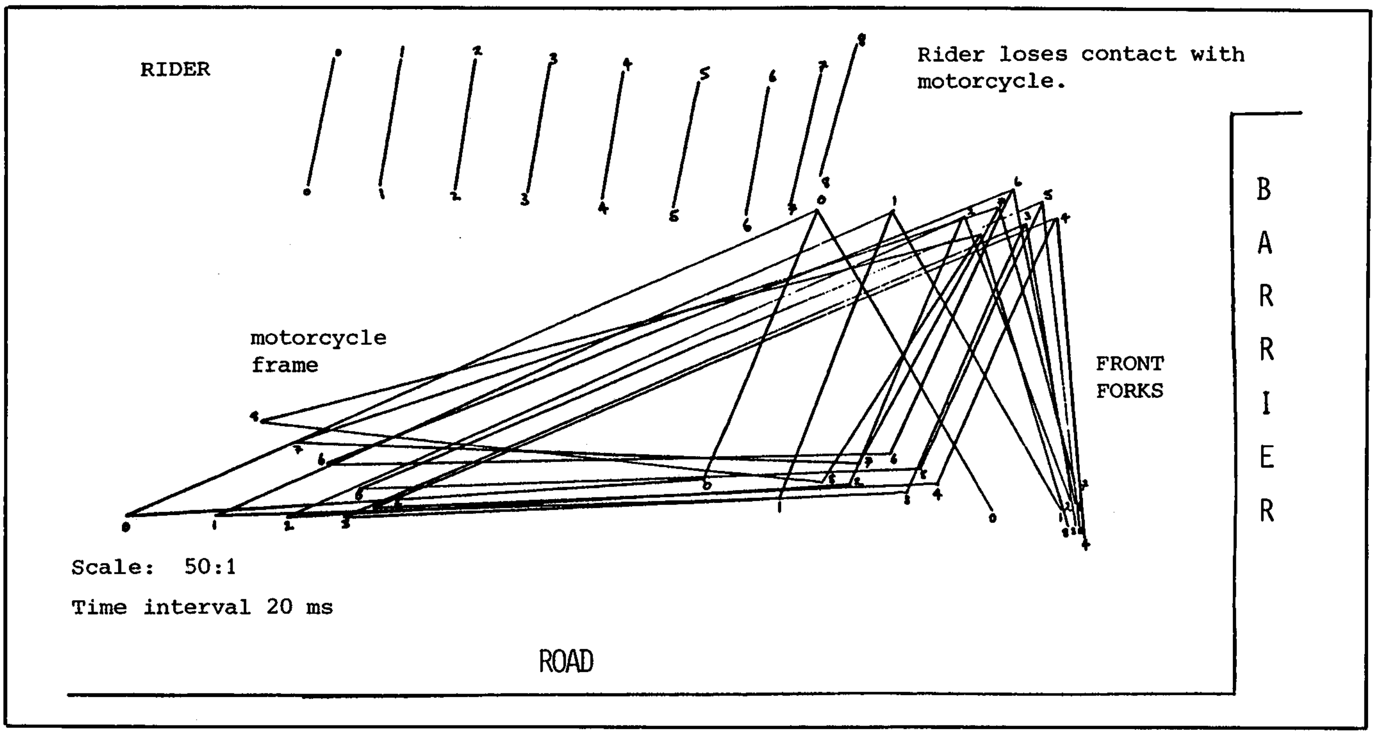

Figure 9. Simulation results for a medium weight motorcycle impacting a rigid barrier head on at 5.63 m/s (12.6 mile/h) with a cast front wheel

more consistent level than the cast machine, which would greatly aid the effectiveness of a restraining safety system.

The rider's vertical velocity on leaving the machine with a cast front wheel was $1\frac{1}{2}$ times that of the wire spoked version, see Table 2.

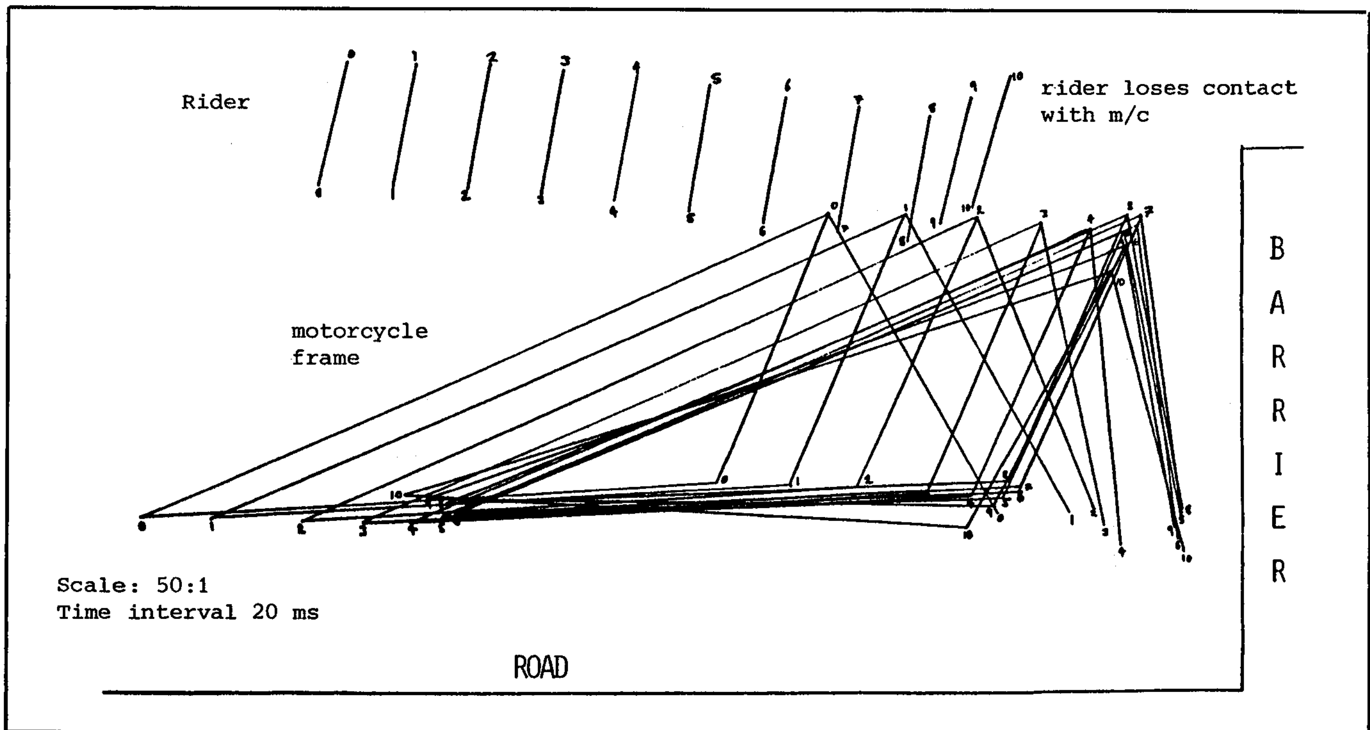

Figure 10. Simulation results for a medium weight motorcycle with rider impacting a rigid barrier head on at 5.63 m/s (12.6 mile/h) with a wire spoked front wheel

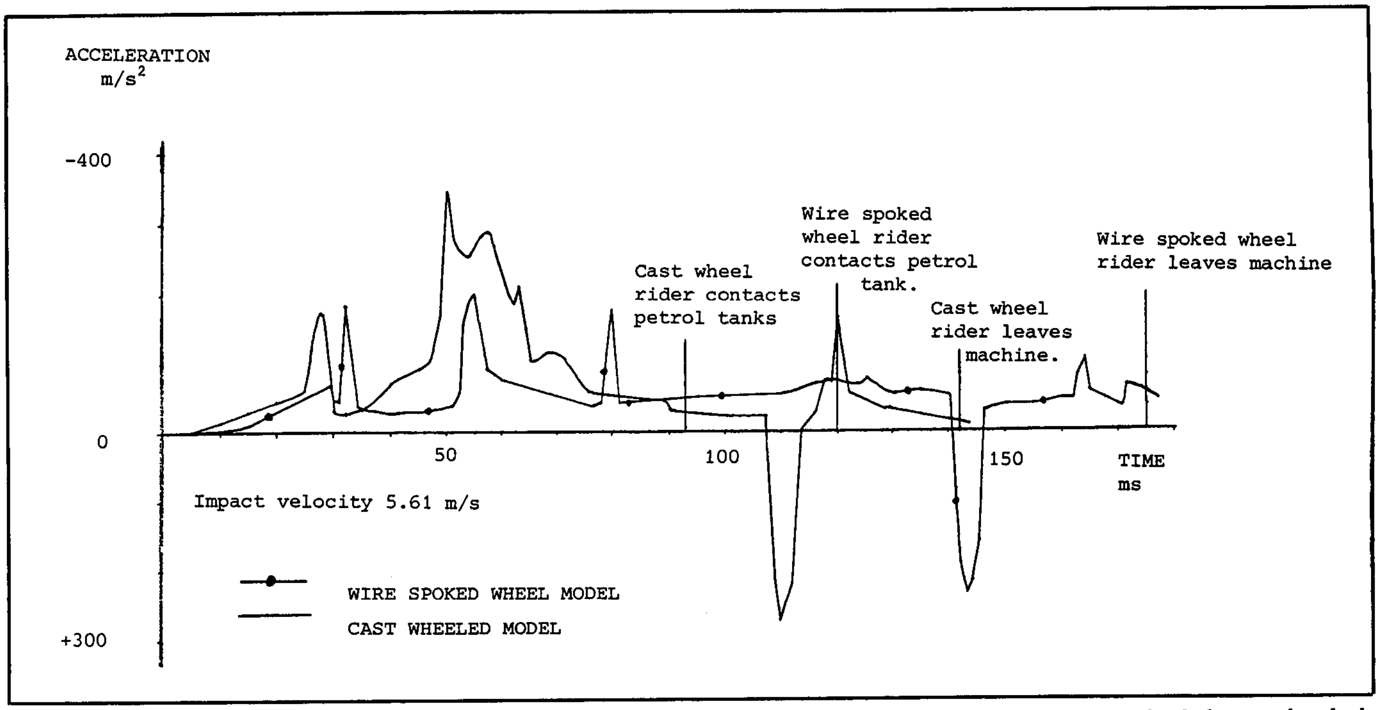

Figure 11. Comparison of motorcycle centre of gravity deceleration for cast and wire spoked front-wheeled medium weight motorcycles with time from mathematical simulations

However, both simulations predict that during the impact there will be an overall decrease in the rider's translational velocity of about 40% and a similar percentage increase in rotational velocity. This has not been observed in practical tests and indicates that a more sophisticated rider/machine contact model is required, possibly one which simulates all the possible contact points, such as both the seat and petrol tank simultaneously. It is considered that this will create more of an effect on the rider when the frame kicks up. The rider impacting the petrol tank appears to initiate the pitching effect, which is also responsible for launching the rider off the machine. The simulation of rider contact mechanics requires further work.

Rider-Restraint System

The computer programme is not yet fully developed. However, it indicates that a rider is retained on the motorcycle, following a frontal impact, for about 50 ms longer if spoked wheels are used than if cast wheels are fitted. In addition, the use of spoked wheels causes the motorcycle to pitch less. Both these factors are favourable to the effectiveness of an air-bag as a rider-restraint. An air-bag of 120 litre capacity (deployed with one inflator) has been assessed in impacts and it reduced the horizontal component of the rider's velocity to zero at the plane of impact. However, in these tests the motorcycle pitch was restricted to that expected if spoked wheels were used. Therefore, in practice, motorcycles would need to be designed with frontal energy-absorbing characteristics similar to those obtained with wheels of this type.

Although the horizontal component of velocity of the rider was reduced to zero by the presence of the 120 litre air-bag, some rotational energy was retained. This was also the case with the 60 litre air-bag used on ESM-2.

ESM-3 is fitted with a 120 litre air-bag similar to that used in the impact tests. The inflator and hence the bag retention is at the rear of the fuel tank. This arrangement has been found to be effective and does not need for any additional attachment of the bag. With development of the computer program, ideal restraint system characteristics should be predictable during impact.

Conclusions

A mathematical simulation of frontal impacts of motorcycles is being developed and shows promise of becoming a valuable research and design tool.

A method of analysing crushable structures has been developed whereby the dynamic crush behaviour for impact velocities up to 14 m/s can be predicted using statically measured characteristics. This type of analysis has been successfully performed on motorcycle front wheels and it is intended to treat front forks similarly. The characteristics predicted for the wheels were used in a mathematical model designed to simulate the motion of a motorcycle and rider when impacting a rigid barrier at 90°, which is known as "head-on". The predicted movements of the motorcycle correlate well with those seen in practical tests. In particular the tendency for a motorcycle to pitch (kick up at the rear) when fitted with a cast front wheel is apparent. Similarly the tendency not to do so when fitted with a wire spoked wheel, is accurately simulated by the model. However, the rider which is represented by a "stick", does not reproduce well, and is wrongly predicted to leave the motorcycle at approximately the same velocity regardless of the front wheel type. Track tests have shown that the velocity is greater with a cast wheel. A more sophisticated rider model is needed and it is intended to develop one.

The model predicts that a safety frontal restraint system would hold a rider on the machine for longer if a spoked rather than a cast front wheel is fitted. This is an example of how the model will be used and it is intended to create a database from which the head-on impact performance of any motorcycle can be simulated and hence its design to minimise rider injury can be readily determined.

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Lower Leg Injuries Resulting from Motorcycle Accidents*

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Lower leg injuries resulting from motorcycle accidents have been well documented over the years in the medical literature and recognized as one of the more frequent injuries resulting from an accident involving motorcycles on the roads worldwide. While many of the previous reports have studied these injuries in depth, few have carefully analyzed the exact mechanism by which the injuries occur. This study has carefully reviewed over 125 motorcycle accidents in which there was an injury to the lower leg. Of the cases reviewed, 58 were selected to do a static reconstruction of the accident, using exemplar vehicles and occupants in order to determine the vehicle dynamics and occupant kinematics of the operator/rider. The accident scenario for each case consisted of a car to motorcycle collision and all were less than 35 mph in speed. The predominant accident configuration was that of a left turning automobile in front of a motorcycle. The major injury was a fracture of the tibia and fibula and underlying soft tissue damage. Transverse fractures of the femur often result from loading of the distal end of the femur and rarely are the result of a direct impact. Fracture of the patella and dislocation of the hip are not a common occurrence in motorcycle accidents. It is quite apparent from the static reconstruction of the accidents that much of the soft tissue injury occurs after the initial impact with the opposing vehicle, and is not the result of the leg being crushed between the vehicles. In fact, the true crushing injury as is often described, is not in our opinion a correct description of the lower leg injuries in motorcycle accidents.

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Introduction

The medical literature has well documented the fact that lower leg injuries are frequently found in motor vehicle accidents involving pedestrians, bicyclists, and motorcyclists(1,2). Although some variation exists from study to study, the 14.1% lower leg injury frequency observed in Hurt's detailed analysis of 900 motorcycle accidents appears representative for motorcycling worldwide when allowance is made for the study data and local conditions. Surprisingly, while the medical literature reports on the frequency and characteristics of these injuries, only rarely has the actual mechanism involved in producing the injury been considered. The injury mechanism is the specific topic addressed in this paper.

All available accident and medical records were obtained for 127 motorcycle accidents involving injury to the lower leg. From this population, 58 accidents were selected for detailed evaluation by means of a static accident reconstruction using identical vehicles and people with similar body dimensions. This reconstruction greatly facilitated establishing the actual vehicle and rider/occupant dynamics during the collision and immediate post collision phases of the accident. All 58 of these accidents involved the motorcycle impacting with the front of a generally left turning automobile and with each vehicle traveling at 35 miles per hour or less. The resulting trauma frequently involved a comminuted fracture of the tibia and/or fibula with underlying soft tissue damage. The static accident reconstructions have established that the observed soft tissue damage was almost always a consequence of the initial impact of the leg with the automobile bumper and was not a consequence of the leg being crushed between opposing surfaces of the colliding vehicles. In effect, the sharp bone fragments themselves become agents of injury as they are propelled away from the impacting surface and into the underlying soft tissue. Furthermore, it was observed that transverse fractures of the femur often result from a distal loading of the femur without an accompanying fracture of the patella and without dislocation of the hip. This distal loading of the femur is generally associated with the knee impacting with the vehicle grill or fender.

Methods

This study was aimed at leg injuries, and therefore the type of accident cases we examined only addressed that area. The number of accident cases studied was 127 in which there was an injury to the lower extremity. For our purpose the lower extremity was defined as any part of the body from the head of the femur, or hip joint, to the foot. The vast majority of injuries were to the tibia/fibula region, which is often described as the lower leg. The next most often injured part of the lower extremity was the thigh region, in which the injury was a fracture of the femur. Injuries to the hip and foot were least often seen in this study, but again this was due to the type of accident studied.

The size of the motorcycle ranged from 90 cc up to the largest touring motorcycles on the roadway today. The presence of crash bars on the accident vehicles in this sample size was notably low, with only five cases in which the presence of a crash bar was noted. The crash bars that were present were the standard off-the shelf type or the smaller version that are considered engine guards.

The speeds for the motorcycle in the accident scenario ranged from zero (stopped to make a turn) to a high of 45 mph. The most prevalent speed was in the range of 25-35 mph. Speeds of the adverse vehicle ranged from 10-45 mph, with the most often speed at the low end of that range.

The usual type of accident seen in this study was the situation in which the car makes a turn in front of the motorcyclist, and strikes the left side of the individual riding the motorcycle. Obviously this results in an injury to the left side of the body. There were a limited number of cases in which there was a stopped motorcycle that was struck by a moving car. Also, there were a limited number of cases in which the accident was a single vehicle accident. In this situation it is usually a matter of the motorcyclist leaving the roadway and striking an object off the roadway.

In studying these accidents, the police report was obtained and reviewed for data concerning the nature of the accident, including size of vehicles, speeds, direction of travel, point of impact, point of rest for the vehicles and occupants, and type of damage and location. Where possible, photographs of the accident vehicles were studied to determine the extent of the damage from the accident. A complete review of the medical records was done to determine the extent and nature of the injury. This included the admitting report, operative report, radiology report, and nursing notes. In most cases, the x-rays were also reviewed. In many of the cases a complete examination of the motorcycle was done to determine and record the extent of the damage. This was useful in assisting with the understanding of the accident scenario. The type of damage to particular parts of the motorcycle would often provide useful data in determining the type of accident and the direction of the forces in the collision. There were a limited number of cases in which the adverse vehicle was also available for study and examination. In these situations we were able to study and document the extent of the damage to the opposing vehicle. Again, this provided significant data as to the type of collision that was involved, as well as helpful data as to the direction of force.

In cases in which one or both vehicles were available for inspection, they often provided valuable information concerning the points of contact of the body and thereby giving us data as to the mechanism by which the injury was produced. Frequently, it is possible to identify fabric transfer on one or both vehicles, location of material from clothing, and in some situations we were able to identify body imprints on one or both vehicles. These included dents in the fender and/or hood areas of the opposing vehicle, or a dent in the side of the gas tank.

All of the injuries sustained in the accident, were recorded onto a data sheet and then given an AIS number(3). In addition to the AIS scaling system, we devised our own system for further defining the type of fractures that occurred in the different accidents. This system allowed us to classify each type of fracture as to bone, location of fracture and nature of fracture (table 1).

In total 125 accidents were studied in depth where there was an injury to the lower extremity. In addition to the lower extremity injuries, a record was also made of any other injuries that were sustained as well in the accident. For each of the accidents a determination was made as to the mechanism by which the injury occurred. Of particular interest was the nature of the fractures of the long bones and the mechanism by which the injury occurred.

Of the 125 cases studied, 58 or 46% were selected for a static reconstruction of the accident. In these situations we used exemplar vehicles which were of the same make, model and year as the accident vehicle. We also would use people who were of the same physical size as those involved in the accident that was being studied. Using the information regarding the damage to the vehicle, we could then align the vehicles to determine the angle of the collision. This would be based not only upon the damage that resulted from the accident, but in addition we also considered the injuries that resulted from the accident. The purpose here was to determine both the vehicle dynamics in the collision, and the rider/occupant kinematics.

Table 1. UofL fracture classification.

ARM 1	FOREARM 2	FEMUR 3	TIBIA 4	SPINE 5	PELVIS 6	HAND 7
PROXIMAL 1	MIDSHAFT 2	DISTAL 3		ANKLE 91	CLAVICLE 92A	OTHER 10
					SCAPULA 92B	PATELLA 92C

SHAFT FRACTURES

A. SIMPLE FRACTURES A1 SPIRAL	A2 OBLIQUE .1 PROXIMAL .2 MIDSHAFT .3 DISTAL	A3 TRANSVERSE
B. WEDGE OR PIECE B1 SPIRAL WEDGE	B2 BENDING PIECE .1 PROXIMAL .2 MIDSHAFT .3 DISTAL	B3 2 OR 3 PIECES
C. BITS & BROKEN PIECES C1 SEGMENTED FX.	C2 BROKEN PIECE .1 PROXIMAL .2 MIDSHAFT .3 DISTAL	C3 FREE PIECES

Results

Tibia

In looking at the fractures of the long bones (i.e. tibia and femur) over half of the fractures were to the tibia, the number being 63% of the total fractures recorded. These ranged from simple non-displaced fractures to compound/comminuted fractures with significant soft tissue involvement. In studying the location of the fracture, the majority of the fractures were in the distal third of the tibia. Whenever there was a comminuted fracture of the middle and distal thirds of the tibia, there was an associated fracture of the fibula at the same location. Fractures of the tibial plateau were often associated with a fracture of the femoral condyles and significant involvement of the knee joint. Fractures at the tibial plateau that were displaced posteriorly often invaded the popliteal fossa, and the vessels and nerves within the fossa.

In examining the mechanism of injury through the use of exemplar vehicles and people, it is quite apparent that the injury to the middle and distal tibia is the result of the leg being impacted by the bumper of the adverse vehicle and then pushed in a rearward direction. This is typical of the situation where the automobile turns in front of the motorcyclist, and the latter then attempts an evasive maneuver to avoid the collision. The angle of collision between the two vehicles is usually less than 30 degrees. The leg is not pushed into the side of the motorcycle, but rather rearward along the side of the motorcycle.

In situations where fracture of the tibia occurs at the tibial plateau or the proximal one-third of the tibia, this is the result of the knee and upper tibia making contact with the fender region of the opposing vehicle. Again, the angle between the two vehicles is less than 30 degrees. When the impact to the leg is in this region of the body, there is often involvement of the peroneal nerve, which is just lateral and inferior to the knee. An injury to this nerve will affect the muscles of the anterior compartment of the lower leg, and the result will be foot drop and a numbness to the dorsum of the foot.

The type of fracture seen is usually a transverse or comminuted fracture in which there were many fragments of bone. A spiral or oblique fracture is seldom seen in accidents in which there are vehicle to vehicle collisions. We did observe a limited number of spiral fractures of the tibia, however, these were usually seen in situations of a single vehicle accident. In a single vehicle accident, the rider goes off the roadway and in an attempt to control himself and the vehicle puts his foot down and impacts the ground with sufficient force to cause a spiral fracture. As the foot impacts the ground, the force is transmitted through the foot and to the distal end of the tibia, often resulting in a tri-malleolar fracture or a spiral fracture of the tibial shaft.

Whenever there is a significant fracture involving the lower leg, with both the tibia and fibula involved, there is also soft tissue damage to the underlying tissue. This can be of particular concern when the blood vessels are damaged, and there is a loss of adequate blood supply. The main cause of the amputations is an inadequate blood supply to the tissue distal to the point of injury.

Femur

The incidence of fractures to the femur were approximately half the number of tibial fractures.

These usually occurred in the distal third, and were transverse in nature with little or no soft tissue involvement. The limited number of fractures at the head of the femur usually were associated with a dislocation of the head of the femur. Another problem seen with fractures of the head of the femur was also a fracture of the acetabulum.

The fracture of the distal third of the femur is almost always the result of an indirect impact, rather than a direct impact as is seen in the tibial fractures. The static reconstruction showed that as the knee would go into the fender or grill of the adverse vehicle, the force would be transmitted along the axis of the femur. While the distal portion of the femur is stronger than the shaft, the force is transmitted from the area of the femoral condyles to the shaft of the bone. The fracture then occurs at a point distal to the actual site of impact, and will usually be between the middle and distal third of the bone.

Fractures of the proximal third of the femur are usually the result of an impact to the ground, and not a direct impact from the opposing vehicle. In our study the limited number of cases where there was a fracture of the proximal third, it was determined for each of these that it was the result of an impact to the ground. When the neck of the femur is fractured, this is the result of the distal end of the femur being impacted, and then pushed rearward as the knee is held by the opposing vehicle. This is often seen in accidents where the knee goes into the compliant fender area of the car and is held there as the rider then slides forward.

Femoral fractures seldom have soft tissue involvement, and are usually transverse in nature. The lack of soft tissue injury is obviously due to the anatomy of the thigh region. The femur is surrounded by large groups of muscles. The notable exception to this is the accident where the neck is fractured and pushed rearward, and damages the muscles, nerves and vessels in the area.

Ankle & Foot

Injuries to the ankle and foot region are the result of the foot being caught by the bumper and pushed rearward. We often see the situation where the foot is caught by the underside of the bumper, and results in fractures to the tarsal and metatarsals. These types of impacts can also have involvement of the blood vessels as they are close to the surface of the skin in this region, and thus are vulnerable to trauma in this area. Impacts to the foot/ankle can and will often result in dislocations of the ankle.

Patella

The incidence of fracture to the patella is quite low. In the present study we recorded only 8 cases where there was a fracture of the patella. This fracture is usually seen where there is significant involvement with the distal femur and the femoral condyles. When there is an explosive fracture of the distal portion of the femur, it is indicative of a high impact, and there will be an associated fracture of the patella. However, from the reconstructions performed that demonstrate a loading of the knee region, with a fracture of the distal femur, there is rarely a fracture of the patella at the same time.

Discussion

In reviewing 127 motorcyle accidents in which there was a leg injury, it is quite apparent that the most frequently injured area of the leg is the region of the lower leg, with a fracture of the tibia and fibula. Whenever there was a comminution of the bone with soft tissue injury of the underlying tissue, the medical report would describe this as a crushing type of injury. While the injury may give this appearance to the attending physician, it becomes obvious when a careful analysis of the accident is made, that seldom is the injury mechanism a crushing injury. Using exemplar vehicles and people, and the information available concerning the speeds, and damages to the vehicles, we were able to reconstruct 63 of the accidents. In all but three of our cases this analysis showed the injury to be an impact type of injury, not unlike the situation in which the pedestrian is impacted by an automobile. The leg is not crushed between the opposing vehicles, but rather is pushed rearward along the side of the vehicle.

Much of the soft tissue injury is the direct result of the sharp bone fragments causing lacerations of the muscles and vessels. Once the bones have been broken by the impact of the bumper or fender of the car, the structural integrity of the leg is then lost and as the leg is moved the sharp bone fragments cause additional injury. This type of injury comes from the inside out, rather from the outside in.

The mechanism of femoral fractures that occur in the motorcycle accident is different from that seen in the automobile accident. The work of Nahum(4) reports on the incidence and type of lower extremity injuries that occur in automobile accidents. The report of States(5) clearly shows the mechanism by which a posterior hip dislocation occurs in an automobile accident. However, in the motorcycle accident, rather than a hip dislocation, the more frequent fracture is a transverse fracture of the distal third of the femur. The reason for this can be attributed to the seated posture. In the automobile, the occupant usually sits with the knees forward and close together. However, the motorcyle rider sits with his legs astride of the gas tank and therefore at approximately a 45 degree angle. If there is a passenger behind the rider, their legs are at even a greater angle. In this position, the head of the femur is rotated deep into the acetabulum and is held firmly. In the normal sitting posture as seen in an automobile, the head is not as deep into the acetabulum, and upon impact, the head is driven posteriorly and can often result in a fracture of the rim of the acetabulum.

In reviewing 127 motorcycle accidents in which there was an injury to the lower extremity, the mechanism by which these injuries occur is a high velocity impact, similar to that seen in pedestrian accidents. Static reconstruction of the accident clearly showed that the leg is not crushed between the two vehicles, but rather it is impacted and then pushed rearward along the side of the motorcycle. In looking at the types of leg injuries that result from motorcycle accidents, the type of injury is very similar to that seen in pedestrian accidents.

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Load Measuring Method of Occupant's Leg on Motorcycle Collision

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Abstract

In the past, researchers have studied the leg protection devices. We can say there is no effective device which protects motorcycle occupant's legs without negative side effect.

In the development of effective leg protection devices, it has become necessary to analyze leg injury mechanism.

In this regard, advancements in the measuring method of leg load analysis are viewed as a significant contribution to the further research of leg protection device.

In this paper we have examined, tested and analized leg load measuring method.

Introduction

Researchers have been studying with the objective of developing leg protection devices for motorcycles. The leg protection devices investigated thus far, however, show a potential for increasing injuries to the other parts of the body. These result from the preservation of leg space which is a necessary premise to leg protection. This contradiction has not yet been solved.

Recent studies have focused on reducing leg injuries without increasing the injury potential to the other body regions. One of pontential approaches to achieve such objectives is to measure the severity of leg injuries quantitatively.

We believe that the leg load is one of factors affecting the injury severity, and that by learning the load characteristics on the leg, which have not fully been explored in the past studies, we will be able to contribute to the further study of the leg protection device.

In this regard, we considered the development of a leg load measuring method as a key factor, and have studied the methods for leg load measurement.

A Review of Leg Protection Devices for Motorcycles

Leg injuries resulting from contact with an opposing vehicle are considered to be significant due to the exposure of the motorcycle occupant's body to the environment.

Therefore studies for leg protection have been carried out over two decades.

A summary of the results on previous experimental leg protection devices is presented for an easier understanding of the situations (Table 1).

As mentioned in Table 1, countermeasures for achieving leg protection effect are accompanied by new problems. At present, effective leg protection

Table 1. Previous studies and experimental results regarding leg protection devices.

Period	Focus/Point of View	Experimental Results
The late 1960's-1971	A conventional bar was used to prevent the occurrence of severe injuries resulting from leg trapping between the motorcycle and an opposing vehicle.(1)*,(2)	The collision load being quite large, the conventional bar is weak enough to be bent thereby demonstrating the absence of any protective effect.
1972-1973	A reinforced bar was tested with the expectation of providing a leg protection effect by preserving leg space. (3), (4)	It preserves leg space under certain collision situations, however, it is necessary to investigate the adverse effects on the other parts of the body. The leg is fractured by hitting bare steel tube structure directly. A countermeasure to prevent the knee from hitting to the device is necessay.
1974-1976	The effect of reinforced leg protection devices was investigated through various collision tests.(5),(6)	It preserves leg space under certain collision conditions. It has potential for increasing head, chest, pelvic and upper leg injury, since the device makes rider ejection easier.
1975-1981	The effect of a knee cushion pad was studied as a countermeasure for protecting the leg from hitting the steel tube structure.(7)	As the knee intrudes into the pad and is trapped, pitching of the torso is increased and head hits the lower portion of the other vehicle's door. As a result, the occupant's neck is subjected to severe flexion and likely fracture.
1983-1985	In order to reduce the head impact velocity, the effect of energy absorbing materials placed on the sides of the motorcycles was investigated.(8),(9),(10),(11)	Energy absorbing materials contributes to a slight decrease of motorcycle impact acceleration. There are two sets of data regarding head velocity, one showing increased head velocity, the other decreased.

devices which do not increase injuries to the other part of the body, have not yet been developed.

The target of current efforts is to eliminate negative side effects while maintaining leg protection effect.

Achieving this target has been quite difficult, since the adverse effect is due to the preservation of leg space, which is the very principle that has been advocated for leg protection.

A Review of Leg Load Measurement Methods

In order to evaluate leg protection devices of various researchers, various methodology were contrived and employed in the past, and they were satisfactory for the purpose of evaluating these leg protection devices.

We have come to recognize from our studies to date, however, that in order to develop an effective leg protection device, it is necessary that we understand the characteristics of the external force, i.e. leg load characteristics (direction, location, timing, mode etc.), exerted against the leg. The methodology employed in the past is hardly satisfactory for such needs. Therefore, we believe that it is necessary to develop a dummy which allows us to measure various load characteristics which may be relevant to the development of a leg protection device.

Viewed from this perspective, a dummy does not exist which can quantitatively evaluate the injury severity. In order to quantitatively evaluate lower extremity injuries during collisions, it is necessary to have:

- A dummy which approximates the human body, to accurately simulate injuries to the lower extremities.

Table 2. Comparisons between various methods for leg load measurement in terms of their merits and demerits.

Measurement Method \ Evaluation Item	Load Direction	Load Timing	Load Mode	Quantitative Measurement	Measurable Area	Fidelity of the Rider Kinematics	Impact Point	Remarks
Leg Acceleration Measurement(1),(3)	N 1 axis	Y	N	Y Acceleration	△ Narrow	N	N	Leg trapping load can not be measured.
Strain Gauge on a Metallic Bone (5)	N	Y	N	Y Bending Moment	△ Sensor vicinity	N	N	If the point of impact misses the position of the gauge the value cannot be obtained accurately.
Knee and Ankle Load Measurement (6)	N	Y	N	Y Load	N quite narrow	N	N	Load Measurement is not possible if the load point misses the load sensor.
Breakable Resin Bone (6),(9)	△	N	Y	N Above or below threshold of fracture load	△ Area between joints	△	Y	When fracture occurs leg tends to detach. Quantitative data cannot be obtained.
Metallic Plate and Aluminum Honeycomb(8),(10),(11)	△	N	N	Y Total amount of Energy	△ Side face only	N	△ Side face only	Effective measurement is limited to the right angle component of impact only. It is difficult to interpret the dent of honeycomb.

Y……measurement possible
△……measurement possible to a certain extent
N……measurement impossible

*Numbers in parentheses designate references at the end of this paper.

- A dummy capable of load measurements to enable accurate assessment of the injury severity and a correlation between injury and impact load to the lower extremities.

In the light of this, it was decided to first focus on and discuss methods for load measurement applied to the lower extremities. Table 2 summarizes the evaluation of the merits and demerits for various measurement methods proposed in the past in terms of the necessary parameters to be considered.

Despite the efforts that have been made to date by researchers, we regret that as of yet, there is no method which has gained universal acceptance by researchers in the load measurement field.

Examinations on Measurement Items and their Methods

Measurement items

As mentioned previously, researchers have made efforts to devise measurement methods. We submit this study with hopeful exception that it will contribute to progress in this area.

When conducting leg injury measurements, it is necessary to consider which region of the leg should be focused on. Thus, these examinations must be based on injury analyses of the actual accidents and dummy impact conditions observed during collision experiments. Here, measurement items to be focused on are chosen based on the dummy impact conditions observed during collision experiments. A summary of the results is presented in Table 3.

The Measurement Methods

Measurement of the Maximum Value, the Position and Direction for the Bending Moment Applied to the Femur and Tibia.

In the event that a load is applied as in Fig. 1-a, if the load point (bumper position) is always constant, attaching a strain gauge to the load point is sufficient for measuring maximum bending moment (Fig. 1-b).

Table 3. Impact conditions for dummy during collision experiments and required measurement items.

	Dummy Impact Condition during Collision Experiment	Required Measurement Items
Lower leg	Lower leg is impacted with the bumper of the other vehicle, while knee and ankle are supported by tank and engine of the motorcycle, respectively. As a result, the lower leg may be trapped. Pelvis is pushed up onto the hood of the opposing vehicle. Less possibility of the lower leg receiving a compression load between the ankle and the knee is observed.	To the tibia ● Bending moment ● Load point ● Load direction ● Torsional moment between the knee and the ankle
Upper leg	The femur hits the edge of the opposing vehicle's hood. The knee penetrates into the opposing vehicle's front grill. The lower leg will be trapped and the torso moves against the opposing vehicle	To the femur ● Bending moment ● Load point ● Load direction ● Compression load ● Torsional moment between the knee and the pelvic joint

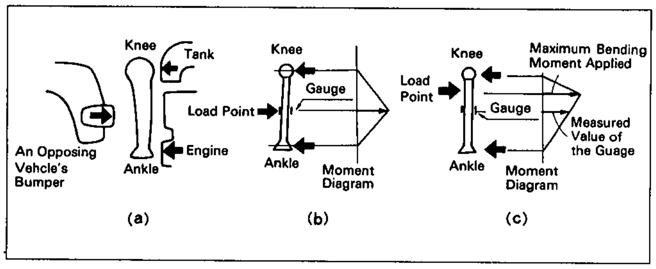


Figure 1. Differences in values measured by the load point and gauge position

If the load point varies and misses the strain gauge position, the strain gauge indicates a lower value than actual maximum load and can not measure maximum bending moment (Fig. 1-c).

Examinations were made as to assuming measurement methods to determine the maximum bending moment and load point under the condition where the load point changes. As a result, when the bending moment sensor is set between the supporting point and the load point, the bending moment at each point along the beam is shown with the line which links the bending moment of the supporting point (bending moment = 0) and that of sensor position.

On the other hand, if the same sensor is set near the other supporting point, another bending moment line can be drawn. The cross point is considered to indicate the value of the actual maximum bending moment and the load point position (Fig. 2).

Thus, when two bending moment sensors are placed near the knee and ankle respectively, the maximum bending moment and load point can be determined. The same analysis or approach applies to the femur.

The direction of the load applied can be determined from the measured bending moment around longitudinal and lateral axes and sensitivity curves of the sensors.

Therefore, to determine the direction, it is necessary to install bending moment sensors around the longitu-

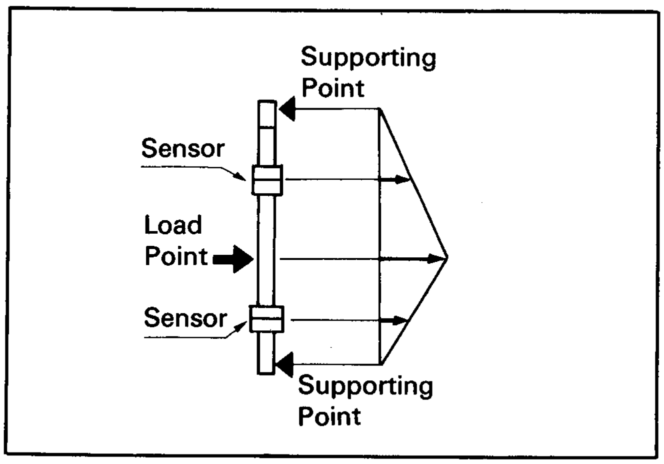


Figure 2. Determination of load point

dinal and lateral axes, and the obtaining sensitivity curves of those sensors.

Measurement of Torsional Moment Applied to the Tibia and Femur

It is reasonably assumed that uniform torsional moment exists between joints. Thus, it is possible to take measurements by installing a torsional moment sensor somewhere between joints.

Measurement of Compression Load Applied to the Femur

Since compression load also works uniformly between joints (as does torsional moment), it can be measured by placing a compression load sensor in one place between joints. A summary of required sensors is shown in Table 4.

Examinations on Various Factors Affecting Measurement Results

Factors effecting bending moment measurements. For the usage of this measuring procedure, factors which have an effect on the measured values are considered and discussed along with the items shown below.

- Joint tightening torque
- Elasticity of muscle around the bone
- Stiffness of bone itself
- Influence of bone fracture phenomenon

Joint tightening torque. Joint tightening torque adjustments for dummies in normal collision experiments are set on the order of one G. Since this level of tightening torque is small as compared to bone fracture level, the effect of the tightening torque is considered to be negligible.

Elasticity of the muscle around the bone. The elasticity of dummy muscle which is currently standardized is considered to be far below the bone fracture level and is also considered to be negligible.

Stiffness of bone itself In a simplified model where rigid mass (m) collides with an elastic body (elastic coefficient (k)) at the velocity V, the maximum force within a linear range can be represented by the formula $F=\sqrt{mk}\ V$.

This means that where the stiffness of the dummy bone is twice as high under the same conditions the resultant load becomes 1.4 times higher.

Table 4. Required sensors.

Tibia	• The bending moment sensors for X and Y directions of the upper part of the tibia • The bending moment sensors for X and Y directions of the lower part of the tibia • Torsional moment sensor between the knee and the ankle
Femur	• The bending moment sensors for X and Y directions of the lower part of the femur • The bending moment sensors for X and Y directions of the upper part of the femur • Torsional moment sensor between the knee and pelvic joints • Compression load sensor between the knee and pelvic joints

X,Y,Z : see Ref. (12)

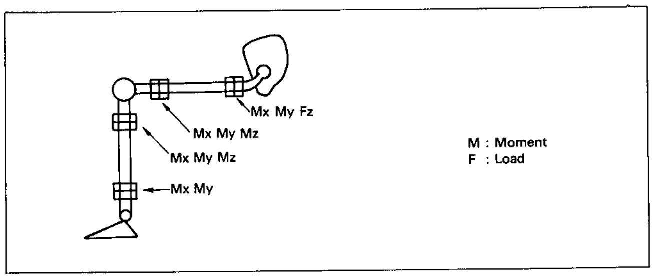


Figure 3. A schematic of sensor positions

Comparison between human and dummy leg bone characteristics from this perspective is shown in Fig. 4.

Judging from Fig. 4, the stiffness of current dummy leg is higher than that of human leg. By attachment of the load sensor to this excessively stiff dummy leg, the data as mentioned previously, will differ from the actual load values for human leg. In order to determine the correct leg load, it is first necessary to obtain the stiffness for dummy leg bones.

Factors effecting torsional moment measurements. The same considerations can be applied for the measurement of torsional moment as for bending moment.

This means that for the measurement of torsional moment, the combined effect of joint tightening torque and muscle elasticity is small and may, therefore, be ignored. On the contrary however, from the perspective of bone torsional stiffness, the maximum torsional moment as well as measurement of load, is proportional to the power of one half of the modulus of the elasticity in shear. Therefore, to measure the torsional moment applied to a human leg bone, it is necessary to obtain the torsional stiffness value for the dummy leg bone in advance.

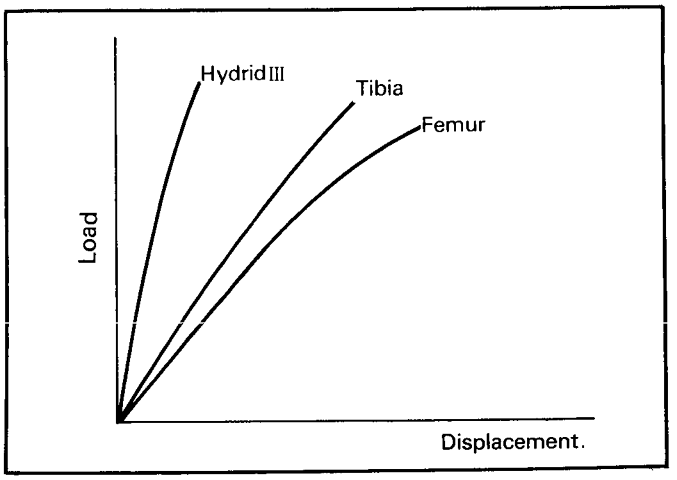


Figure 4. Comparison of human versus dummy bone characteristics

Table 5. Hybrid III dummy measuring system.

Tibia	Lower Tibia Sensor	Mx Fy Fz
	Upper Tibia Sensor	Mx My
	Knee Clevis Sensor	Fz
Femur	Lower Femur Sensor	Mx My Mz Fx Fy Fz
	Upper Femur Sensor	Mx My Fx

Verification of Measurement Procedures

Selection of Dummy

A verification has been conducted to determine whether the measurement methods discussed in this paper would be practical by using the Hybrid III dummy leg sensors which are considered to be nearly in conformance with our needs, although it does not have all of the sensors mentioned in Table 4.

Measurement parameters with the Hybrid III dummy used in this study are shown below.

Load Determination

The tibia of the Hybrid III dummy was used in impact experiments employing an impactor. A study was carried out to find out whether the load position and direction could be determined from measured data.

A photo is shown in Fig. 5.

Determination of Load Direction and Bending Moment at Sensor

In order to determine the load direction, sensitivity curves for Mx and My sensors were first obtained. Fig. 6 shows the sensitivity curve for the Mx sensor as an example. Sensitivity curves for all sensors are similar to a circle as well as the sensitivity curve for the Mx sensor.

Therefore, the load direction (θ) and actual moment (Mu) at the upper tibia sensors can be determined by applying the measured values Mxu, Myu of the Mx, My sensors to the following formulas (Fig. 7).

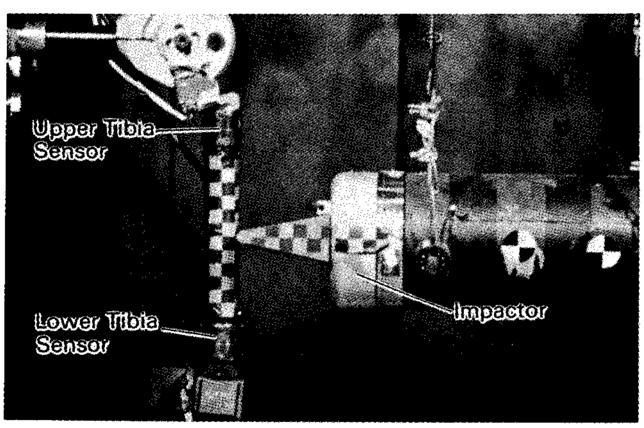


Figure 5. Impact apparatus

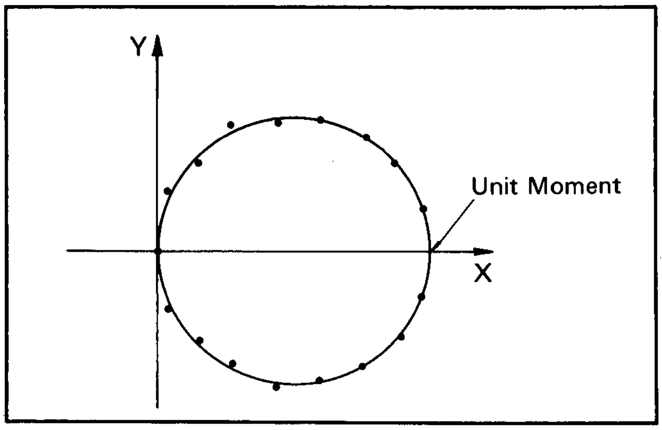


Figure 6. Mx sensitivity curve

On the other hand, the actual moment at the lower tibia sensor (M_L) can be obtained from the measured value M_{XL} of M_X sensor using the formula $M_L = M_{XL}/\cos\theta$

Determination of Load Point and Amount of Load

The load point (X) is determined by using Mu (refer to Fig. 7) and M_L (Fig. 8).

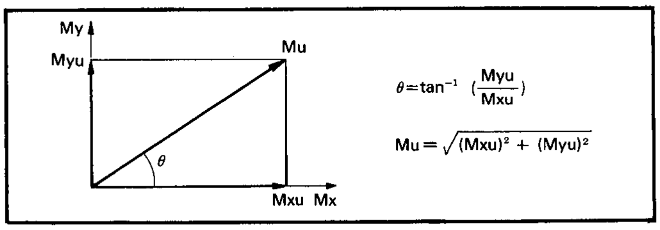


Figure 7. The relation between Mxu, Myu, θ and Mu

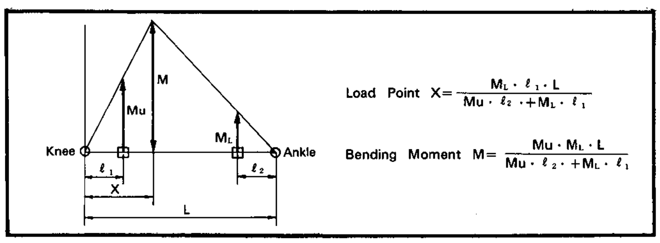


Figure 8. Determinations for load point and bending moment

The amount of load (W) is determined by using M in the formula below (Fig. 9).

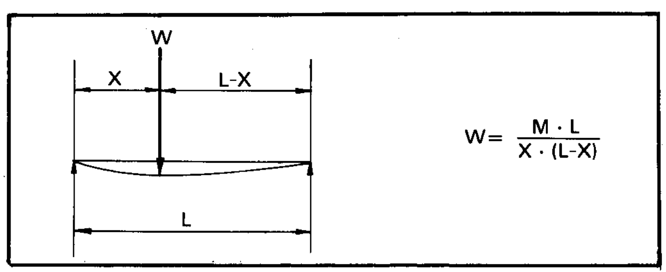


Figure 9. The determination for the amount of load

Table 6. Impact condition and results of analysis (N = 3).

	Load Direction (deg)	Load Point (mm)	Load Amount (kg)
Impact Condition by an Impactor	Oblique forward 45°	Below the knee 215	850±50
Results analyzed by the Proposed System	40°±5	220±10	610±50

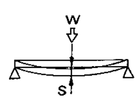
Table 6 shows the results of an analysis of a known impact condition by using the above procedure.

Effect of Leg Bone Bending Stiffness on the Measured Value

The square root of the bending stiffness*1 of the metallic leg bone employed in this dummy leg was compared with that of a human bone and was found to be 3.25 times higher. Therefore experiments were conducted employing a Bakelite bone which has a stiffness similar to that of human bone for comparison purposes. The results are shown in Table 7.

Table 7. Leg bone bending stiffness and impact load (N = 3).

	Bending Stiffness*1 (kg/mm)	$\sqrt{\text{Stiffness ratio}}$	Impact load*2 (kg)
Human tibia bone	29.6 (14)	1	–
Breakable Bakelite bone	26.8	0.95	270±25
Hybrid III dummy Tibia (Aluminum Bone)	313.5	3.25	850±50



Note:
*1 Bending Stiffness=W/S
*2 Impact Condition=Impactor Mass 30kg
Impact Velocity 2.0 m/s

Judging from Table 7, it can be seen that the impact load is influenced by the stiffness of the leg bone material. The Hybrid III leg bone shows a distinctly higher value due to its high rigidity.

In this impact experiment, however, a Hybrid III dummy original leg was employed, and so the value of the impact load was also influenced by the stiffness of the knee clevis and joint. As a result, the impact load of the aluminum bone from Table 7 indicated a slightly lower value than (the value for the impact load of the Bakelite bone) × $\sqrt{\text{stiffness ratio}}$.

The Effect of the Leg Load Measurement System on Rider Behavior

The current leg sensors for the Hybrid III dummy are attached to the metallic leg bone. As a result, fracture does not occur, as it did with the breakable Bakelite bone previously employed. Thus, an excessive load can be applied with no fracture and can influence dummy kinematics, especially, lean behavior of the dummy torso in the broadside collision. Therefore, dummy kinematics was observed by simulated collision experiments on sled tests. Fig. 10 shows the device used in the experiments.

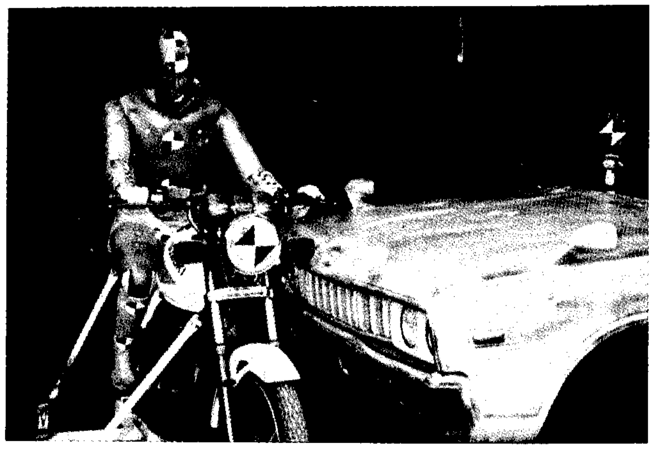


Figure 10. Sled tests set-up for simulated collision

With the dummy having the metalic bone, as opposed to the dummy having the breakable Bakelite bone which fractures upon impact, there will be no fracture even though a torsional moment is applied to the metalic bone when the inclination of the dummy pelvis exceeds the movable range of the pelvic joint, which tends to restrict the inclination of the pelvis. Fig. 11 shows the difference in the behavior of the pelvis and lumbar spine.

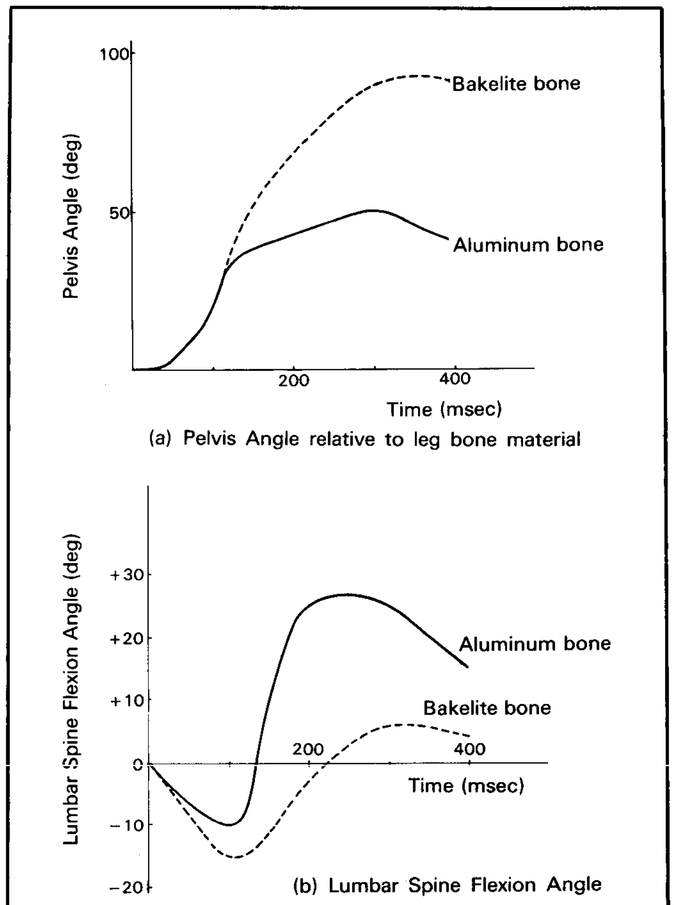


Figure 11. The difference in pelvis and lumbar spine behavior by leg bone materials

In observing aluminum leg bone, it was noted that along with the difference in pelvic behavior, the lumbar spine tended to be bent severely, and hip lift tended to be restricted. This is shown in Fig. 12.

Therefore, in comparing metallic bones to breakable Bakelite bones, behavioral differences were observed as follows:

- Pelvic inclination was restricted
- Lumbar spine was bent tremendously
- The amount of hip lift was lesser

We found that since the fracture phenomena cannot be simulated with a current metallic bone, the occupant's behavior was impaired. A consideration of fracture characteristics was found, therefore, to be necessary in order to properly simulate a occupant's behavior.

Discussion

Measurement Methods

The methodology in which sensors are placed in the leg was found to be useful in determining the load point, load direction, amount of the load and load timing from the experiment results.

Characteristics of Bone Stiffness and Fracture

The results of the tests verified that it is important for correct measurement, to obtain resultant stiffness of the leg system, including joint portions beforehand.

Furthermore it must be considered that a human leg bone is fractured if load exceeds a certain value. Therefore, excessive load can not be applied.

With metallic bones, however, fractures are not produced even after having reached the level at which human bones would fracture. In this case, the load continues to be applied. This has an effect on the primary kinematics of the dummy in the collision experiment.

Since it is usual to observe not only the measurement of applied load to the occupant's legs but also occupant's kinematics, it is necessary to take into consideration both fracture as well as stiffness characteristics.

Simulation of leg bone fracture characteristics leads to restricted measurement below the bone fracture level. It is important, therefore, to select an appropriate dummy leg bone for the purposes of particular experiments.

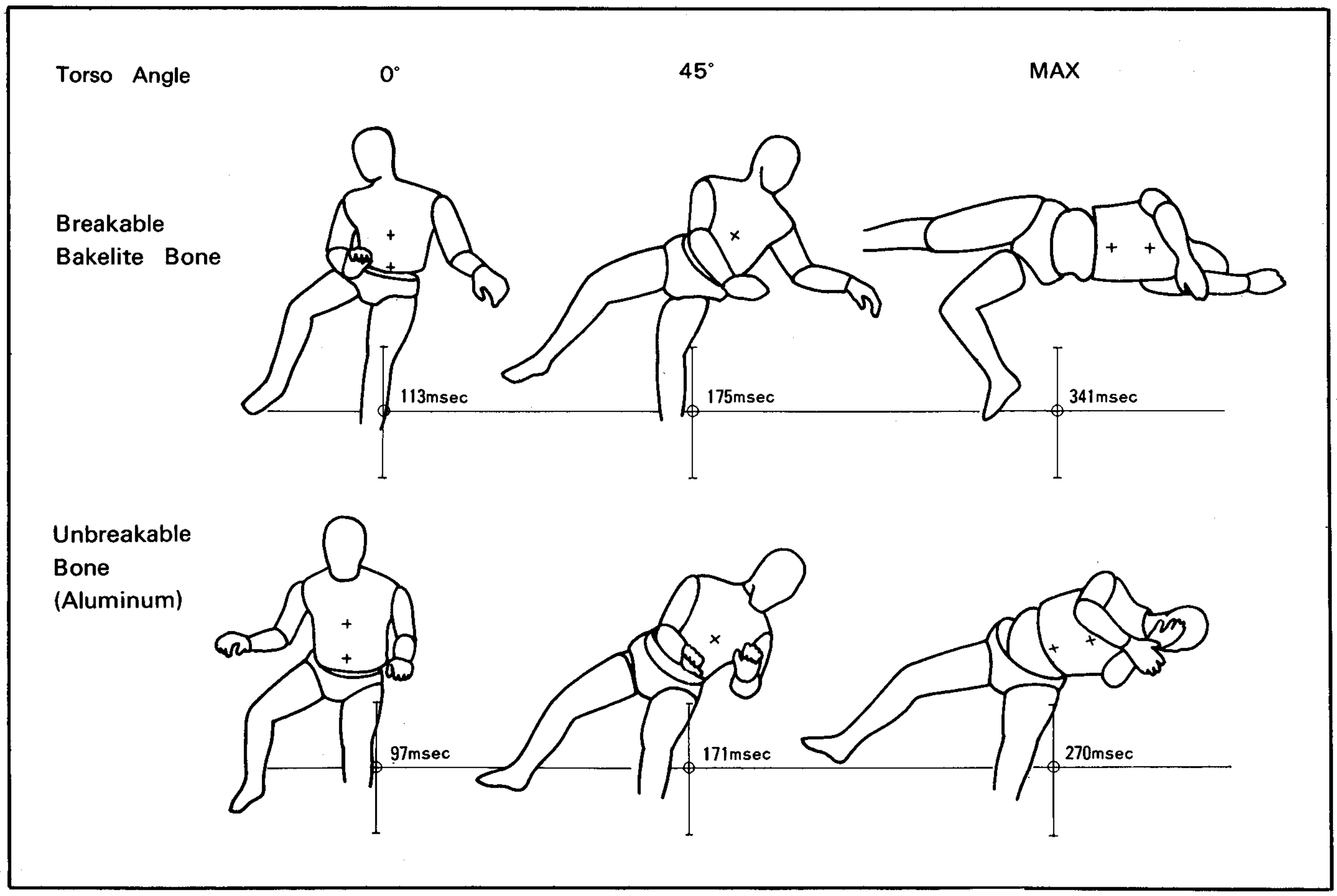


Figure 12. Difference in lumbar lean behavior depending on leg bone materials

Measurable Area

In the case of the measurement of the bending moment with the proposed measurement system, the measurable area is restricted to a leg bone portion which connects the sensors. Thus, the larger the joints and sensors are, the narrower the measurable area is.

Therefore, to widen the measurable area, it is necessary to devise sensors and joint portions as small as possible.

Cross Sensitivity

Cross sensitivity has two meanings. The first is to pick up the signal of the direction which crosses the main measurement axis. The second is those cases where undesired various items other than the target item are measured. The former becomes a cause of errors in determining the load direction, however, the upper tibia sensors employed in these experiments showed no significant error.

The lower femur sensor system, employed this time, unified several sensors, and was found to have unfavorable characteristics. For example, if a compression load is applied, the sensor indicates as though bending moments around X and Y axes are applied.

A sensor having this type of characteristic is not suitable for motorcycle collision analysis which requires strict distinction between compression and bending. Therefore, it is necessary to develop a sensor having minimal cross sensitivity.

Characteristics of Bone Stiffness and Fracture

Researchers in the past endeavored to measure injuries to the lower extremities for evaluation of leg protection devices. However, no measurement method has been accepted worldwide. Each method has its strong and weak points. As an example, the fracture simulated bone employed in JAMA studies (6), (9) indicates whether or not a fracture exists and also the fracture mode.

Those employed by G.W. Nyquist et al. (13) demonstrate the existance of bone fracture. With respect to those points they are considered to be superior but they do not measure quantitatively.

The method used by B.P. Chinn et al (8), (10), (11) whereby aluminum honeycomb is attached to a metallic bone can express the total amount of energy applied. An accuracy could be expected if the impact came from right angle to the honeycomb. In addition, it is also difficult to distinguish the load mode.

The strain gauge system utilized by Bartol et al (5), and JAMA study (6) is effective only with a definite load point.

Further improvements are required before the procedures which have been designed and proposed in this paper thus far can be useful. We believe, based on this study, that the system employed has the capability of measuring the time, location, mode, direction and amount of load.

Conclusion

The items below have become clear with regard to leg load measurements.

- The methodology for measuring leg load have their own strong and weak points, and they can not provide measurements on all the required items.
- There is no standardized method at present which is accepted by researchers for motorcycle collisions. Researchers have devised their own methods depending upon their particular needs at the time.
- The dummies developed for automobiles in the current state are not entirely suitable for the leg load measurements as we needed in this study, because the dummy receives different modes of load in the motorcycle collisions than in the automobile collisions.

Based on the observation of the collision experiments, it was found necessary to measure the following items:

- Bending moment and torsional moment applied to the tibia.
- Bending moment, torsional moment and compression load applied to the femur.

Regarding those items for which measurements are necessary, we examined the methodology to determine the impact point and direction, the load amount, load timing as well as mode of load.

As a result, placing sensors on the leg bone was proposed. With numerous sensors attached to the leg of the Hybrid III dummy, this leg measurement system was subjected to impact tests for verification.

The result indicated that following elements can possibly be determined.

- Load point
- Load amount
- Load direction

Through these discussions and tests, some points were noted which need to be improved. For example:

- To devise sensor and joint portion which are as small as possible.
- To develop a sensor having minimal cross sensitivity.

Although the measurement value and behavior of the dummy can be approximated to those of human being in the range short of bone fracture by making the characteristics of the dummy bone similar to the human bone, the usage of such bone will impair the measurement above that range. Therefore it is essential to select a dummy bone suitable for particular tests.

Further studies for the improvement of the methodology proposed here will be conducted in the future.

It is hoped that a more suitable dummy for load measurements will be developed after advancements in the field of the accident injury analyses. And it is also hoped that through the use of dummy above mentioned, a clue to the development of a practical leg protection device can be found.

Acknowledgments

The author wishes to express a special word of thanks to MR. K. Miyazaki and other members of Japan Automobile Research Institute for their expertise to conduct the experiments.

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Reduction of Injury Severity Involving Guardrails by the Use of Additional W-Beams, Impact Attenuators and 'Sigma-Posts' as a Contribution to the Passive Safety of Motorcyclists.

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Abstract

Motorcycle collisions with guardrails are a severe safety problem. Especially hitting the posts lead to severe injuries of riders and pillion riders. A total number of 150 fatalities out of 1,000 per year is estimated for the Federal Republic of Germany. To avoid fatalities and to reduce the severity of injuries these solutions are presented:

- fitting of the so-called "Sigma-Post" instead of the mentioned "IPE-100"
- fitting of an additional W-beam
- fitting of special impact attenuators to guardrail posts

Biomechanical tests prove the effectiveness of impact attenuators, of which approximately 20,000 have been fitted to guardrails in the Federal Republic of Germany since 1984/85.

Although not yet valid and reliable, field tests seem to prove the effectiveness of both the additional W-beam and the impact attenuators in terms of reducing the injury severity. Additional W-beams seem to reduce the number of accidents, too.

Finally the results of the cost-benefit studies are presented, which prove that under certain, realistic conditions the implementation of protective measures is to be considered positive.

Definition of the Problem

Motorcycle accidents of various configurations, falls or collisions with other road-users, can result in riders sliding along the road surface and hitting a section of guardrail after having been separated from their machines.

The Federal Highway Research Institute systematically analyzed motorcycle accidents in the Tuebingen area to find out whether they led to riders colliding with an obstacle beside the roadway. In 1984, 2793 accidents occurred in this region with 7 of the 44 deaths resulting from impact on guardrail (Domhan, 1987, page 205).

If taken as a basis for the whole Federal Republic of Germany, this and other regional surveys give us an approximate figure of 150 "guardrail deaths" from a total of about 1,000 (in 1986: 972) deaths per year. This is a share of 15%.

The fallen motorcyclists usually sustain their injuries from a collision with the guardrail posts (Schueler et al, 1984). Until 1985 nearly all posts to be fitted were the so-called "IPE-100-posts" which are particularly aggressive owing to their form and material.

The probability of hitting a guardrail after a fall is relatively high due to the fact that 10% of the 500,000 km of roads in the Federal Republic are equipped with such. Naturally there exists a higher density of guardrail on dangerous stretches of road than suggested by the statistic average (Motorrad, 9/87, p. 228).

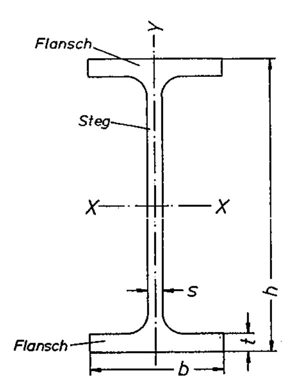
The results of an accident involving guardrail are grave. The injuries sustained by riders who are not killed are severe (Schueler et al, 1984)

The following description by the regional police authority is a typical example:

On June 21, 1985 an accident occurred in the area of Ludwigsburg, near the city of Rosswag. According to the Police the cause of the accident was speeding and lack of driving experience. The motorcyclist was thrown over the guardrail and suffered severe injuries. The 16-year-old female pillion rider slid over the road surface and crashed into a guardrail post. She was killed in this accident by fracture of spine. The deformation of the guardrail post showed the power of the impact.

Solution of the Problem

Because of the great statistical and traumatological significance of motorcycle accidents involving rider impact on guardrail, the Institut fuer Zweiradsicherheit (Institute for Motorcycle-Safety) commissioned a



h	b	s	t
mm	mm	mm	mm
100	55	4,1	5,7
140	66	5,7	8,6

Figure 1. "IPE-100" guardrail post, cross-section

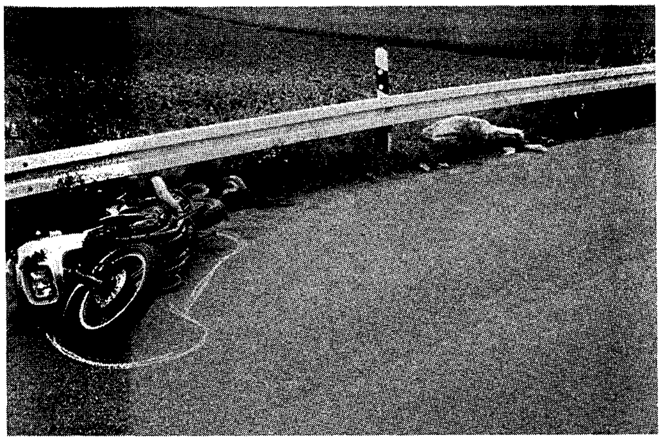


Figure 2. Scene of an accident, deformed guardrail post

research project the task of which was not merely to supply an exact definition of the problem, but also to develop possible solutions.

As a result the following possibilities for improvement were put forward:

- "Sigma-post"
  A "Sigma-post" differs from an "IPE-100 post" at least from a frontal view by being less aggressive owing to its form.

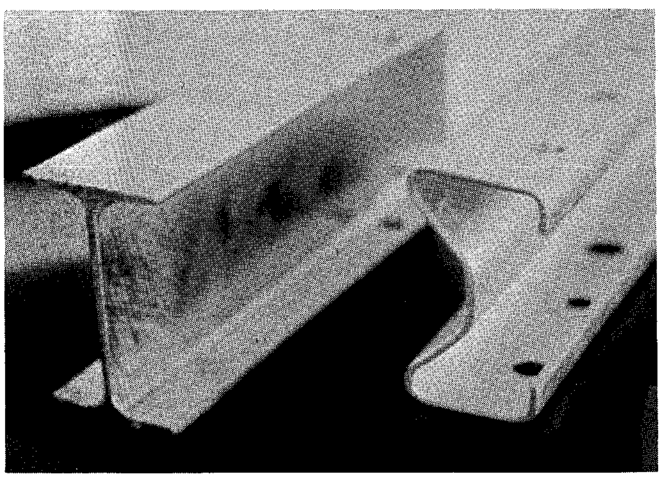


Figure 3. "Sigma-Post" (right) compared with "IPE-100-Post"

- Additional W-beams
  The lower, additional beam may be assembled in a fixed position and directly under the upper beam, or with a gap to the upper beam and springy. By this the lower beam is within given limits—able to compensate energy by replacing a point impact with an area impact.
- Enveloping of posts with special impact attenuators. These impact attenuators, which were specially developed for the protection of

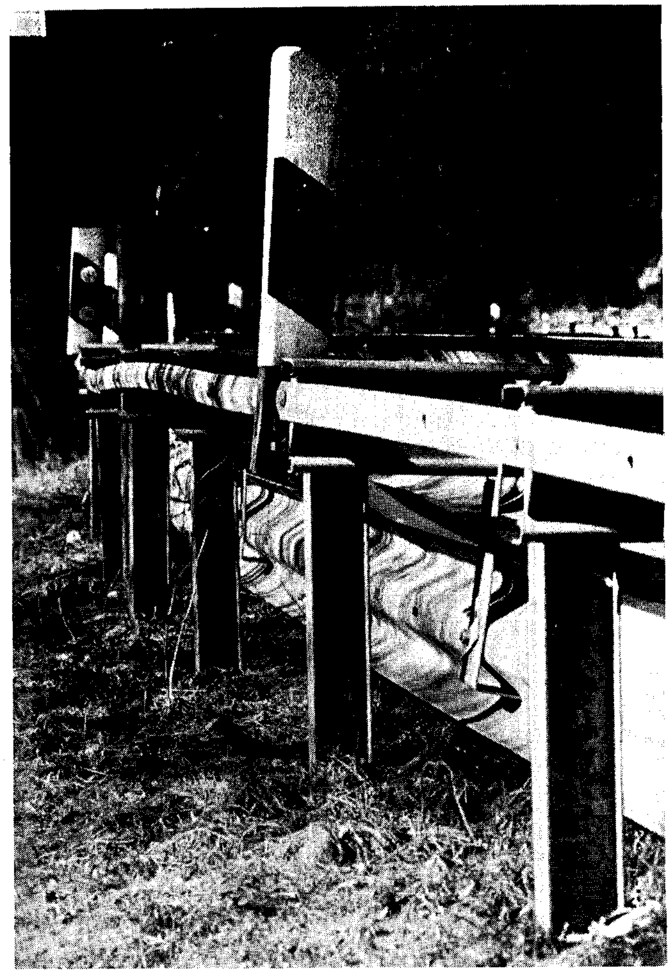
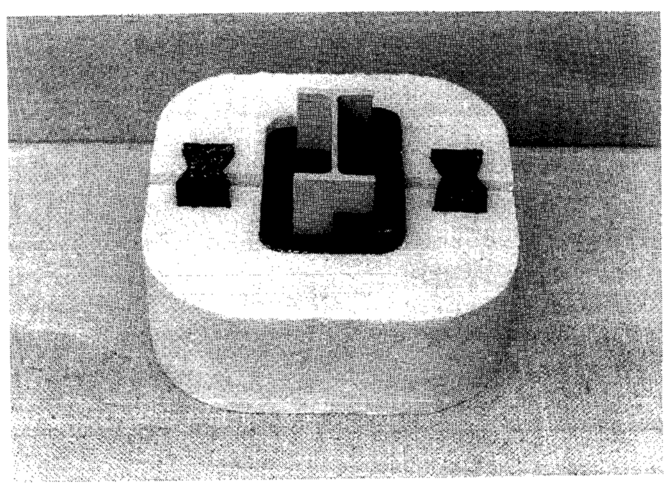


Figure 4. Additional W-beams

motorcyclists, consist of various synthetic materials. They have two tasks concerning the reduction of injury severity: first, to extend the impact surface and second, to absorb energy by distortion.



Picture 5. Cross-section view of impact attenuator fitted to post

Inspection of Effectiveness of Impact Attenuators

In order to ascertain the effectiveness of these impact attenuators guardrail crash tests were carried out in two projects by Schueler (1985) and Jessl (1985).

At the Institut fuer Rechtsmedizin, University of Heidelberg, an impact attenuator consisting of closed cell polyethylene foam with the brand name of "Neopolen" a 1 mm polyurethen outer coating was tested. The basic density of the polyethylene was 30 kg/cbm. Post mortem test objects were used to find out the reduction of injury severity. (Schueler 1985).

Jessl tested a polystyrene impact attenuator with a density of 22 g/1 and a 1 mm outer coating made of polyurethene based paint. He used a Sierra Hybrid II/Part 572 dummy as test object.

In both tests the test objects hit the "IPE-100" guardrail posts, which were fitted with the aforementioned impact attenuators with the inside of an extended arm. Corresponding tests were carried out with uncovered posts.

Jessl also carried out head crash tests.

In each case the impact velocity approximately was v = 32 km/h.

In the corresponding test with the uncovered "IPE-100-post", Schueler established a sub-total traumatic arm amputation (Maximum Abbreviated Injury Scale Value = 3) near the shoulder, whereas the collision with the covered post caused only minor injuries (MAIS = 1).

In the corresponding head crash test Jessl established an impact force of 9410 N and a maximum delay of 1214 m/sec2. The figures for the post with impact attenuator were 18080 N and 2500 m/sec2.

Schueler also carried out crash tests with "Sigma-posts" under the same conditions. These led to a reduced severity with the uncovered "IPE-100-post", but the results of injuries were worse than with the covered "IPE-100-post". According to the findings the injuries were MAIS = 2.

In summary, one can say that the effectiveness of the tested impact attenuators has been proved—both traumatologicaly and according to the recorded measurements. A considerable reduction in the severity of injuries sustained from impact on guardrail can be assumed.

Implementation of Protective Measures

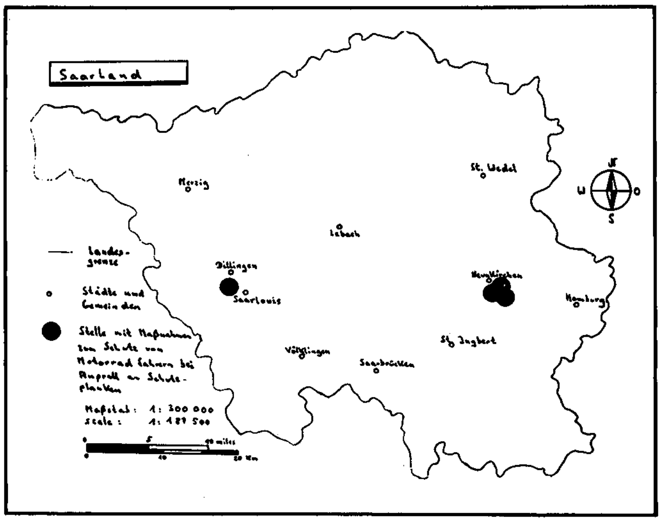
Based on these positive evaluations the authorities passed the impact attenuator and recommended its installation at accident black spots. (Administrative regulations concerning impact protection for motorcyclists issued by the Secretary of the Interior, Baden-

Picture 6. X-ray

Wuerttemberg, reference 7/4562/42, on November 17, 1986) In this respect petitions and other measures to raise public awareness employed by motorcycle associations and the Institut fuer Zweiradsicherheit were very helpful (Domhan, 1987 b).

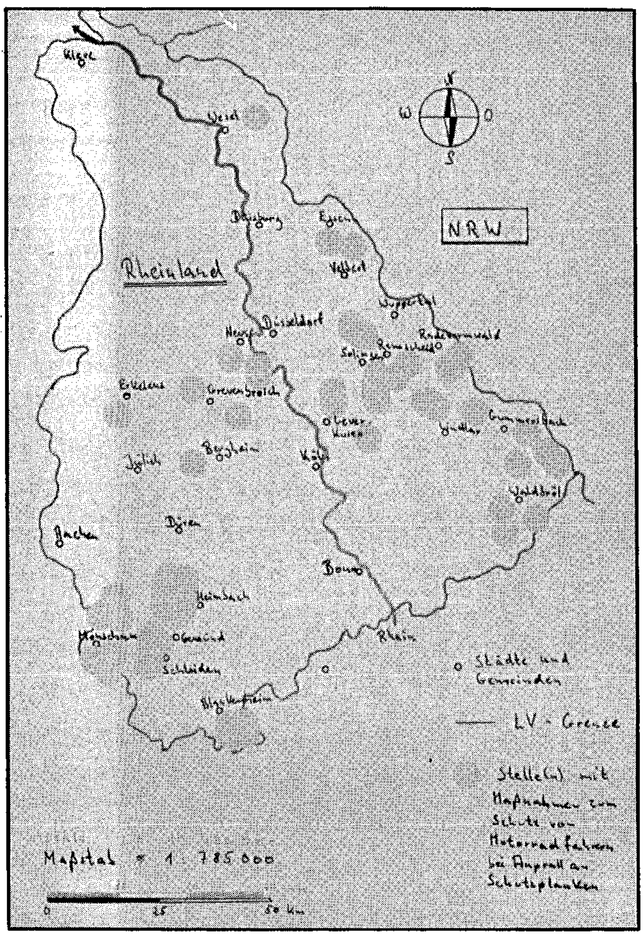
The finding of suitable road sections proved to be difficult. The criteria used by each regional administrative body vary. A widespread survey by the Institut fuer Zweiradsicherheit in 1987 showed that in most cases the initial step is to analyze the available accident statistics. The second step is to decide which accident figure per road section necessitates installing impact attenuators. This figure varies between one and more than two accidents per road section per year.

In some cases the authorities only became aware of problematic sections because motorcyclists themselves came forward and reported them. Not only are there differences in the process of selection, but also in the achieved density of implementation per road section. The differences between the various federal states can be seen in the following maps which were drawn up within the framework of our project. 7 of 11 federal

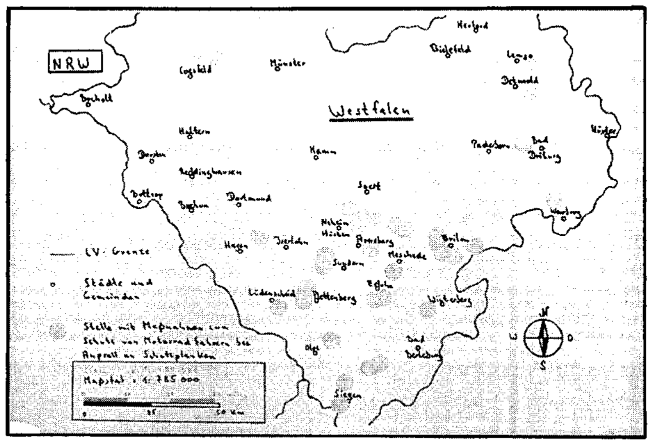


Picture 7. Federal state of the Saarland

states which received our questionaire actually replied. 6 federal states have installed impact attenuators and 2 of these have fitted additional W-beams. At present a total of 20,000 impact attenuators have been installed on West-German roads.



Picture 8. Federal state of North Rhine Westfalia I



Picture 9. Federal state of North Rhine Westfalia II

In some areas, for example in the federal state of Schleswig-Holstein, there was no above-average concentration of motorcycle accidents on guardrail road sections and therefore impact attenuators were not fitted. Pressure from the public did however lead to the decision to use the less aggressive "Sigma-posts" on all new road sections. This policy has also been adopted by all the other federal states which we interviewed!

Initial Examination of Protective Measures in Practice

The present findings of our investigation vary considerably concerning a reduction in the severity of injuries resulting from the implementation of protective measures.

Empirical results from an observation in the field on "Sigma-posts" (which have already been fitted) are not yet available. Therefore one can only assume that they are less dangerous, as established in the biomechanical tests. (Schueler 1985)

However the positive effect and success of the additional W-beam can be verified statistically. According to statements by the police authorities responsible a considerable reduction in both the number of accidents and the accident severity was achieved at 2 well-known accident black spots examined by us near the Nuerburgring. Whereas the reduction in accident severity is something which was expected from the modification, the drop in the number of accidents is a remarkable additional effect. The strong optical impression which a W-beam guardrail has on the motorcyclists obviously has the effect of a warning signal which influences their behaviour on the road positively.

A valid and reliable evaluation of the effectiveness of impact attenuators fitted at accident black spots since 1984/85 cannot yet be achieved due to this short period and the consequently small number of cases.

However, there are very strong indications in favour of these measures according to statements by the responsible police authorities.

Reports often show that the number of registered major accidents has in some cases dropped to zero. In several cases minor damage to the impact attenuators clearly resulting from crashes was registered although no accident had been reported. The police conclude from this that there must have been a collision without serious injuries which without the impact attenuator, would have led to major injuries and consequently to the accident being reported.

However, meaningful, empirically sound results cannot be expected within the next two to three years.

The fact that most departments expressed themselves positively and intend to increase the fitting of protective measures can also be seen as an encouraging sign.

Cost-Benefit Analysis of Guardrail Safety Measures

In order to specify the perspective for further installation of additional W-beams and/or impact attenuators the Federal Highway Research Institute has already carried out cost-benefit studies. (Domhan, 1984, p. 10-15) These studies revealed that modification by all guardrail sections is not sensible. The following table by Domhan shows the estimated cost-benefit factors for this proposition:

	Additional W-beam	Impact Attenuator
Motorways	0.2	0.3
Interstate Highways	0.5	0.7
State Highways	0.3	0.3
Country Roads	0.2	0.2

Alternatively he works on the assumption that 20%, 30% or 40% of all accidents involving the collision of riders with posts/guardrails occur on only 10% of the total guardrail sections. He arrives at the following cost-benefit relations:

	Additional W-beam			Impact Attenuator		
	20%	30%	40%	20%	30%	40%
Motorways	0.4	0.6	0.8	0.6	1.0	1.4
Interstate Highways	1.0	1.5	2.1	1.3	2.0	2.7
State Highways	0.6	0.9	1.2	0.7	1.0	1.3
Country Roads	0.4	0.6	0.8	0.4	0.6	0.8

From a cost-benefit point of view it follows that impact attenuators should have priority over additional W-beams. The valid conclusions reached by the Federal Research Institute for the Federal Republic of Germany are as follows:

- "Fitting all guardrails with protective devices is an uneconomical proposition;
- "On motorways, such additional protective devices can be justified only at accident black spots, representing considerably less than 10% of all guardrail sections. Median barriers can generally be neglected here;
- "On interstate highways, the use of additional protective devices may be justified in the case of a selected number of guardrail sections. This also applies to state highways, but for a still smaller number of guardrail sections.
- "On country roads and unclassified roads, additional protective devices are only justified at accident black spots."

Summary

Motorcycle accidents resulting in a collision of the rider or pillion rider with guardrail posts pose a serious safety problem. This is shown by the high number of major injuries and the fact that 15% of all motorcycle deaths in West Germany are in this category. There are 3 possibilities to solve or minimize the problem, namely the use of "Sigma-posts" instead of the conventional "IPE-100-posts" the installation of additional W-beams and finally, the envelopment of posts with impact attenuators.

Even though a final, empirically sound evaluation has not yet been possible owing to the limited period of practical experience, the effectiveness of additional W-beams has been established and the efficiency of impact attenuators is a justifiable assumption.

Moreover, under certain, realistic conditions the implementation of protective measures is definitely to be considered positive regarding the cost-benefit aspect. This means that the benefit gained is greater than the costs.

A direct transfer of the solutions developed by us to other countries is certainly not possible. But, on the other hand it does seem a reasonable proposition for people working in accident research to examine whether motorcycle accidents involving guardrails occur in statistically relevant numbers to justify and necessitate measures to increase motorcyclists' safety. Furthermore, accident researchers would face the task of monitoring these measures in the field and examining them critically regarding their effectiveness.

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ESM—A Motorcycle Demonstrating Progress for Safety

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Introduction

At the seventh International Technical Conference on Experimental Safety Vehicles in Paris, June 1979 the United Kingdom exhibited their first Experimental Safety Motorcycle ESM 1(1). This vehicle was based on a Triumph 750cc motorcycle and incorporated six features, which were in the prototype development stage, to provide solutions to the problems of motorcycle safety, derived from a number of studies since 1974.

For the occasion of the tenth International Conference on Experimental Safety Vehicles in Oxford, July 1985 the United Kingdom exhibited the second Experimental Safety Motorcycle ESM 2(2). This vehicle was based on a BMW 800cc twin cylinder motorcycle to Police specification. ESM 2 had seven safety features and showed the progress made to those originally fitted to ESM 1.

On the occasion of the eleventh International Safety Conference on Experimental Safety Vehicles, the United Kingdom now exhibits their third Experimental Safety Motorcycle ESM 3. (Figure 1.) This vehicle is based on the Norton Interpol II rotary engine motorcycle nominally of 600cc. ESM 3 has seven safety features and shows the progress of development from the prototype to the production stage for most of the features which can be incorporated on machines for daily use. ESM 3 is considered to be the last Experimental Safety Motorcycle in the present series which have been based on large machines.

Brakes

Earlier work at the Transport and Road Research Laboratory (TRRL) had provided a solution to the problem of inconsistent braking (3) in wet weather and in response to regulations requiring new motorcycles to be tested with the brake system both wet and dry the problem has largely been overcome. This is not the case with replacement pad linings where no such requirements exist and users may have inconsistent brake performance if they fit replacements of a lower standard. During the normal expected life of a motorcycle several sets of replacement pad linings will be fitted, which means that the braking performance could be significantly affected during a large part of the life of a vehicle. This situation can only be remedied by the legislative authorities introducing replacement lining controls.

Anti-lock brakes are another essential feature to improve the safety of motorcycles. The TRRL has been carrying out research work leading to the application of anti-lock brakes to motorcycles for over twenty years. This was summarized at the Tenth Conference on Experimental Safety Vehicles in 1985(4). The anti-lock system fitted to ESM 1 was based on a car system using electronic control which had been modified and adapted for motorcycle use by TRRL. ESM 2 was fitted with a mechanical system by Lucas Girling which was fully developed and tested. ESM 3 also has this system fitted. The three experimental safety motorcycles from the United Kingdom have all provided the rider with continuous information that the anti-lock system is functioning by driving the vehicle speedometer from the anti-lock sensing system. Failure of the speedometer indicates failure of an anti-lock unit and this can be backed up by either audible or visual warnings to the rider. The system used on ESM 3 results in zero reading of the electronic speedometer if either rear or front anti-lock systems fail. We feel that continuous monitoring of the anti-lock device either by this simple system or by some other method is an important feature of a total anti-lock device which will provide riders with increased confidence in these advanced braking systems.

Although those organizations who have been involved with the development of motorcycle anti-lock systems would have carried out their own in-house testing and evaluation, nowhere other than in the United Kingdom have these brakes undergone a field trial using members of the public. Our intention to carry out this trial was announced at Oxford in 1985 when the first motorcycle to go into Police service was included in the United Kingdom exhibition. The field trial consists of a number of motorcycles fitted with the Lucas Girling system in Police service together with a machine in use with a Courier company which is carrying out a high mileage in routine service. This is backed up by further machines for research and instructional purposes together with a single machine to cover short term loans outside the Police and Courier use.

The trial is now half way through the evaluation period and results are very encouraging. There have been some problems which were mostly due either to ancillary components or to the motorcycle itself, particularly in relation to suspension characteristics. The results to date show that the concept of introducing reliable anti-lock on motorcycles has been fully justified. The full report of the field trial appears in the proceedings of this conference (5).

Conspicuity

It is now accepted that a major factor in accidents involving motorcycles during daylight is the failure of road users to see motorcycles in time to avoid a collision. Research aimed at reducing the problem has resulted in suggestions being made to riders on how to improve their conspicuity by both rider-based and machine based features(6). As for the previous Safety motorcycles, ESM 3 is fitted with two daytime running lights as an effective machine based option to improve daytime conspicuity. These daytime running lights also have the advantage that their combined wattage is lower than that of the single headlamp. It has also been shown that this lamp itself is an effective conspicuity aid.

The evidence from accident studies suggests that, as in daylight, a significant proportion of motorcycle accidents in nighttime are associated with some kind of perceptual error on the part of the drivers of the other vehicles. But, unlike the daytime problem where it appears that drivers overlook the motorcycle, the difficulty of seeing motorcycles at night is complicated by errors of identification and interpretation of speed of approach as well as of simple detection.

The results of work by TRRL into overcoming the night time problem are given in "Safety Considerations of Motorcycle Lighting at Night" by Donne and Fulton(8) in the proceedings of this conference. In the United Kingdom all new vehicles, other than motorcycles, have been required to be fitted with dim/dip lighting if manufactured after 1 October 1986 and first registered after 1 April 1987. This may eventually have a benefit for improving motorcycle conspicuity at night. The findings in the TRRL paper should be regarded as a stimulus to the design of forms of lighting which can meet the nighttime requirements of motorcycles and be compatible with the needs of daytime conspicuity.

Leg Protection

Research on protection systems for the legs of riders has been carried out for five years at TRRL. The problems which were regarded as almost insoluble in terms of test techniques and remedial solutions are now approached optimistically as a full understanding of the various engineering features of protection devices has been reached. The work has progressed through a programme of accident and injury investigation, to experimental testing and analytical simulation studies. During the period of this research the TRRL has been keen to keep various organisations in touch with progress and for those interested in this aspect of motorcycle safety a summary of all the relevant documents are available at this conference(9).

The work has covered light, medium and heavy machines impacted into rigid barriers, stationary cars and more recently into moving cars. In all some 55 tests have been carried out and in all cases the resulting dummy injuries in terms of damage to dummy legs and high head velocities have been much reduced with motorcycles fitted with leg protection devices. The trajectory of both the rider and motorcycle are important and it has been found that leg damage increases as the angular velocity in pitch of the motorcycle goes up. The design of the leg protector is all important as this can increase or decrease angular velocity of the motorcycle to the disadvantage or advantage of leg injuries and head velocities.

The current position of the TRRL work on leg protection can be seen on ESM 3 in the form of a safety fairing constructed to TRRL criteria by Norton Motors. This is appropriate for large machines. The research has provided similar criteria for lightweight and medium weight motorcycles. A state-of-the-art has confidently been reached by which designers can use these criteria to produce fairings which should reduce many injuries to riders involved in accidents.

"Protecting Motorcyclists Legs" by Chinn and Hopes(10) reports on work since ESM 2 at Oxford and is available in the proceedings of this conference.

Safety Fairing

ESM 2 was fitted with a safety fairing which had been made from an extensively modified fairing that was available as a factory option. ESM 3 is also fitted with a safety fairing incorporating leg protection, which has been built by Norton Motors to specifications supplied by TRRL and will form the basis of fairings fitted to their machines in the future.

The sectioned fairing on display in the supporting exhibition shows both the energy absorbing cones inside the outer skin panels (removed) and the tubular structure carried forward to the headlamp aperture to provide the second load path which reduces the vehicle pitch on frontal impact. The inner skin panels also provide the crush zone immediately in front of the riders knees and legs.

Other Features

Speedometer

ESM 1 and ESM 2 were both fitted with digital speedometers as this type of instrument can be read quickly by riders. ESM 3 is provided with a conventional dial instrument as this machine represents a move towards a production model. The instrument is however electronic which also provides the continuous monitor to the rider that both anti-lock brake systems are functioning.

Fuel System

Reports on accidents show that 30 per cent of motorcycle accidents involve spillage of fuel from the machines. In one third of these accidents the spillage is reported to be excessive. ESM 2 was fitted with a modified fuel tank which prevents fuel being leaked down the fuel supply lines if the machine falls over on either side. ESM 3 is also fitted with a similar system but the installation has been considerably simplified by using the existing petrol taps and containing their feed pipes on the inside of the petrol tank. This feature is clearly seen on the sectioned components displayed in the supporting exhibition and the principle of operation of this non-spill fuel tank is shown in Figure 2. Designs for both gravity fed and pumped feed fuel systems are available.

Interlocked Stands

Like ESM 2, ESM 3 has ignition interlocks fitted to both stands. The system is now considerably simplified and also provides an override so that the engine can be run for servicing while the vehicle is on the centre stand.

Frontal Impacts

Following the experimental restraint system on ESM 1(1) using a chest pad, attention was given to the sensing and initiation system which could be used on a motorcycle to restrain the rider with an air-bag. Because of the ease of packaging, the air-bag would not have the problems of acceptability associated with the experimental chest pad restraint and the first successful inflatable bag system was shown with ESM 2(7).

Work has continued on this restraint system with both experimental testing and computer simulation. Although the simulation program is not yet fully developed, it indicates that a rider is retained on the motorcycle, following a frontal impact, for about

Table 1. Preliminary estimates of costs and savings for safety features.

	Number of accidents	Estimated cost of accidents £m/yr	Estimated number of casualties saved or with reduced injury severity	Estimated cost of injuries saved £m/yr	Estimated equipment cost £/motorcycle	Estimated cost to fit 250,000 new machines/yr £m/yr	Remarks
Daytime conspicuity	18.000	72(2)	6.000	26	10	2.5	Original equipment or retro-fit
Anti-lock brakes	12.000	52(2)	6.000	26	Large M/C 200 Small M/C 25	32.5	Treatment 150.000 Large 100.000 Small
Leg protection	38.000(1)	90(1)	(5282 serious converted to slight) (7778 slight)	37	75-300	19-75	Safety fairing cost Large machine £300 Medium machine £75(3)
Forward impact	42.000	183(2)	8.500	37	15	3.75	Safety fairing would also have to be fitted

(1) 65% serious casualties
60% slight casualties
(2) Using average casualty cost

(3) quotation from commercial supplier producing from limited tooling.

Using 1983 Casualty Data(1)
Riders and Passengers of Two Wheeled Motor Vehicles

Fatal	963	Cost £'s of Casualties	Fatal	150.045
Serious	20.317		Serious	6.950
Slight	43.214		Slight	170
All severities	64.494		Average all severities	4.370

50ms longer when spoked wheels are used than when cast wheels are fitted. In addition, the use of spoked wheels causes the motorcycle to pitch less. Both these factors are favorable to the effectiveness of an air-bag as a rider-restraint. An air-bag of 120 litre capacity (deployed with one inflator) has been assessed in impacts. It reduced the horizontal component of the rider's velocity to zero at the plane of impact. However, in these tests the motorcycle pitch was restricted to that expected if spoked wheels where used. Therefore, in practice, motorcycles would need to be designed with frontal energy-absorbing characteristics similar to those obtained with wheels of this type.

Although the horizontal component of velocity of the rider was reduced to zero by the presence of the 120 litre air-bag, some rotational energy was retained. This was also the case with the 60 litre air-bag used on ESM 2.

ESM 3 is fitted with a 120 litre air-bag similar to that used in the impact tests. Both the inflator and the bag are located at the rear of the fuel tank. This arrangement has been found to be effective without the need for additional attachments of the bag. With development of the computer program, optional restraint-system characteristics should be predictable. Work will continue with these restraint systems tested on both faired and unfaired motorcycles.

Simulation

Experimental impact testing to assess the effects of changes in features is a necessary part of this type of

Figure 1. E.S

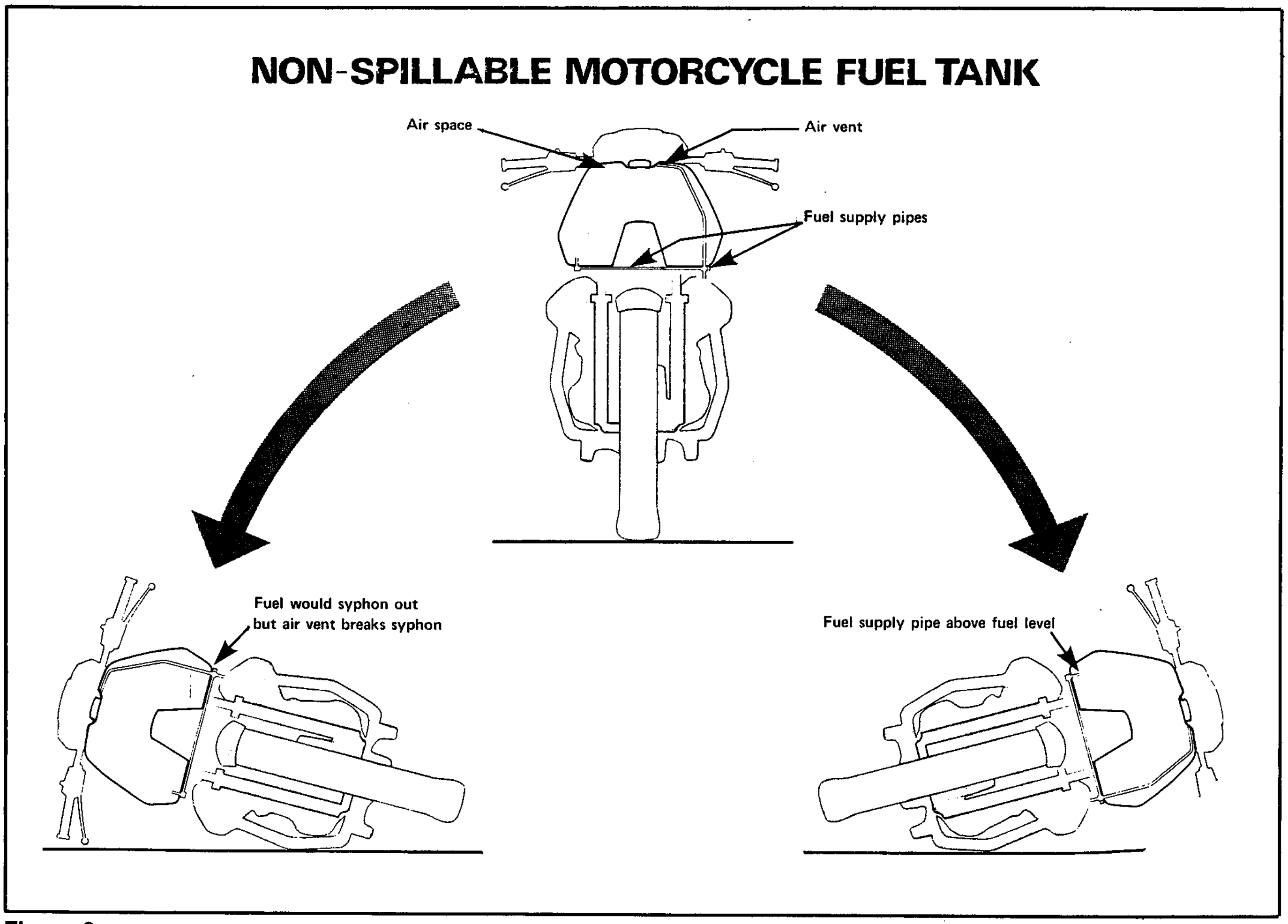

Figure 2

work and is an aid to the preparation of legislative requirements. For research purposes this in itself is too narrow and at best only results in a pass or fail situation for the set of conditions of each test. The work at TRRL now includes mathematical simulation which, when verified by practical tests, produces a design aid. This provides information that would not be available from experimental work alone. ESM 3 is the first of the three safety motorcycles which has benefited from information derived from the simulation technique.

The simulation studies have developed in two ways. For frontal impacts the two-dimensional model looks at the side elevation of the motorcycle and rider. Details and progress on this work are given in the paper "Motorcycle Impact Simulation and Practical Verification"(11) which appears in the proceedings of this conference. A two-dimensional model also exists for studies in plan view(12)(13).

Assessment

Accident and casualty data in Great Britain have shown that motorcycles are ten times more likely to be involved in a casualty accident per mile travelled than are cars (not allowing for rider and driver age differences). This will probably continue unless improvements are made to the roads and vehicles and to the skills of riders. Although the trend of reducing motorcycle mileage continues and this has produced fewer casualties, fluctuations in motorcycle mileage have been seen before and when the motorcycle mileage increases again the increase will contain a larger proportion of inexperienced riders. This will tend to increase the accident rate thus producing a disproportionate rise in casualties. Some preliminary estimates of costs and savings for safety features were made for the Oxford conferences on four of the safety features of the ESM 2 motorcycle. These are still relevant to ESM 3 and are reproduced in Table 1.

It is worth reflecting on the progress of the three safety motorcycles and the benefits which have arisen from exhibiting them at the International Technical Conferences for Experimental Safety Vehicles. The response and interest which has been shown in all three of these safety motorcycles both at the conferences and afterwards indicate that the ESV conferences are the proper forum to exhibit the vehicles and discuss their safety aspects. The progress being made towards incorporating new safety features in production machines reinforces this view.

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Investigation Into Motorcycle, Driver and Passenger Safety in Motorcycle Accidents With Two Motorcycle Riders

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Abstract

Investigations into motorcycle accidents with motorcycle carrying not only driver but also pillion passenger have so far been unknown. However, qualitative observations of accidents indicate that, owing to the mutual influencing of the motorcycle riders, this accident type follows a different sequence from one in which the motorcycle is carrying only a driver.

In experimental accident simulations, motorcycles, carrying two riders, impacted laterally at 90 degrees and diagonally (45 degrees) against stationary passenger cars in the speed range between 50 and 60 km/h.

There is a description of the essential differences in the motion and impact characteristics as well as in head and body postures between driver-only and driver-plus-passenger accidents. Derived from this are safety-related considerations regarding design measures on the motorcycle in order to optimize the motion paths of the riders.

Bases for the reconstruction of this accident type are elaborated.

With regard to the safety of a passenger car and its occupants in the event of motorcycle impacts, the structure of the side assembly and the testing of its strength are discussed.

Introduction

Investigations into collisions between motorcycles and passenger cars with the motorcycle carrying not only a driver but also a pillion passenger have so far been unknown. Qualitative observations of real accidents indicate that, in this accident type, the motion of the pillion passenger differs from that of the driver and that the pillion passenger may cause changes in the collision sequence.

The following results from an initial rough evaluation of 130 accidents with pillion passengers/1/:

Pillion passengers have the same helmet-wearing rate as drivers, suffer less severe injuries and show a lower fatality rate.

As regards the kinematics of pillion passengers, it is found that they reach their final positions in free flight more frequently than the drivers and more frequently undergo the collision without changing direction.

As regards injuries, it may be said that in 35% of cases the driver was injured more severely than the pillion passenger and that in 31% of cases the injuries to driver and pillion passenger were of equal severity. In 30% of cases the pillion passenger was injured more severely than the driver.

As a continuation of joint series of tests of DEKRA Accident Research and the Vehicle Safety Department of Adam Opel AG, collisions were simulated for the first time in which motorcycles carrying two dummies impacted against the sides of stationary passenger cars, with head- and body-injury data being measured. Following on from earlier tests/2,3/and in order to evaluate the test results with pillion passenger, tests were also conducted in which the motorcycles had only a driver.

The determination of the impact events, the motion sequences of dummies and motorcycles as well as their decelerations in the course of the collisions contribute towards indicating motion paths and collision protection of the riders for a given standard of design and towards discussing possible improvements.

With regard to car-occupant protection in the event of the impact of motorcycles and riders, there is a discussion of the structure and strength of the door-side assembly on passenger cars.

Finally, new findings can be gained regarding the reconstruction of this type of accident.

Test Setup and Procedure

The tests took place in the Safety Center of Adam Opel AG, Rüsselsheim. In a total of eight tests, Figure 1, motorcycles impacted in the collision-speed range between 50 and 60 km/h against stationary passenger cars of the latest Opel Kadett range in the saloon version with four doors and in the hatchback version with three or five doors. Six tests, of which five were with driver and pillion passenger on the respective motorcycle, served to simulate ninety-degree motorcycle impacts against the driver-side or front-passenger-side door of the passenger car. The equivalent accident type with a 45-degree motorcycle impact was performed once with and once without pillion passenger.

An already repeatedly described acceleration sled/2,3/was used to accelerate and guide the motorcycles together with the riders represented by 50% hybrid II pedestrian dummies. The sled was braked just before the point of collision, whereafter, owing to the mass inertia, stabilized by the gyrostatic moments of the motorcycles, the motorcycle with the riders moves on at almost constant speed up to collision with the passenger car which has been positioned in accordance with the desired collision configuration.

Three high-speed cameras (frame rate 800 frames per second) were used to document the collision events. Two of these cameras with fixed recording directions, horizontal and at ninety degrees to the motion of the motorcycle before the collision and vertical onto the collision, provided film sequences suitable for in-depth motion analyses. The third camera was swung horizontally during the test procedure in order to provide additional observation of the collision events.

A digital clock (1 ms resolution) positioned in the recording areas of the horizontally directed cameras was started by an electric contact sensor on the passenger car precisely at the commencement of impact of the front wheel of the motorcycle and made it possible to allocate precise times to the individual film frames, Figure 2.

To measure the decelerations of the riders, in the case of motorcycles with two riders, the dummy representing the pillion passenger, and in the case of motorcycles with one rider, the dummy representing the driver was equipped with triaxial acceleration sensors in head, chest and pelvis. The associated measured signals were fed via cables into an evaluation computer, the output unit of which provided analog recordings of the respective deceleration components as well as the resulting decelerations up to 300 ms after the commencement of motorcycle impact and associated biomechanical data for head injury (HIC) and chest injury (SI).

After each test, the passenger-car and motorcycle damage was documented photographically and the vehicle final positions, motorcycle-wheelbase reductions and indentation depths on the passenger car were measured.

The dummies wore protective leather clothing and a new helmet for each test.

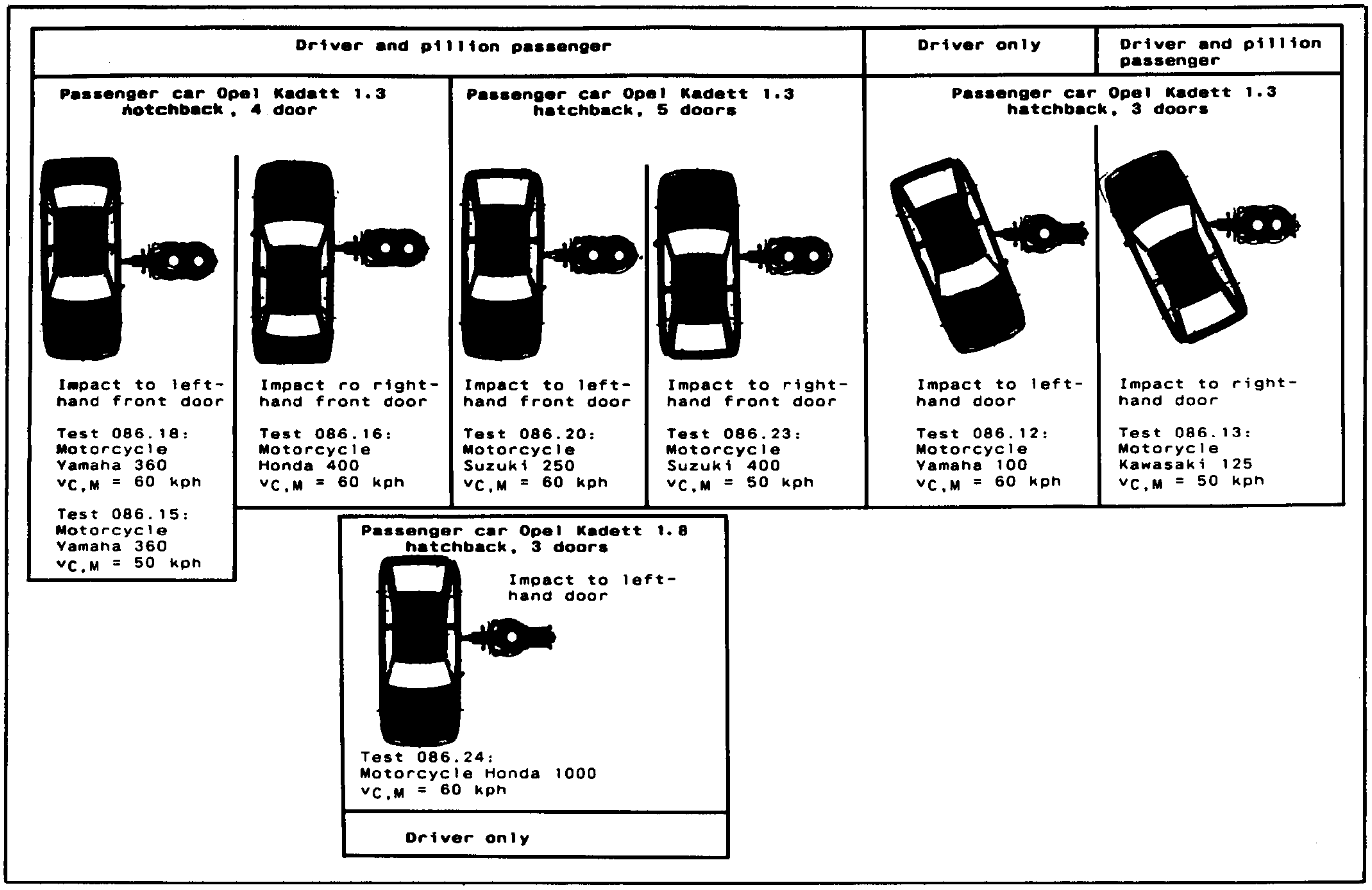


Figure 1. Summary of tests

Test Results

Events and Time Sequences of Motorcycle/ Rider Motions after Start of Collision

Typical events of motorcycle/rider motion after initial contact of the front wheel of the motorcycle with the passenger car and the time sequence of these events are shown in Figure 3. Shown also for comparison are the results from six earlier tests in which motorcycles with driver only impacted against the left-hand or right-hand doors of stationary passenger cars of the Opel Monza model/2,3/. For a total of 12 tests in which the motorcycles impacted at ninety degrees at collision speeds between 50 and 70 km/h against the doors of stationary passenger cars we can thus in all cases see very similar time sequences of the individual events with regard to the motions of motorcycle/rider as of the start of the collision:

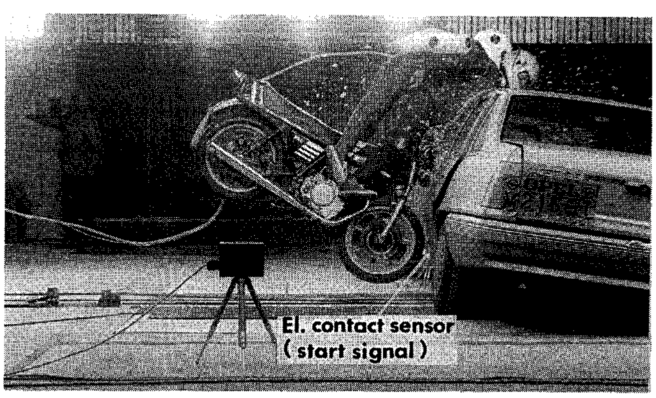


Figure 2. Digital clock for allocation of precise times to individual film frames

Start of front-wheel fork deformation, contact of front wheel with motorcycle engine block, contact of steering head/lamp of motorcycle with passenger car, upward motion of motorcycle rear wheel.

The major forward motion of the motorcycles during penetration into the passenger car is completed approximately 60 to 120 ms after the start of collision.

There are no clearly recognizable differences in the order or in the time sequence of these motion characteristics between motorcycles with and without a pillion passenger. Conversely, in the two 45-degree collisions, it was possible to establish a reversal of the order of events between contact of the motorcycle front wheel with the engine block and contact of the steering head/lamp with the passenger car. This is due to the fact that in these collisions—similar to 90-degree motorcycle collisions against the sides of mov-

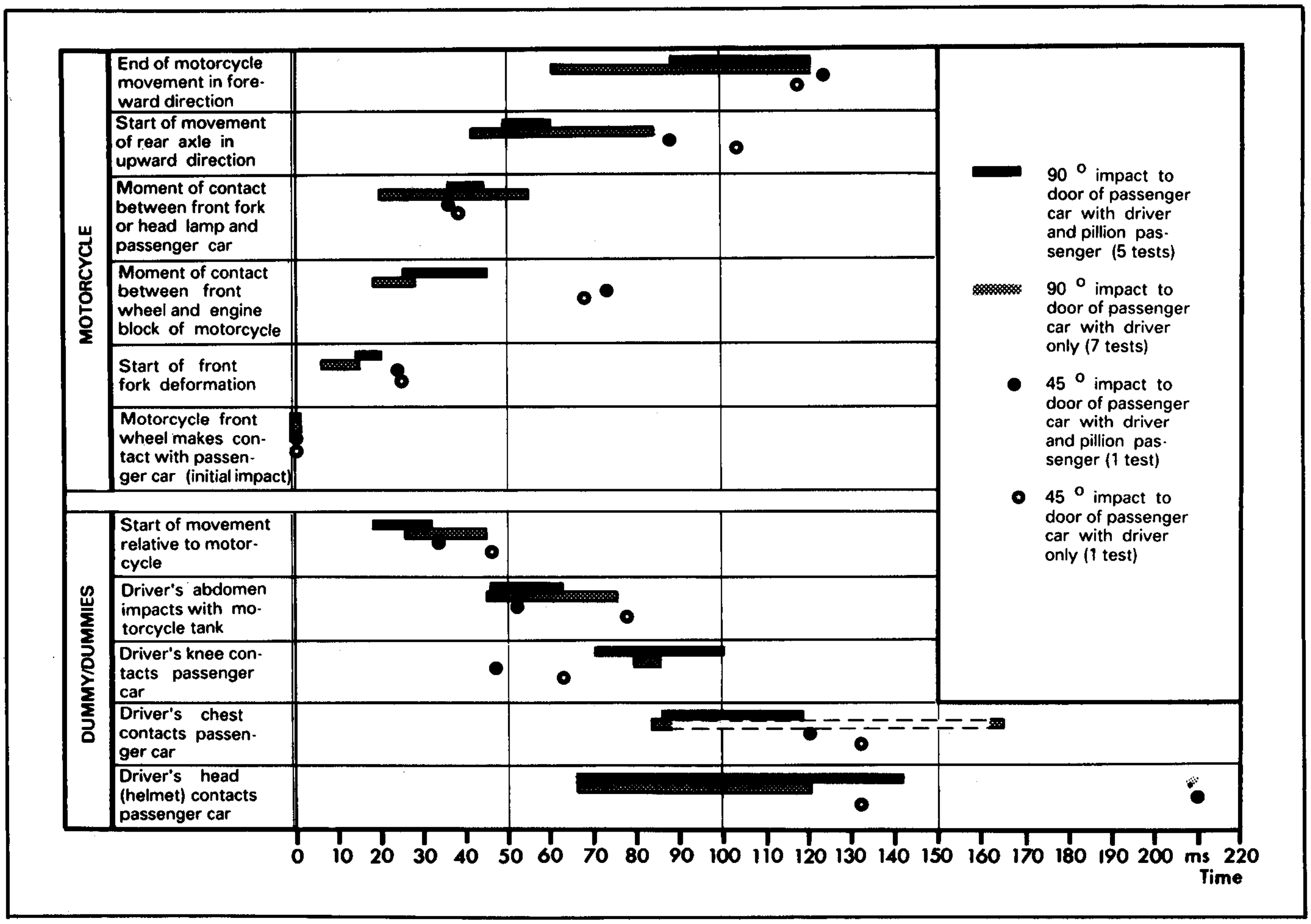


Figure 3. Events and time sequences of motorcycle and dummy motions for motorcycle impact at speeds between 50 and 70 km/h against stationary passenger cars (90-degree impact against passenger car without pillion passenger on motorcycle supplemented by the results from six tests from an earlier series of tests/2,3/)

ing passenger cars/4/—the front wheel is deflected after contact with the passenger car, so that the motorcycle engine block does not come into contact until later with the motorcycle front wheel.

The typical motion of the rider/riders is characterized initially by the start of a relative motion between riders and the already decelerated motorcycle with subsequent impact of the torso of the driver against the motorcycle tank. In the tests with pillion passenger and upright seating position of the driver, these two events were followed always by a knee impact and later by a chest impact of the driver against the passenger car. In one test with a pillion passenger and a forward-inclined seating position of the driver, it was possible to establish first of all the chest impact, then the knee impact of the driver against the passenger car.

In the tests in which the motorcycle had only a driver, there was not always a knee or chest impact, since, depending on the seating position of the driver and the shape of the motorcycle handlebar, the forward motion of the dummy was already so far diminished either on head impact or on impact of the thighs upon the motorcycle handlebar that knees or upper body did not come into contact with the passenger car in the direct sequence of collision events. With regard to knee impact in the two tests with 45-degree collision, it is striking that, owing to the shorter distance between the knee facing the side of the vehicle and the body of the passenger car, the impact of the driver's knee took place earlier, approximately simultaneously with the impact of the torso on the motorcycle tank.

In the tests shown, the helmeted head of the driver dummy impacted at the earliest approximately 65 ms after the start of motorcycle impact against the passenger car. Also in the tests with pillion passenger, the seating position of the driver was decisive with regard to the order in which chest and head impacted.

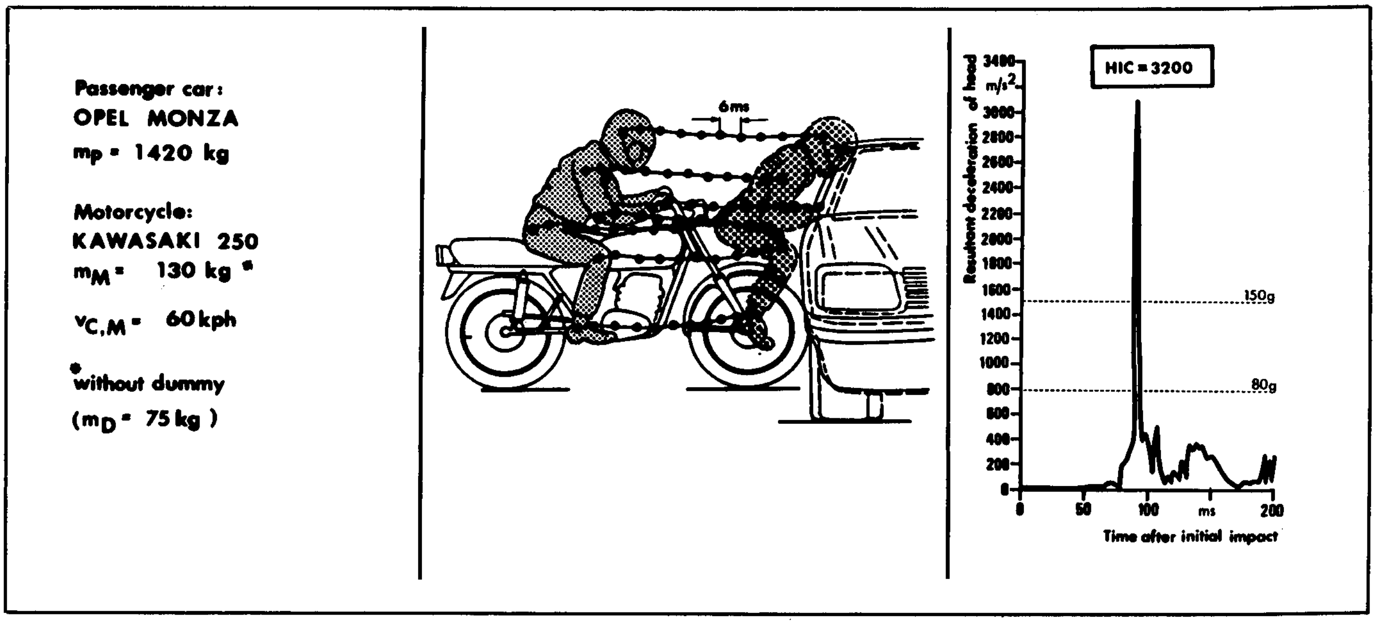


Figure 4. Dummy trajectories and resulting deceleration in the head of the dummy for 90-degree motorcycle impact against the door of a stationary passenger car (test from/1,2/)

Trajectories and Measured Decelerations of Rider Motions

Ninety-Degree Motorcycle Impacts in Passenger-Car Door Area, Motorcycle without Pillion Passenger. In the earlier tests with motorcycle without pillion passenger and with ninety-degree motorcycle impact in the door area of stationary passenger cars, hypercritical injuries to the helmeted head of the dummy were established during impact against stiff parts of the roof edges of the passenger cars/2,3/. Deceleration peaks between 2340 and 3400 m/s^2 and HICs in the range between 1837 and 3209 in the case of direct helmet impact against the roof edge of the passenger car, Opel Monza model, suggest that the main attention should be focussed on the further development of the potential of protective helmets. Figure 4 shows an example.

A test conducted for purposes of comparison in the current series of tests and involving a motorcycle with driver only, traveling at the same collision speed and with virtually identical head impact as in Figure 4, resulted in a considerably reduced head injury with an HIC = 561, Figure 5.

Subject to the exact reproducibility of head impacts in real accident simulations, this low HIC value is certainly also attributable to the less aggressive shape and greater flexibility of the roof edge of the latest version of the Opel Kadett in comparison with the Opel Monza. This points to the benefit of measures on the vehicle to protect two-wheeler riders in the event of a collision.

As already in the earlier tests, the decelerations shown for the chest and pelvis in Figure 5 are at a considerably lower level than that for the head and are to be classified as subcritical. Directional changes in trajectories and concentrations in the target positions shown at 6-ms intervals point to the fact that there are forces acting between the dummy and

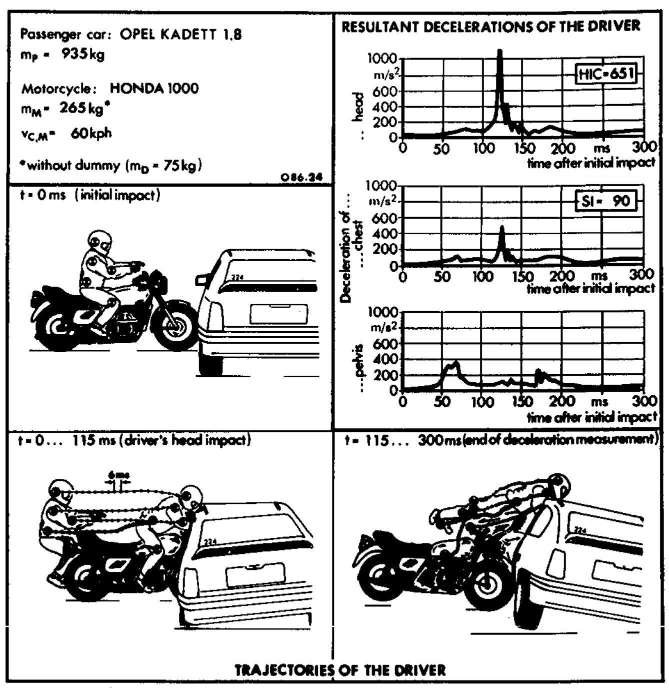


Figure 5. Dummy trajectories and measured decelerations in head, chest and pelvis of the dummy for 90-degree motorcycle impact against the door of a stationary passenger car (test from current series)

motorcycle that decelerate the dummy and/or change its direction of motion. The head of the driver impacts at virtually undiminished velocity.

Typical of the shape of the tank of the Honda 1000 is a raising of the center of gravity of the body with decelerations upon impact on the tank or with the driver's torso sliding over the tank, with the inclination of the upper body being shifted forwards and the trajectory of the head being given a downwards tendency. In a real accident, the supporting of the driver on the handlebar—insofar as this is possible by human muscle power—in conjunction with leg pads fitted directly in front of the driver's knees might compensate for the intensified downwards inclination of the upper body, and the upwards shifting of the center of gravity might also bring about an upwards tendency in the head trajectory, so that the head does not impact directly on the roof edge, but on the softer part of the roof behind the edge.

After the impact of the head, the chest of the dummy impacted on the passenger car in the test shown. In the further course of the collision, there was no knee impact on the passenger car, but a rotation of the upper body accompanied by further upwards motion with stretching of the legs. The velocity of the dummy, the hands of which had smashed through the side window of the passenger car with the result that upper arms and chest were entangled, was thus completely lost. The dummy reached its final position on the ground near the door of the passenger car that had been impacted.

Ninety-Degree Motorcycle Impacts in Passenger-Car Door Area, Motorcycle with Pillion Passenger. In four tests with ninety-degree motorcycle impact against the driver-side or front-passenger-side door of the Opel Kadett and with upright seating position of the driver, the driver's knees, the driver's chest and the helmeted head of the driver impacted one after the other on the passenger car, Figure 6. In a similar test with forward-inclined seating position of the driver, first of all the driver's head, then the driver's chest and finally the driver's knees impacted on the passenger car, Figure 7.

Common to all motion sequences shown in Figure 6 is that the trajectory of the driver's head before impact on the passenger car is not deflected downwards, but remains at least horizontal or is even given a predominantly upwards tendency after the impact of the driver's torso and after the driver has slid over the motorcycle tank. Consequently, the head does not impact directly on the roof edge, but behind it on the softer part of the roof of the passenger car in the region of the chin.

As will be shown, this positive tendency is attributable to the influence of the pillion passenger.

The measured decelerations in the pelvis of the pillion passenger show two more or less pronounced maxima and an initial rise approximately at the time at which the driver's torso impacted the tank or slid over the latter. The first maximum occurs on the impact of the driver's knees on the passenger car when the not-yet decelerated passenger impacts against the already decelerated driver. If the driver slides over the tank, the driver's body is subjected below its center of gravity to forces that in themselves ought to cause an intensification of the inclination of the upper body. This is counteracted by small supporting forces of the arms of the driver dummy, which, in themselves, are not able to compensate for the intensification of the inclination of the upper body.

The upward shifting of the driver's center of gravity and also head trajectory is essentially attributable to the influence of the passenger. Firstly, the pushing action of the pillion passenger's torso reduces the deceleration of the driver's torso as it slides over the tank or as the driver's knees impact. Consequently, the relative speed between the driver's head and torso is not so pronounced and the upright seating position of the driver remains, so that the raising of the center of gravity causes a raising of the head, too. In addition, the pillion passenger slides with its thighs like a wedge under the driver, which has been decelerated as a result of its knee impact on the vehicle and is already slightly raised in the pelvic region. This supports the upward motion of the driver.

A further pointer to the pushing effect of the torso of the pillion passenger is that, in the tests with pillion passenger, there is always an impact of the driver's knee on the passenger car.

The second deceleration maximum of the passenger's pelvis is reached when the driver has impacted also with its upper body against the passenger car and the passenger begins to slide up on the virtually braked body of the driver.

Of interest in this connection are the positions of driver and passenger at the start of deceleration and at the start of the upward motion of the rear wheel of the motorcycle. Since it is intended to use the deceleration of the motorcycle to trigger an airbag, yet the deceleration occurs only at an advanced stage of the collision, the question is raised as to whether the airbag can still afford any protective effect. In addition, it must be clarified whether the observed upward motion of the motorcycle rear wheel and of the motorcycle seat bench in the course of the motorcycle impact can be used to support the upward motion of driver and passenger.

Shown in Figure 8 for all six ninety-degree motorcycle impacts in the door area of a passenger car is the

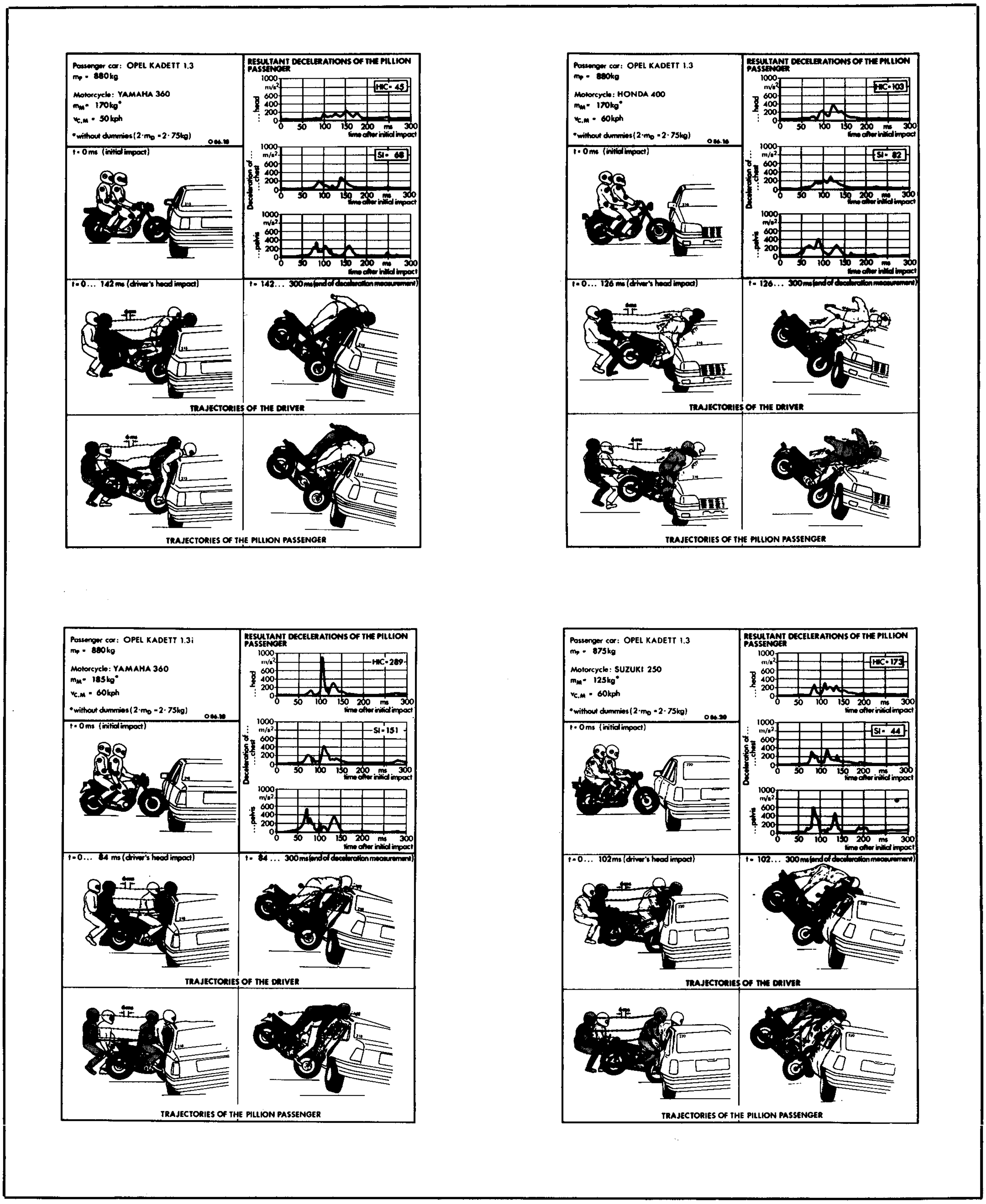


Figure 6. Dummy trajectories and measured decelerations in head, chest and pelvis of the dummy representing the pillion passenger for 90-degree motorcycle impacts against the driver-side or front-passenger-side door of stationary passenger car (driver's seating position upright)

situation at the start of the upward motion of the upper rear-wheel spring mounting of the motorcycle with the associated trajectory from initial impact to 120 ms thereafter.

The concentrations of the target positions, shown in each case simultaneously at 6-ms intervals, start approximately 12 to 18 ms before the start of this upward motion. Thus, in the tests performed, there would be a suitable degree of deceleration for the activation of sensors to trigger the airbag even before the torso of the driver impacts on the tank (at the earliest 45 ms after the start of impact of the motorcycle front wheel, see Figure 3), and, in addition to protection for head and chest of the driver, the impact of the abdomen on the tank could also at least be lessened.

As is also shown in Figure 8, the knees of the driver are still so far away from the passenger car that the effect of the airbag, starting in this situation, might—assisted by knee-pads—even prevent the impact of the driver's knees on the passenger car.

As already stated in/5/, the knee-pad, which is in close contact with the driver's knee, is intended, firstly, to at least greatly decelerate the relative motion between dummy and motorcycle—i.e. the early sliding-forward of the dummy towards the passenger car—and, secondly, to promote the upward motion of the dummy. In conjunction with the airbag, an even better protective effect can be expected for the collision configurations described here.

A rapid increase in the target concentrations can be seen after the start of the rebound of the rear wheel of the motorcycle. The upward motion of the rear sections of the motorcycle—as shown in Figures 5 and 6 for the head impact of the driver—is usually greatly pronounced, but has only very minor influence on the motion of the riders, because they have already slid away forwards out of the area of the seat bench that is effective as regards the upward motion.

45-Degree Collisions. Two collisions between motorcycles and passenger cars in which the collision angle between motorcycle and passenger car was 45 degrees were intended to provide initial findings on phase and motion sequences differing from those for

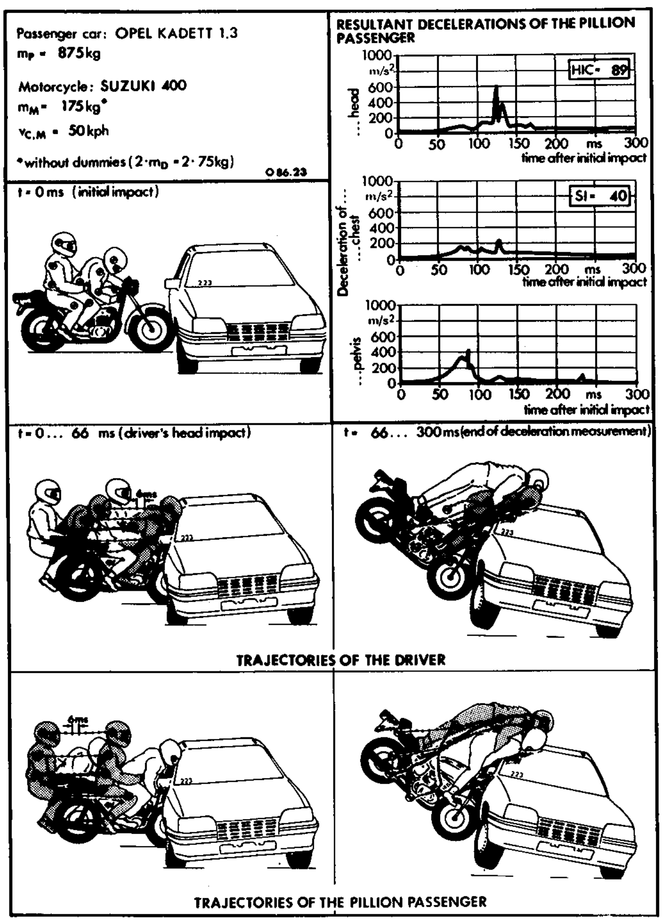


Figure 7. Dummy trajectories and measured decelerations in head, chest and pelvis of the dummy representing the pillion passenger for 90-degree motorcycle impacts against the front-passenger-side door of a stationary passenger car (driver's seating position forward-inclined)

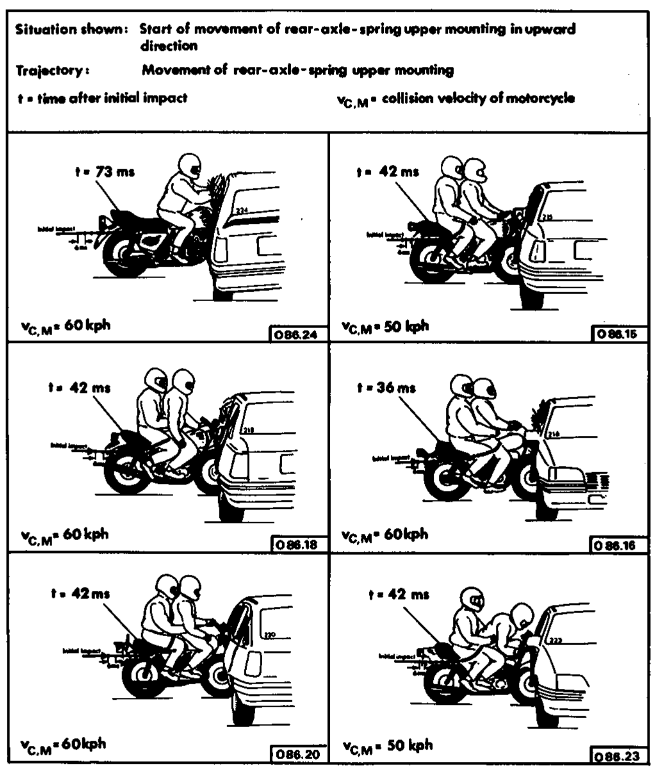


Figure 8. Trajectory of upper rear-wheel mounting on the motorcycle and situation at start of rebound of rear wheel of motorcycle (90-degree motorcycle impacts against the driver-side or front-passenger-side door of stationary passenger cars)

ninety-degree collisions, Figure 9. Approximately the same phase and time sequence applies to both versions of the test (driver and driver/passenger).

In the diagonal collision, after the start of contact, the front wheel of the motorcycle, sliding along the passenger car, is first of all turned until it is aligned parallel to the side contour of the passenger car. Approximately 20 ms after the start of the collision, the deformation of the fork is initiated, this being followed approximately 15 ms later by the contact between lamp/steering head and passenger-car door. Now, the motorcycle front wheel, which in the meantime is in broad-area contact with the passenger car, and the steering-head and lamp unit become entangled with the already deformed door, so that, as a result of the pushing action of the motorcycle, the virtually braked front wheel of the motorcycle comes into contact with the engine block. Approximately 120 ms after initial contact, the forward motion of the motorcycle is more or less completed and, while the rear axle is being raised slightly, the motorcycle folds also with the rear wheel against the passenger car.

The driver's knee facing the passenger car impacts on the vehicle door approximately 50 ms after the start of the collision. The impacts of pelvis and chest take place in the time phase between 50 and 140 ms.

Approximately 130 ms after the start of the collision, the head of the driver (only rider) impacts in a glancing manner on the comparatively soft vehicle roof, with the HIC being relatively low at 173.

The head impact of the passenger is against the back of the driver and is, therefore, heavily damped (HIC = 31). In this collision configuration, an airbag is certainly not able fully to develop its protective effect for the driver. As regards the knee impact of the driver, however, a protective effect can be expected if the knee-guard is so designed that it is also capable of protecting against direct lateral impacts.

Owing to the multiplicity of possible impact angles and passenger-car contours, more detailed findings can, of course, not be expected until after further tests.

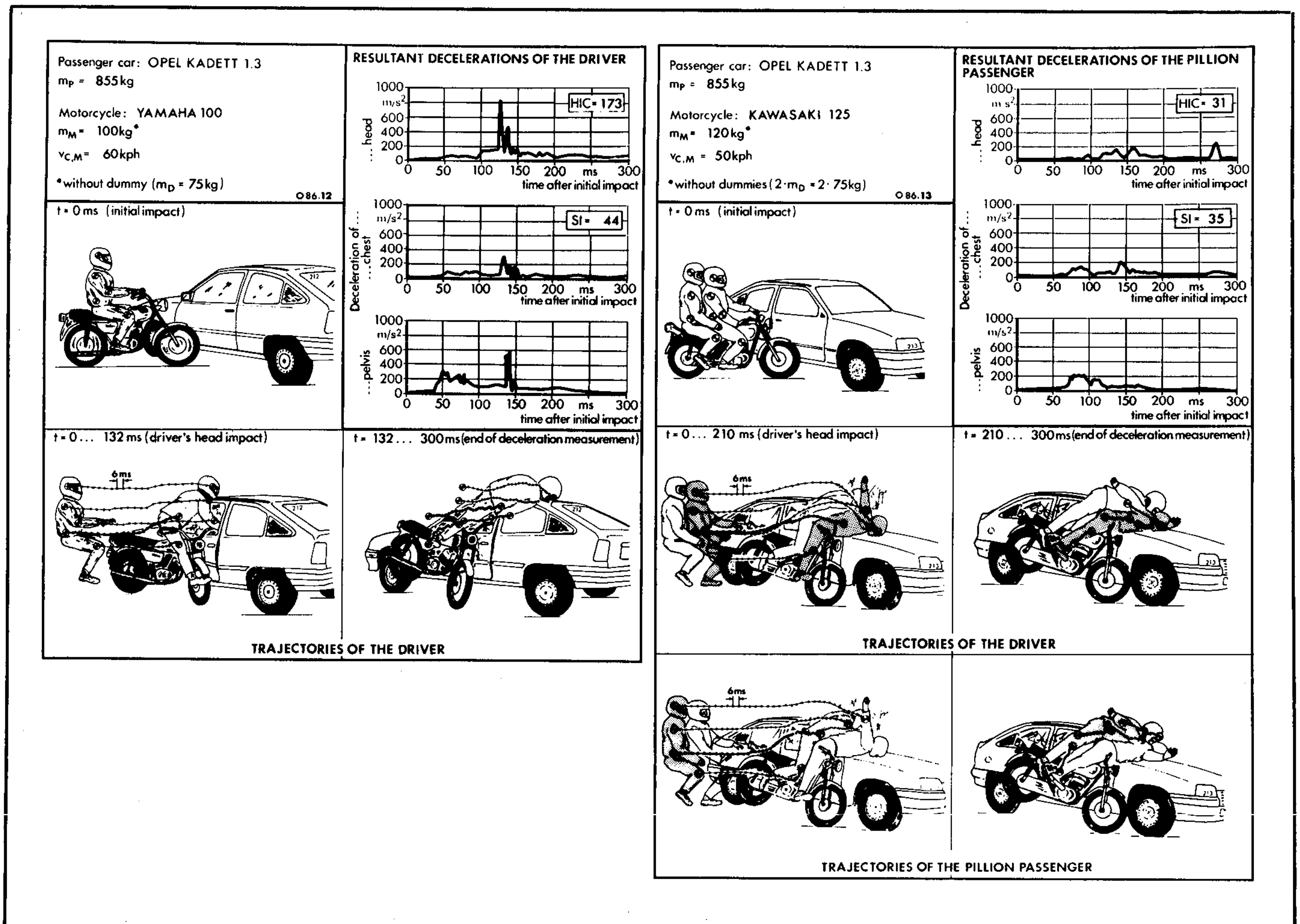


Figure 9. Dummy trajectories and measured decelerations in head, chest and pelvis of the dummies representing the driver and pillion passenger for 45-degree motorcycle impact against the driver-side or front-passenger-side door of stationary passenger cars

Strength of Door/Side Assembly

During the development phase, the side assembly of Opel passenger cars is dimensioned in tests for the impact of passenger cars. The main emphasis is placed on retaining the tensile assembly according to the principle of the chain. Therefore, A-, B- and C-pillars, doors as well as locks and hinges are matched to the loading.

In contrast to passenger cars, motorcycles, particularly very heavy ones (Honda 1000, m = 265 kg, Figure 5), apply concentrated forces to the side of the vehicle. Therefore, high penetration depths are expected.

In the tests presented here, the side assembly stopped the motorcycles in all cases from penetrating into the passenger compartment. A major contribution in this regard was played by the door sill. Owing to the heights of the front wheel of the motorcycle and of the door sill, there was a favorable overlap and thus a reduction in the load on the door. The penetration depths were of the order of magnitude of 300 mm, Figures 10 and 11. The car occupants are at risk as a result of the penetrating door, this being alleviated by the upholstered inside panel of the door. A further risk to the occupants is produced by the penetration of the motorcycle driver, assuming that the latter is in the appropriate (low) seating position.

Taking into account the vehicle class (lower mid-class) it can be said in this regard that there is a good protective effect.

Additional Findings with Regard to the Reconstruction of Accidents Between Motorcycles and Passenger Cars

Practical reconstruction methods often make use of the permanent shortening of the wheelbase on the motorcycle and—if the rider moved in flight—of the distance the driver was thrown. In addition, qualitative comparisons of degrees of damage on the vehicles provide pointers to the possible collision speeds.

All hitherto performed simulations of passenger-car/two-wheeler accidents make it clear that, given the multiplicity of possible motion sequences during the collision, the overall interpretation of degrees of damage cannot lead to the desired goal.

In the case of accidents involving the ninety-degree impact of a motorcycle against the door-side area of a passenger car of the Opel Monza model or of the

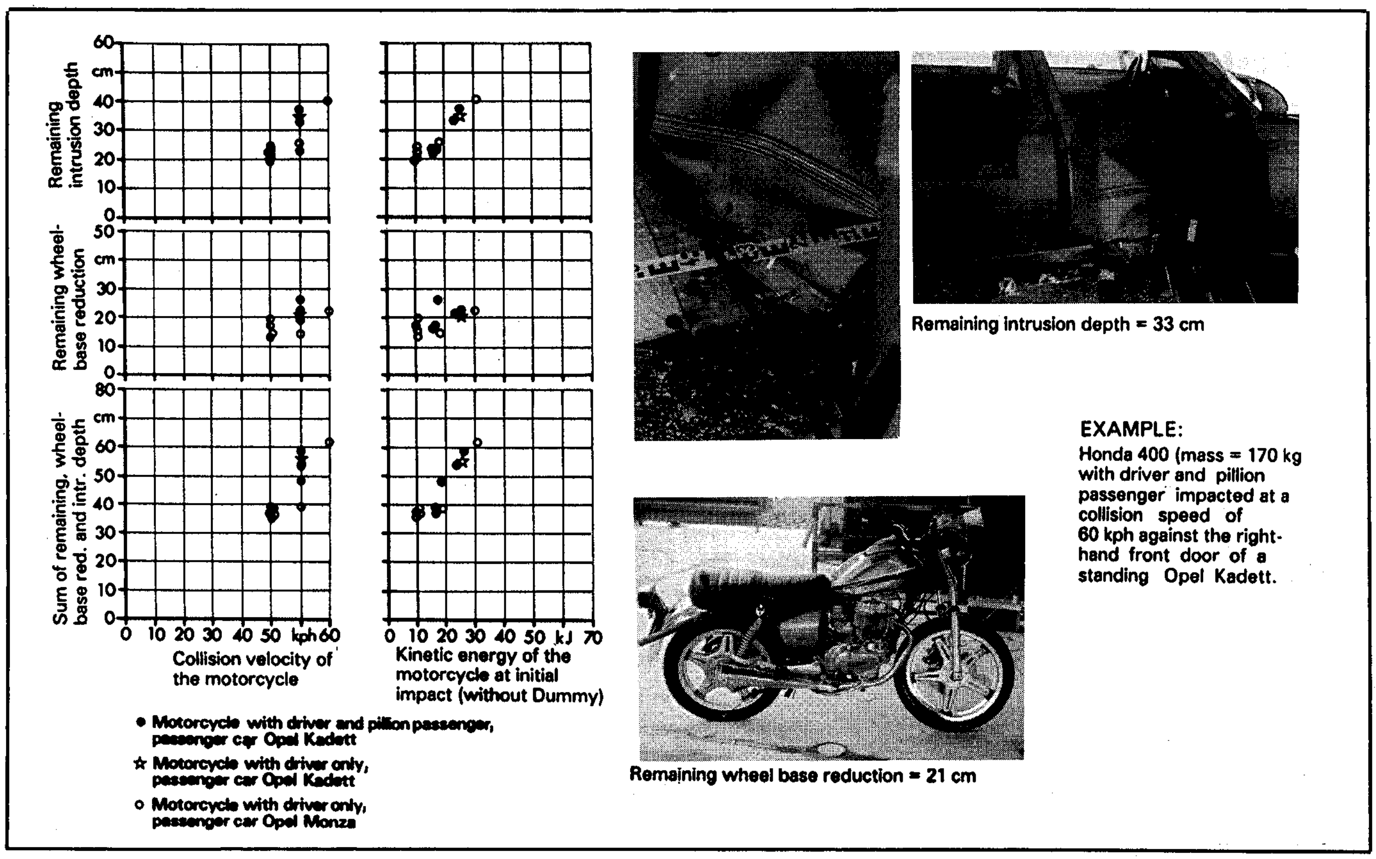


Figure 10. Penetration depth on passenger car and wheelbase shortening of motorcycle as a function of impact velocity and impact energy of motorcycle for 90-degree motorcycle impacts against the driver-side or front-passenger-side door of stationary passenger cars

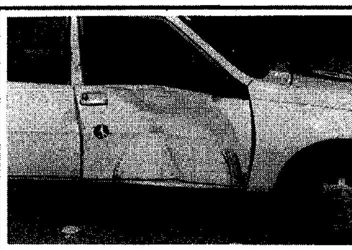
90° IMPACT	Passenger car: Opel Kadett 1.3 $V_{K,P}$ = 0 kph Remaining intrusion depth = 17 cm Motorcycle: Kawasaki 125 m_M = 120 kg $v_{C,M}$ = 50 kph 086.13	
45° IMPACT	Passenger car: Opel Kadett 1.3 $V_{K,P}$ = 0 kph Remaining intrusion depth = 22 cm Motorcycle: Yamaha 360 m_M = 170 kg $v_{C,M}$ = 50 kph 086.15	
90° IMPACT	Passenger car: Renault R 14 $v_{K,P}$ = 23 kph Remaining intrusion depth = 12 cm Motorcycle: Suzuki 125 ER m_M = 93 kg $v_{C,M}$ = 47 kph W85.6	

Figure 11. Different degrees of damage on passenger car after motorcycle impacts in the door area

latest version of the Opel Kadett, it was possible in the impact-speed range between 50 and 70 km/h to detect an approximately linear relationship between the maximum permanent penetration depth on the passenger car—arising from the impact of the motorcycle front wheel—and the impact speed/impact energy of the motorcycle. In contrast, less clear were the relationships between permanent wheelbase shortening of the motorcycle and the latter's impact speed/impact energy. The most favorable relationship for a rough delimitation of possible impact speeds of the motorcycle appears to be that between the kinetic impact energy of the motorcycle and the sum of permanent penetration depth on the passenger car and permanent wheelbase shortening of the motorcycle, Figure 10. In the reconstruction of real accidents of the type simulated, it might be possible in this manner to calculate, first of all with reference to the sum of penetration depth and wheelbase shortening, the kinetic impact energy of the motorcycle and, by means of the known mass of the motorcycle, then the impact speed. It does not appear worthwhile additionally to take account of the mass of the rider, because the deformation on the passenger car arising through the penetration of the motorcycle front wheel is already largely completed by the time the rider is decelerated or by the time of the major impact of the rider or the passenger car. Figure 11 shows, for comparison, three degrees of damage on passenger cars.

Finally, the tests are used also to derive pointers as to the flight tendency of the motorcycle riders. Figure 12 is a compilation of the center-of-gravity trajectories of driver and passenger in the case of ninety-degree impact against the doors of stationary passenger cars from the start of impact to 300 ms thereafter. It should be noted that unhindered flight is not possible in these cases.

It can be seen that the motion of the driver is deflected upwards within a relatively narrow range of between 15 and 20 degrees.

The angular range of the upward motion of the passenger extends from 18 to 45 degrees, i.e. it tends towards steeper lines of motion. This is expressed also in the qualitative observation that the passenger dummy always slides further onto or over the passenger car than the driver dummy, the latter usually after the collision coming to lie near the passenger-car side facing the impact.

The examples indicated here make it clear that, owing to the multiplicity of the possible impact configurations between two-wheeler and passenger car, considerable research remains to be conducted in order to solve these problems.

Summary

As a continuation of joint series of tests of DEKRA Accident Research and the Vehicle Safety Department of Adam Opel AG, collisions were simulated for the first time in which motorcycles carrying two dummies impacted at 90 degrees or 45 degrees against the sides of stationary passenger cars, with head- and body-injury data being measured. The collision speeds of the motorcycles were in the range between 50 and 60 km/h.

The 90 degree impact yielded no clear differences in the order or in the time sequence of the motion characteristics between motorcycles with and without pillion passengers. In the 45-degree collisions, the impacts on the passenger car were predominantly of the glancing type, with the result that there was a change in the degree of deformation.

Compared with earlier tests, lower HIC values were measured for the impact of the helmeted head of the dummy on the roof edge of the test vehicles, because the roof edges of the latest Opel Kadett range used in the tests have a less aggressive shape and are more flexible.

The measured decelerations of the pillion passenger in head, chest and pelvis were always subcritical, because the impact was diminished by the driver.

As a result of its pushing motion, the pillion passenger supports the favorable upward motion of the driver, with the result that the driver's head is

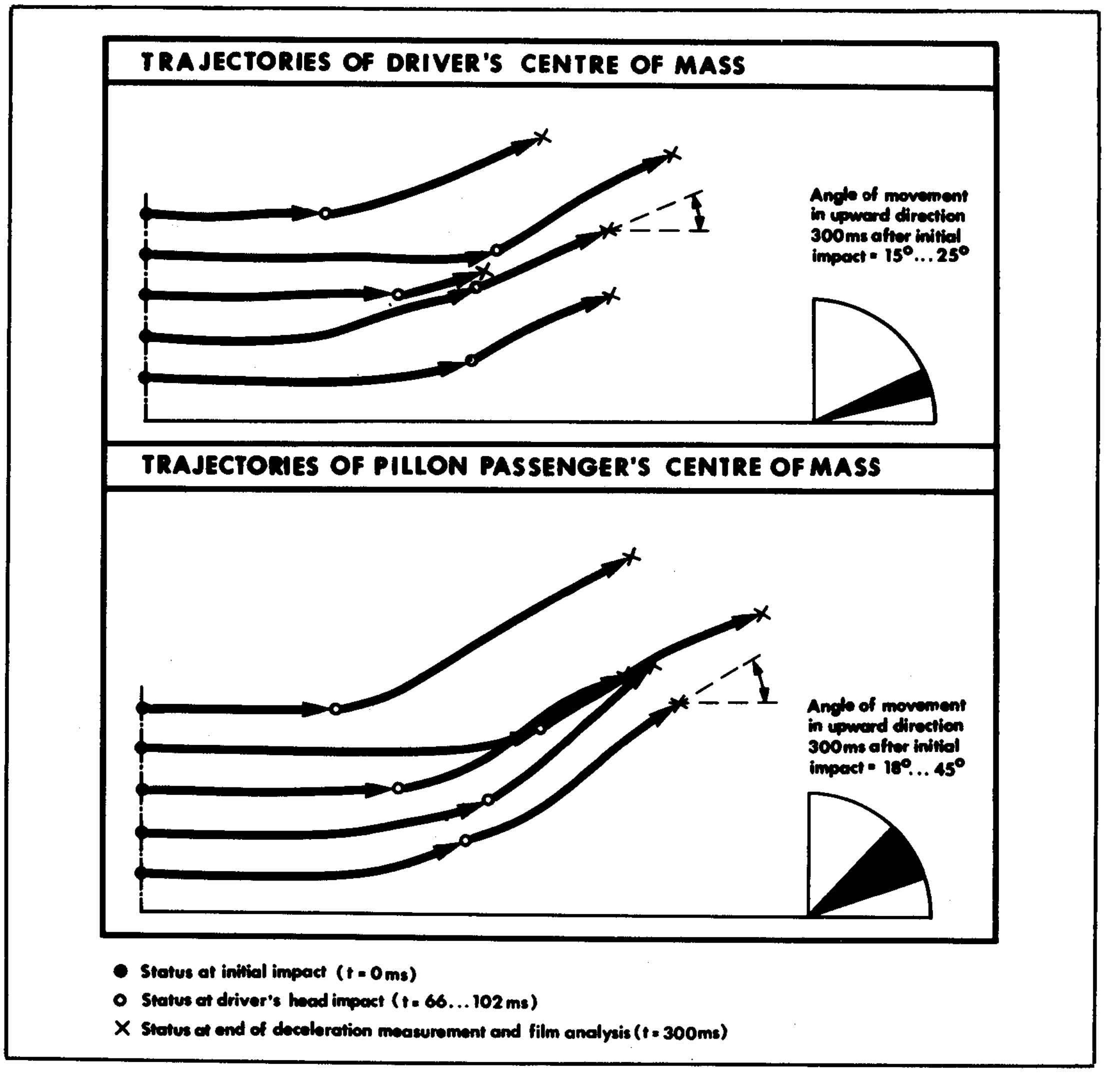


Figure 12. Center-of-gravity trajectories of dummies representing driver and pillion passenger for 90-degree motorcycle impacts against the driver-side or front-passenger-side door of stationary passenger cars

lifted above the critical area of the roof edge. Additionally, the wedge effect of the thighs of the pillion passenger promotes the upwards tendency of the motion of the driver even before impact against the passenger car.

Motorcycle decelerations of sufficient magnitude to trigger an airbag via sensors were measured in the tests even before the driver's torso impacted on the motorcycle tank, with the result that the protective effect of the airbag can be taken into consideration also for the abdominal region of the driver. Closely fitting knee-pads, intended to prevent early sliding-forwards of the dummy on the motorcycle seat bench and to promote its upward motion, can be expected—in combination with a motorcycle airbag—to provide an increased protective effect both for the driver and for the pillion passenger.

In the case of diagonal collisions, the protective effect of the airbag is certainly not optimal, and the knee-pads should also afford lateral impact protection. Further tests may provide more reliable findings in this regard.

The strength of the door-side assembly on the passenger car has proved to be adequate in the case of a concentrated motorcycle impact.

With regard to the reconstruction of accidents involving ninety-degree motorcycle impact against stationary or slowly moving passenger cars, a relationship was established between kinetic impact energy and sum of permanent penetration depth on the passenger car and permanent wheelbase shortening of the motorcycle, with it being possible for this relationship to be used as a further basis for reconstruction.

The center-of-gravity trajectories of the riders showed a more pronounced upwards tendency in the pillion passenger than in the driver. In conformance with findings from real accidents, the tests showed that the pillion passenger is more likely to go over the roof of the passenger car, possibly flying further than the driver.

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Improvement of Conspicuity of Motorcycle Drivers by Passive Materials

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Abstract

Based on the measurements of the contrast of motorcycle drivers against the background, different features like protective clothing with retroreflective materials were investigated. Out of these partly dynamic tests improvements of the marking of motorcycle drivers are developed. Proposals of an optimal marking is derived from these experiments.

Introduction

Starting from the measured contrast of motorcycle drivers in the street, different markings were investigated. The tests of comparison were carried out dynamical and statical in the normal street situation and in a down scaled test set-up. The influence of the following parameters were investigated

size and shape of the marking
distance between marking.

As criterion a 9-rating scale was chosen.

Marking of the Driver

The motorcycle driver was marked in his vest-area, as shown in Figure 1. The geometry is layed down in Table 1.

Assessment of the Luminance of the Marking

In the experiments a 9-rating scale was used as summarized in Table 2.

The tests were carried out indoor and outdoor with 10 ennetropic test-persons. The marking of the motorcycle driver was illuminated by a typical European low-beam headlamp. In the 1 : 10 down scaled

Table 1. Geometry of the used markings.

Marking	"a"/cm	"b"/cm
1.1	3,5	3,5
1.2	5,0	5,0
1.3	7,0	7,0
2.1	3,5	3,5
2.2	5,0	5,0
2.3	7,0	7,0
3.1	3,5	7,0
3.2	5,0	10,0
3.3	7,0	14,0
4.1	5,0	5,0
4.3	7,0	7,0

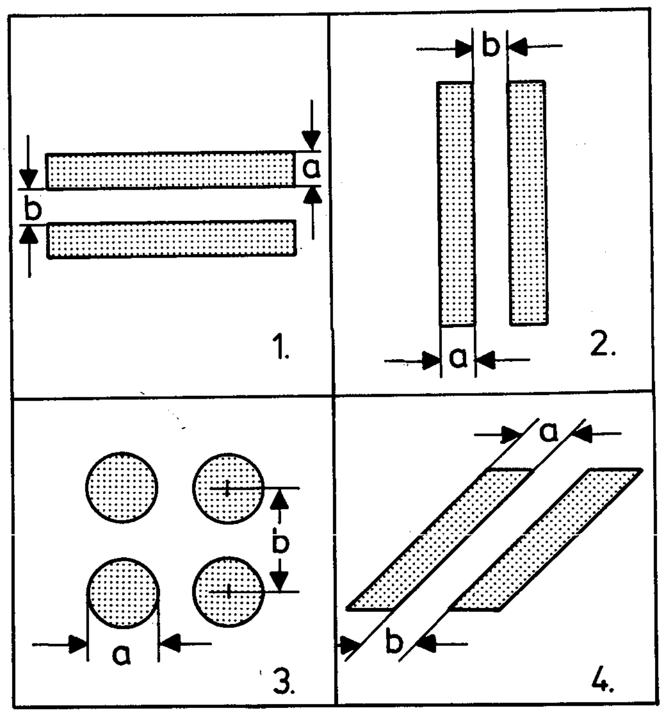

Figure 1. Different markings used in the experiment

Table 2. Assessment of the luminance of the markings.

grade	assessment
1	too glaring not recognicible
2	
3	bright recognicible
4	
5	optimal recognicible
6	
7	dark recognicible
8	
9	too dark not recognicible

experiment the illumination was changed by means of projectors.

Test-Results

Test-results are shown in Figure 2 ...5, for the different markings as described in Figure 1 and Table 2.

The results show the dependence of the optimal luminance L of a marking on the distance d in meter where this marking is shown to the test-person. All curves have a similar shape. Beginning at relative low luminances for short distances, the luminances increase rapidly for larger distances.

A comparison of these results is shown in Figure 6. For certain viewing distances d the optimal luminance L is plotted for the 4 different markings. These curves are the results from calculation of the regression. For small distances the marking 1 (horizontal stripes) is optimal, for large distances the markings 2 and 3.

In Figure 7 one of the test results (marking 3) is compared with the luminances which can be reached with normal retroreflective materials.

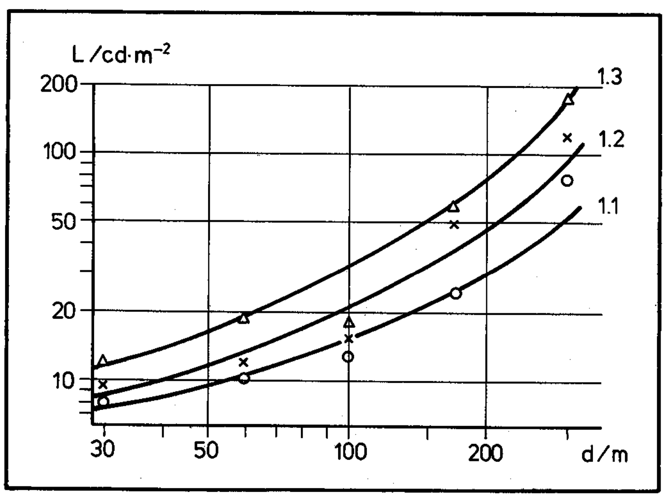

Figure 2. Luminance L and viewing distance d 1.1, 1.2, 1.3: marking with different geometric forms

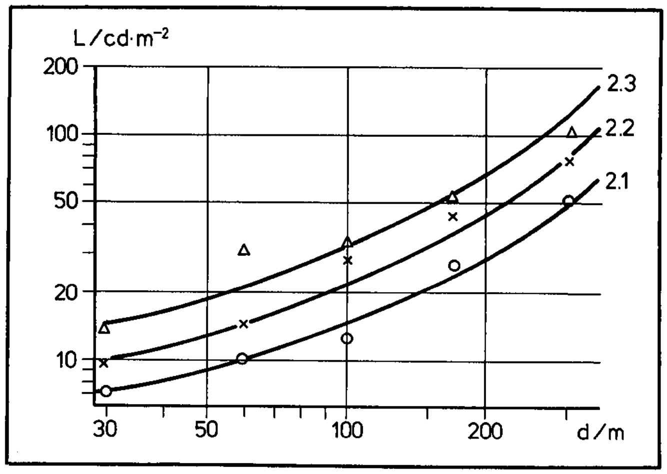

Figure 3. Luminance L and viewing distance d 2.1, 2.2, 2.3: marking with different geometric forms

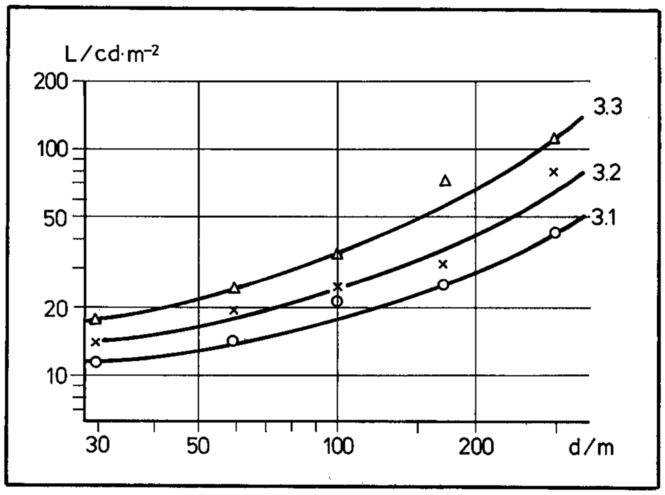

Figure 4. Luminance land viewing distance d 3.1, 3.2, 3.3: marking with different geometric forms

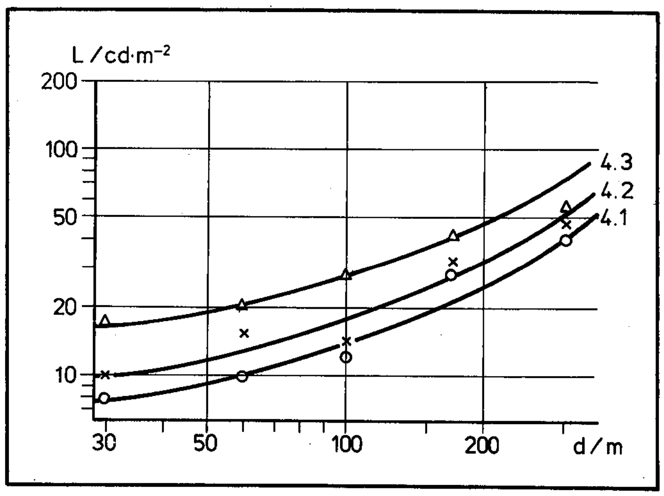

Figure 5. Luminance L and viewing distance d 4.1, 4.2, 4.3: marking with different geometric forms

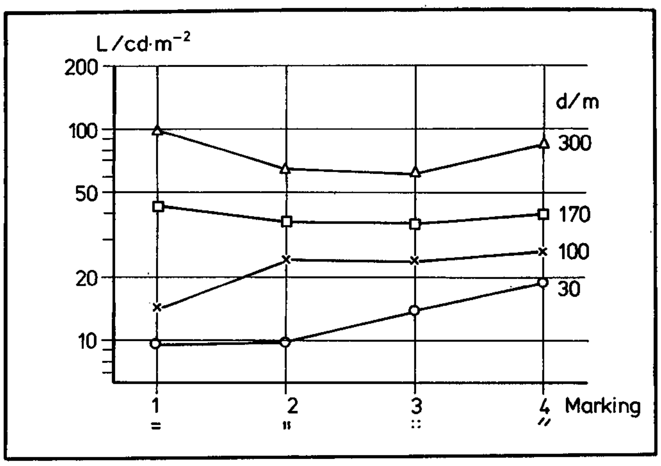

Figure 6. Luminance L for different markings d : viewing distance

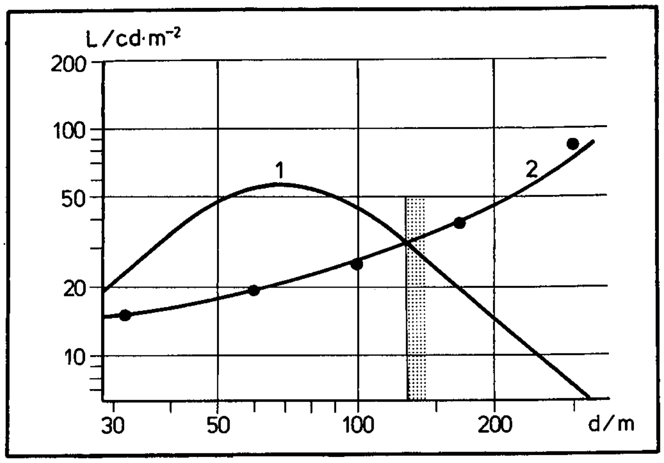

Figure 7. Luminance L and viewing distance d
1 : Luminance of white retroreflective material illuminated by low-beam-headlamps
2 : optimal luminance for marking with dots

Curve 1 shows the luminance of a retroreflective material illuminated by a low beam. Curve 2 is the result for the rating "optimal recognition" for the marking with dots. For distances d up to 130 m the requirement for "optimal recognition" can be fulfilled. Similar results can be reached with other markings.

Conclusions

The marking of motorcycle drivers can be improved by means of retroreflective materials. Up to distances of d ≈ 100 m the marking with horizontal stripes seems to be best. Up to d = 130 m an optimal marking is possible.

These results were gained without oncoming traffic and other glaring light sources. The results should be proved by a large scale experiment.

Protecting Motorcyclists' Legs (Written only paper)

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Protecting Motorcyclists' Legs

Introduction

An essential part of the development of leg protectors for motorcycles is the development of realistic test procedures to check whether they are satisfactory. This paper describes the test procedures which were set up for this work and then gives some test results which compare the performance of the currently preferred design of leg protection on motorcycles, with that of the unmodified machines. Performance is measured by recording head and chest accelerations on a dummy rider together with estimates of the energy absorbed in damage to its leg. This is the leg on the side of the motorcycle which is damaged on impact with the target car.

Test Procedure—Stationary Target Car

The essential features of the impact of a motorcycle into a car can be represented by tests with a stationary target car, and the leg protectors were developed using such a technique. The leg protectors consist of semi-conical leg guards just ahead of the lower legs of the rider, with knee padding to protect the knees. They are firmly attached to the frame of the motorcycle.

Four step-through and 4 BMW motorcycles were impacted at 48 km/h into a stationary car (Marina 4 door). Each set of tests comprised two impacts into the side and two into the front of the car. The impacts into the car side were aimed between the A and B post on the driver's side, and as though the car and motorcycle were travelling in the same direction at 30° to each other (see fig. 1). The impacts into the car front were aimed at the centre of the front. *All* the impacts were with the target face inclined at 30° to the motorcycle direction of travel, and each pair of impacts consisted of one with an unmodified machine and the other with leg-protectors. The tests of two

B.M.W.'s into the side of the car showed that the interaction of the horizontal cylinder head with the car induced a different and more violent impact than might otherwise be expected and one that is probably not typical of large machines. Therefore the remaining tests were with the cylinder heads removed and the equivalent mass replaced by lead weights inside the crankcase. In addition to these tests a B.M.W. with the leg-protecting fairing was impacted into the front corner of a car to provide more information on leg protection.

The leg-protection that was used is a foam filled metal semi-cone which is the one preferred from a previous series of tests into a rigid barrier(1)(2). In the barrier tests the protector absorbed about 5 percent of the total impact kinetic energy for the step-through motorcycles and 10 percent for the medium and large machines. It was assumed however that when impacted into a car that much of the energy would be absorbed by deforming the car body as well as the leg protector.

The foam filled energy absorber was built into a glass fibre fairing based on the B.M.W. R.T. type(3) and this was fitted to two of the large machines. The unmodified equivalents were tested without a fairing fitted.

The dummy rider is a fiftieth percentile OPAT fitted with the aluminium honeycomb injury indicating legs(1)(2) and with accelerometers in its head and chest. Accelerometers were also fitted to the motorcycle, and high speed film was used for the trajectory analysis.

Results

The results given below are for the tests into a stationary car and in some instances they are compared with those obtained previously from impacts into a rigid barrier. In this way differences between the two test procedures are highlighted.

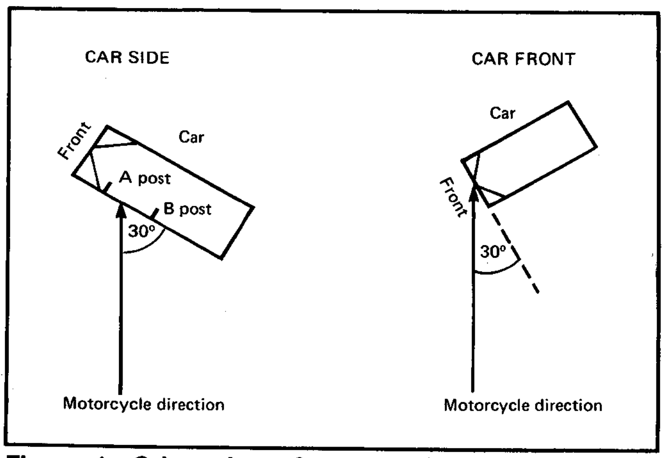

Figure 1. Orientation of motorcycle in impacts with the side and front of a car

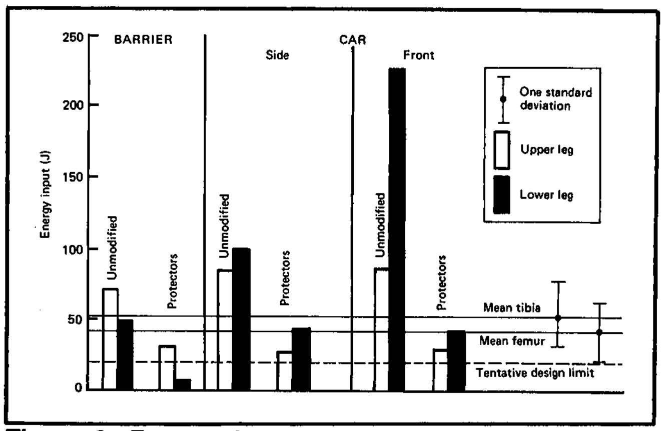

Figure 2. Energy absorbed by leg in impacts of small (step-through) motorcycles

Leg Injury

The energy absorbed by the leg of the dummy rider is given in figs. 2 and 3 and is expressed as previously(3). A tentative limit of 20J is suggested. Mean values which correspond to breaking the femur (upper leg bone) and tibia (lower leg bone) are shown in figs. 2 and 3 together with a range of ± one standard deviation. The values which are estimated using results given in references(4) and (5) are, for the femur (± one standard deviation) 24 to 62J with a mean of 43J and for the tibia (± one standard deviation) 30 to 78J with a mean of 54J.

Heavy Machines

As has been stated(2)(3) the results for the heavy machines into barriers are inconclusive because the trajectories of the motorcycles were severely affected by the horizontal cylinder heads and resembled that of a violent frontal impact. The leg damage was fairly low in all tests (see fig. 3), although the potential head injury was greatly reduced by the leg protectors.

The impacts into the side of the car were also affected by the cylinder heads but not to the same extent, because the car body panels deformed whereas the barrier could not. The cylinder head protected the lower leg but the impact energy sustained by the

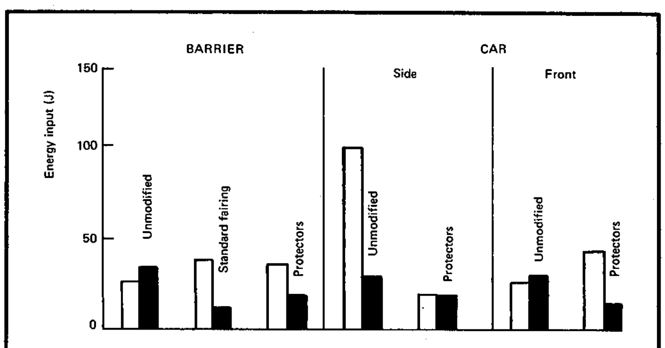

Figure 3. Energy absorbed by the leg in impacts of large (BMW R80) motorcycles

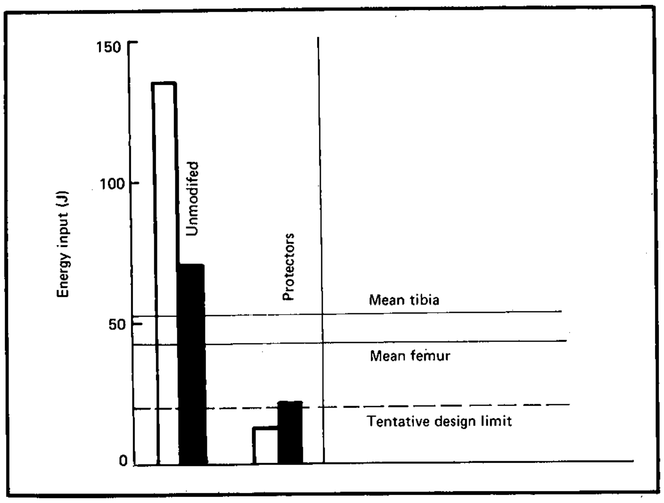

Figure 4. Energy absorbed by leg impacts of medium weight motorcycles into oblique barrier at 48 km/h

upper leg was high and well above the tolerable level. The leg-protective fairing greatly reduced the energy sustained to below the design criteria of 20J, for each limb.

In the next set of tests the unmodified machine interacted violently with the front of the car (despite the lack of cylinder heads) and produced a typical frontal impact with the dummy being thrown off head first (fig. 6). The trajectory of the modified machine (fig. 5), was satisfactory as was expected. The leg impact energy sustained was similar for each test (see fig. 3). The result with the modified machine confirmed the need for an efficient knee energy absorber; earlier research(2)(3)(6) had shown this to be essential.

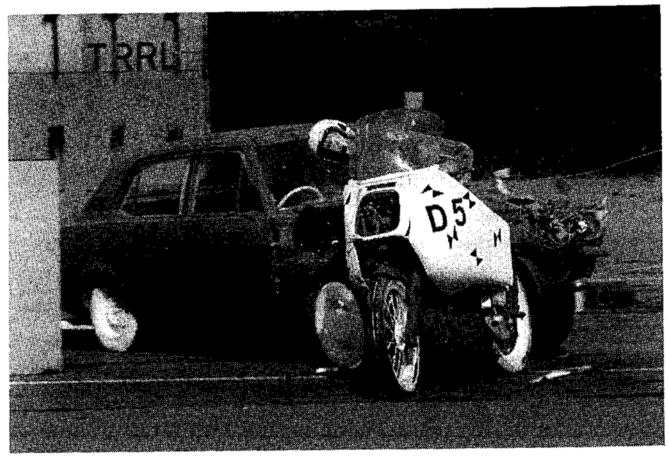

Figure 5. Motorcycle fitted with leg protecting fairing. (During impact)

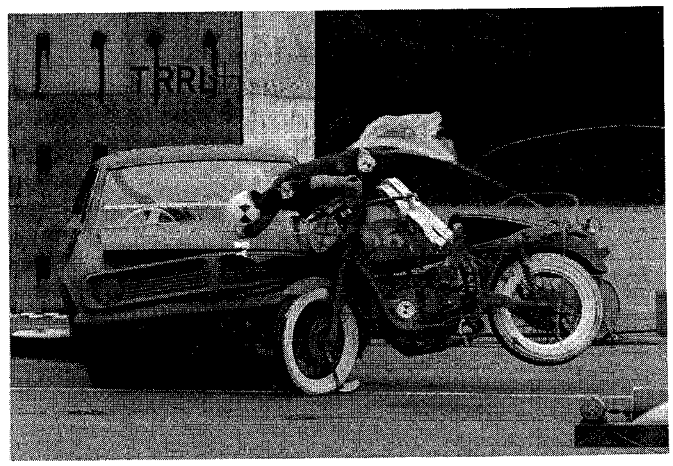

Figure 6. Unmodified motorcycle. (During impact)

Table 1. Dummy rider head & chest measurements—motorcycle car impacts.

MOTORCYLE LAYOUT	MEASUREMENT	STEP-THROUGH			LARGE		
		SIDE	FRONT	MOVING FRONT	SIDE	FRONT	CORNER
UNMODIFIED	HEAD HIC	400	*	1353	327	164	NA
WITH PROTECTORS	HEAD HIC	*	*	876	259	*	145
UNMODIFIED	HEAD PEAK g	109	*	144	93	32	NA
WITH PROTECTORS	HEAD PEAK g	*	*	115.0	67.9	*	54.2
UNMODIFIED	HEAD 80g EXCEEDED ms	1.8	*	8.3	3.3	0	NA
WITH PROTECTORS	HEAD 80g EXCEEDED ms	*	*	3.3	0	*	0
UNMODIFIED	CHEST PEAK g	35	1	45	22	14	NA
WITH PROTECTORS	CHEST PEAK g	2	13	58	12	NA	23

* = NO CONTACT

NA = NOT AVAILABLE

Step-through Motorcycles

The results from this series of tests are given in fig. 2. The barrier tests suggested that leg protection could be effective, the car tests have reinforced this finding and also tend to agree with the accident studies, which show that the lower leg is more frequently and seriously injured than the upper leg.

The energy sustained by the upper leg is almost the same for all three tests and is caused by contact with the protector. Although energy absorbing foam was fitted ahead of the knee, it did not completely resist the solid metal substrate of the dummy leg and was fully crushed. A human knee would not have penetrated the foam so deeply, and would not have contacted the protector attachment struts, as happened in these tests. Careful observation of the high speed film shows that at no time did the leg make contact with the car.

Head and Chest Injury

In the first series of tests(1)(2)(3) horizontal head velocity measured at the barrier vertical plane was considered indicative of potential head injury. However in the impacts into cars the head invariably struck either the roof or bonnet while moving vertically downwards. It is considered therefore that vertical velocity might be more relevant, and where appropriate it is given in Table 1. Also given are H.I.C. (Head Injury Criterion) values, peak resultant acceleration, and times for which 80g was exceeded, all were evaluated from the head tri-axial accelerometer. Chest peak resultant acceleration is also given, and again this is evaluated from a tri-axial accelerometer.

These results should be considered as a set. For every pair of tests in which head contact occurred the H.I.C. was reduced when the machine was fitted with leg-protectors. In the tests into the car side no head contact occurred when leg protection was fitted, whereas with the unmodified machine the head struck the "A" post and the bonnet. The tests of the step-through machines into the car side showed that leg-protection prevented head contact. The chest accelerations are generally low and again leg-protection tends to effect a reduction.

It has been said that vertical velocity may be a good indication of potential head injury and a graph was plotted of vertical head velocity against H.I.C. (fig. 7). With the exception of one point the tendency is for HIC to increase with vertical head velocity. However more data is needed to confirm this.

Motorcycle Trajectory

The results from the research suggest that leg injuries tend to be potentially more severe when the motorcycle angular velocity is high during the first 100 m/s of the impact. This is illustrated by Fig. 8 which shows the average angular velocity during this period for the unmodified machine, and by Fig. 9 which gives the values for the motorcycles with leg protectors.

The results for the large machines are affected by the cylinder heads and so appear anomalous but they do illustrate that leg-protection can both increase and decrease the angular velocity to the advantage of the legs and the head. If however the linear forward velocity changes rapidly on impact, as with the large unmodified machine into the car front, then the rider will leave the motorcycle head first (see fig. 6). This is well before the rotation of the machine can have any effect. For leg protectors to be effective both linear and angular velocity must be controlled (see fig. 5).

The angular velocity of the unmodified step-throughs (mean for all tests is 5.2 rad/s) is consistently higher than for those with leg protection (mean 2.5 rad/s). This result supports the contention that leg injuries are related to angular velocity.

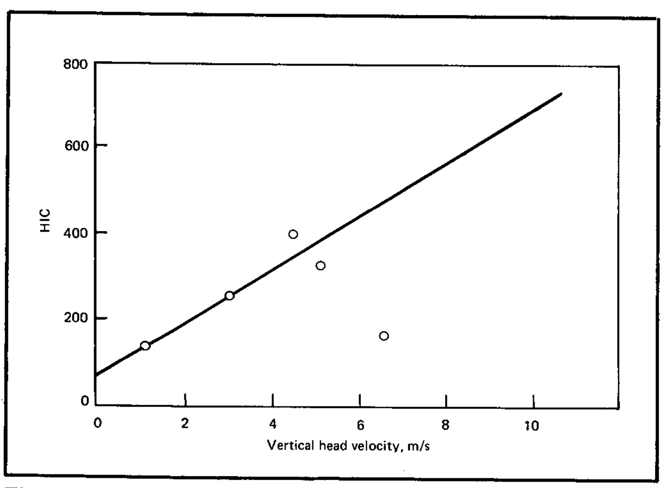

Figure 7. Vertical head velocity against HIC

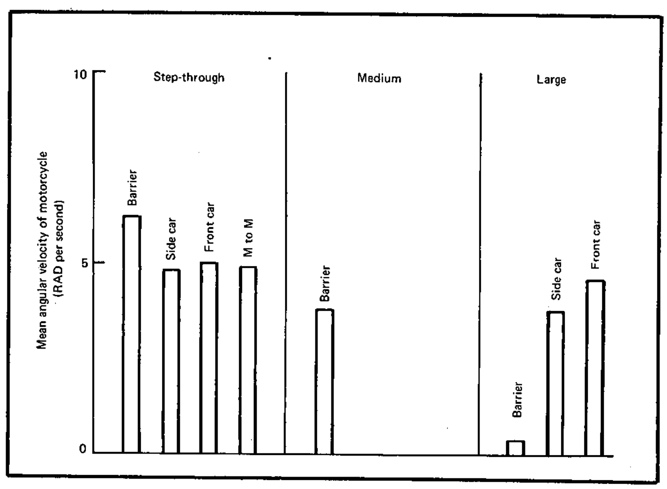

Figure 8. Mean angular velocity of unmodified motorcycle during first 100ms

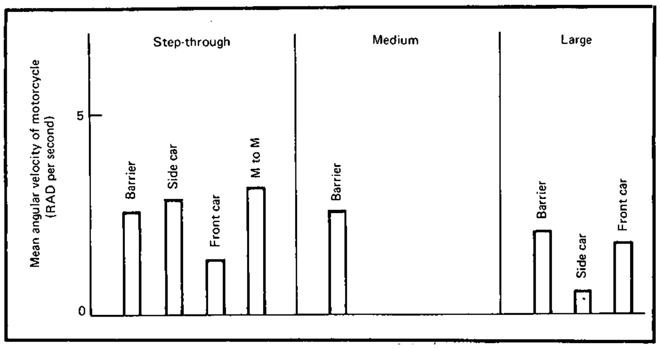

Figure 9. Mean angular velocity of modified motorcycle during first 100ms

Test Procedure—Moving Target Car

Doubts have been raised about the realism of impacts into stationary cars and so motorcycles were impacted into a moving target to check this. As well as this, a simple theoretical model is derived later to compare the two procedures. Two step-through motorcycles were impacted at 48 km/h (30 mile/h) into the front of a car moving at 24 km/h and at an impact angle of 30°. One motorcycle was unmodified the other was fitted with leg protectors.

The impact point was intended to be the centre of the car front but the first impact occurred at about a quarter of the distance across. No changes were made to the system and the second impact was identical. An adjustment will be made for future tests. The results of these impacts are described in a later section.

Apparatus

Fig. 10 shows the layout of the apparatus and fig. 11 shows the motorcycle launch trolley with its wheels in the guide rails. The car is similarly guided. The towing car (a large Oldsmobile) pulls the two vehicles together and when they are near the impact point the towing cables are automatically released as is the

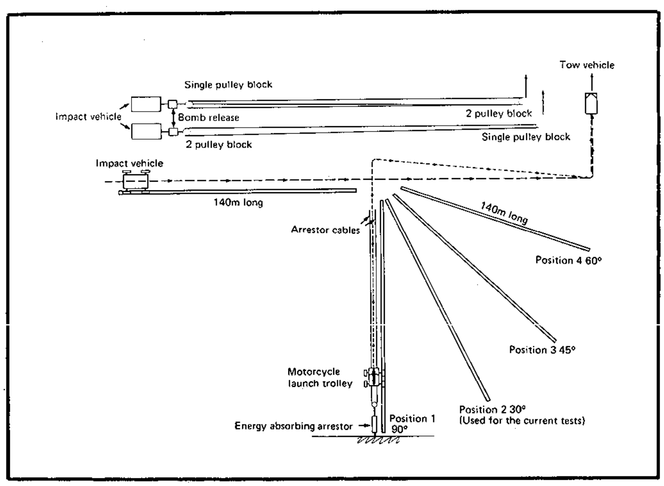

Figure 10. Impact apparatus for moving motorcycles to moving car tests

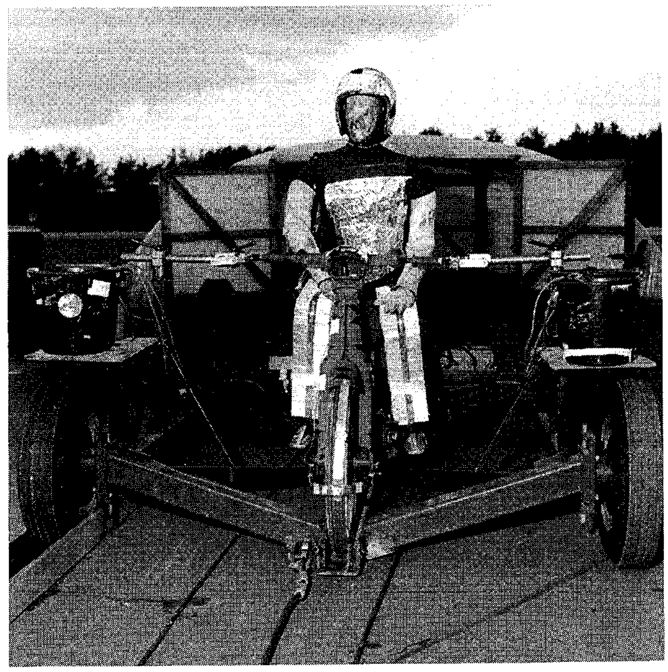

Figure 11. Motorcycle launch trolley for impacts with both vehicles moving

catch holding the motorcycle to the trolley. The trolley is brought to rest by an energy absorbing arrestor which causes the motorcycle to be launched. The mechanism by which it is launched is identical to that used previously when a launch frame was fitted to a Land Rover.

The speed ratio, motorcycle to car, can be 1:1, 2:1, or 3:1. The second two ratios are achieved by inserting the appropriate pulley blocks between the

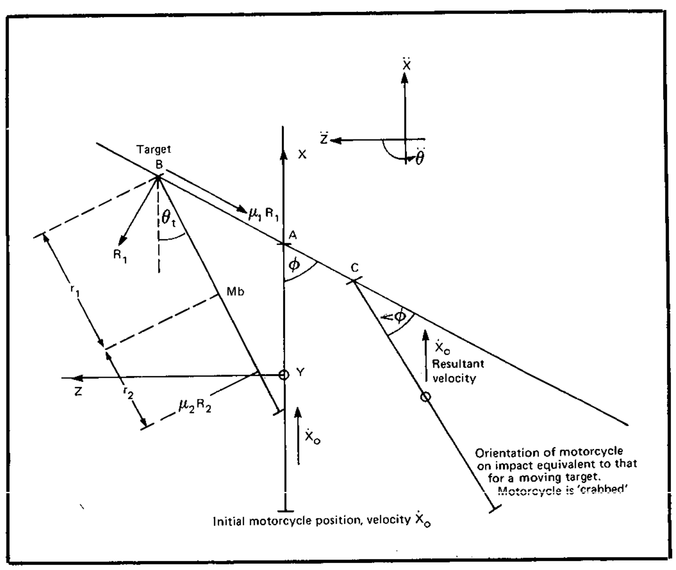

Figure 12. Simplified motorcycle, striking stationary target at A at t = 0 and slides to B at time t. Orientation of motorcycle at t = 0 for an impact equivalent to one with a moving target.

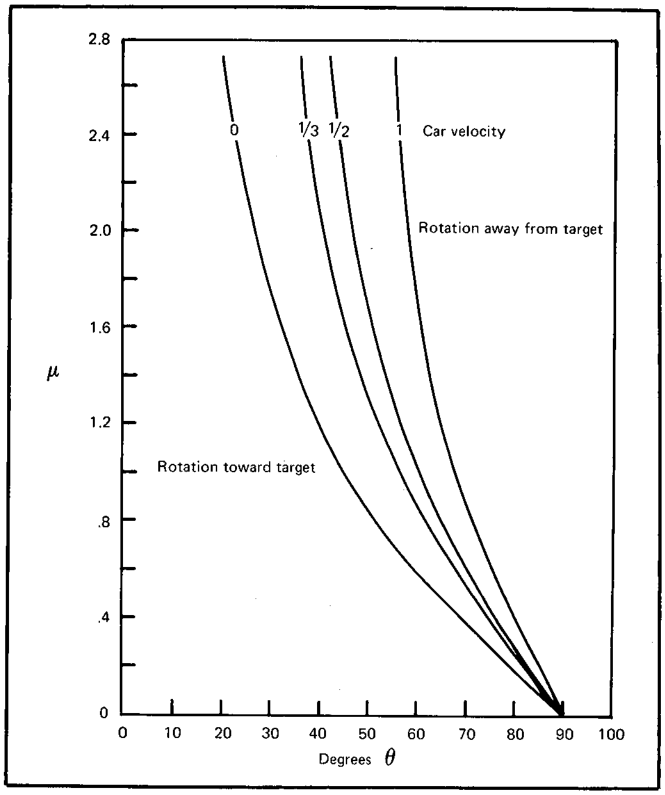

Figure 13. Graph to show the effect of friction on motorcycle rotation for angled impacts into stationary and moving targets.

towing vehicle and the car as shown in the two illustrations above the main diagram. The pulley block attached to the car is fitted with a metal "skid", which allows the pulley to slide easily along the ground after it has been released.

Simple Representation of a Motorcycle Impact

Previous work has shown that the coefficient of friction, which is generated on impact between a motorcycle and its target, greatly affects the subsequent motion. In order to study this a simple theoretical model was derived from which could be predicted some possible differences between the test procedure with a stationary and a moving target. The friction affects the findings.

Fig. 12 is a diagram of a simplified motorcycle during an oblique impact with a stationary target. The motorcycle is assumed to be equivalent to a blunt rod with a deformable front end which will crush during the contact period. μ_1 is the coefficient of friction between the motorcycle and the target and μ_2 the coefficient between the rear tyre and the road. R_1 is the reaction at the target face and R_2 the reaction between the rear wheel and the road. Mb is the mass of the bike at the centre of gravity and $r_1 r_2$ define its position relative to the front and rear tyre contact

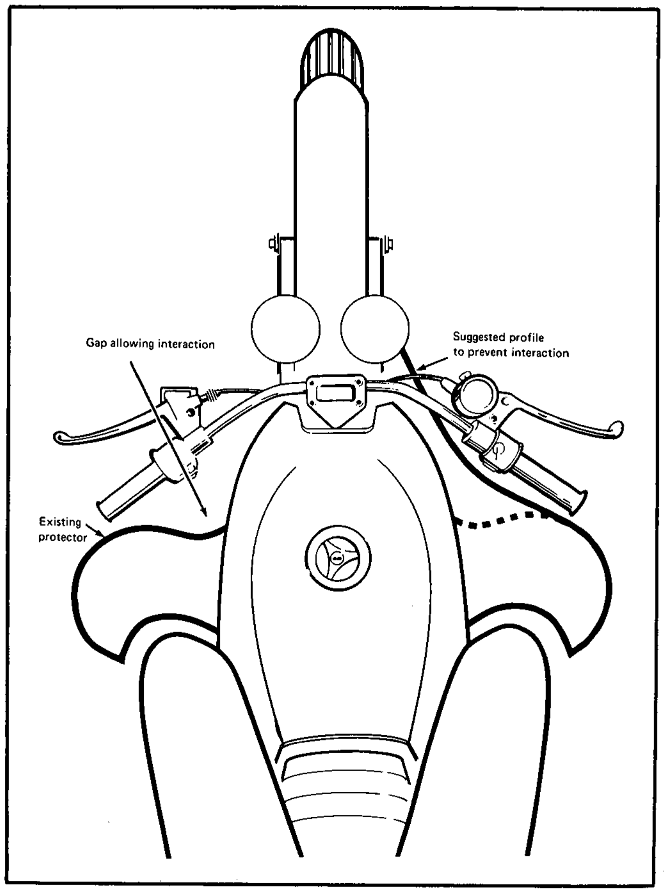

Figure 14. Diagram to show how existing protector can be modified to prevent interaction

points. Ib is the moment of inertia of the motorcycle in yaw when it is assumed to rotate about the front forks. ϕ is the impact angle of the motorcycle relative to the target and the Θ angle moved through at a given time.

The motion most critical for leg injuries is rotation and using the equation of rotational motion (Inertia × angular acceleration = Torque) gives (i) below,

$$\text{Ib}\,\ddot{\Theta}_t = \left(R_1 \cos(\phi - \Theta)\right) r_1 - \mu_1 R_1 \sin(\phi - \Theta)\, r, - \mu_2 R_2 r_2 \quad \ldots \quad (1)$$

(N.B. The front wheel is assumed to be parallel with the target face and rolling with zero friction.) If leg injuries are to occur the rotation on impact is towards the target face so that the motorcycle slides along it, i.e. anticlockwise in fig. 13 and this depends on μ_1

at impact t = O and Θ_o = O and (1) becomes

$$I_b\,\Theta_o = R_1(\cos(\phi)) r_1 - \mu_1 R_1 (\sin \phi)\, r_1 - \mu_2 R_2 r_2$$

Rotation towards (turning parallel to) the target will occur if Ib Θ_o is positive. $\mu_2 R_2 r_2$ reacts only to oppose the motion and is zero when t = O, therefore IbΘ_o will be positive if $\mu_1 R_1 (\sin \phi)\, r_1 < R_1 (\cos \phi)\, r_1$

$$\text{ie } \mu_1 < \cot \phi$$

Consider the motion of the motorcycle relative to a moving target. The resultant velocity of the motorcycle can no longer be considered to be acting along its longitudinal axis and if the resultant impact velocity is at the same angle ϕ relative to an equivalent stationary target then the motorcycle must be orientated at an angle less than ϕ i.e. "crabbed" as illustrated in fig. 13.

Fig. 13 shows graphs of the resultant impact velocity angle (relative to the target) against the friction coefficient critical to the direction of rotation, and illustrates that for a given coefficient, rotation toward and along the target (leg injurious) can occur over a greater range of impact angle if the target is moving. The extent of the range depends on the ratio of the motorcycle and target velocity, and the ratios available for the practical tests are (target car velocity-/motorcycle velocity) = 0 (car stationary), $\frac{1}{3}$, $\frac{1}{2}$, 1. The graphs in figure 13 correspond to these ratios and when compared indicate that a static target at 30° (test condition) provides the conditions for rotation equivalent to a moving target at 44° ($\frac{1}{3}$), 49° ($\frac{1}{2}$), and 60° ($\frac{1}{1}$) for a coefficient of friction less than 1.7. The rate of rotation depends on the resultant impact velocity, and for one of 53 km/h (33 miles/h) the motorcycle will be travelling at 40 km/h (25 miles/h) when the target is doing 20 km/h ($12\frac{1}{2}$ mph) and the relative angle is 30°. The resultant velocity for 45° is 48 km/h (30 miles/h). It can be said therefore that an impact into a stationary target at 48 km/h and a 30° angle is approximately the same as an impact at 40 km/h into a target moving at 20 km/h at a 45° angle if the friction generated is the same.

(It is interesting to note that an accident study (7) has shown that the mean speeds for motorcycle to car collisions are 39 km/h for the motorcycle and 23 km/h for the car. This is for an injury based sample.)

Figure 14 shows that a high value of friction greatly reduces the chance of rotation, and this is known from tests to be detrimental to the head. A low coefficient greatly increases the chance of rotation which is known to be detrimental to the legs.

An optimum is required and a correctly designed energy absorbing fairing will provide this.

Results—Moving Target Car

In both tests the step-through stopped rapidly, and although the impact with the modified machine was the less violent, it demonstrated a known design weakness in the current protector which allowed more interaction with the car than is desirable. This occurred because the curvature adjacent to the machine bends to the rear. This can be easily corrected as shown in fig. 14.

The leg damage is not fully analyzed but a subjective assessment indicates a similar result for both tests. However a substantial indent in the side of the car wing caused by the knee of the rider of the unmodified machine indicates that serious knee-femur-hip injury might have been sustained.

The head and chest accelerations are given in Table 1 and the motorcycle angular velocities in figs. 8 and 9. The H.I.C. indicates a fatal head injury for the rider of the unmodified machine whereas with leg-protectors fitted the value was well below 1000. The chest acceleration is slightly higher for the modified machine but is non-fatal. This is preferable to a fatal head injury.

The angular velocity is high for the unmodified machine, and had the impact occurred further across the front of the car the leg would probably have sustained extensive damage.

Conclusions

Tests with static and moving targets have shown that leg protection can affect a reduction in a motorcycle rider's potential leg and head injuries. The trajectory of the motorcycle during the impact is important as this has a marked effect on the trajectory and hence the potential injuries of the rider.

The potential for leg injuries increases as the yaw angular velocity increases. The potential for head injuries tends to increase as the yaw angular velocity decreases probably because this is related to pitching and high motorcycle deceleration. High friction and interaction cause this situation which can be readily predicted theoretically. Correctly designed energy absorbing leg-protection can control the trajectory of the motorcycle rider, induce an optimum motorcycle yaw angular acceleration and hence minimize a rider's injuries. Performance criteria for leg-protection can now be specified for all sizes of machine, and will be published as a separate paper.

It has been shown theoretically and practically that a test procedure using a stationary car can represent an impact into a moving target albeit of a different speed and angle.

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A Field Trial of Motorcycles Fitted With an Anti-Lock Brake System (Written only paper)

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Abstract

Although all types of vehicles are subject to skidding, particularly in adverse weather conditions, powered two-wheeled vehicles have the highest incidence of accidents in which it is a factor. The inherent instability of these vehicles makes capsize almost certain if a wheel is locked during braking, leading to risk or injury to the rider. There is a growing body of evidence, both from accident studies and observation of the behaviour of motorcyclists, that riders do not brake in a safe and effective manner. In particular, the front brake is used insufficiently or not at all; this is said by many riders to be because of their fear of the consequences of locking the wheel.

Research into the application of anti-lock brake systems for motorcycles has been conducted by TRRL during the last twenty years. This has demonstrated the possibility of reducing the incidence of skidding and improving braking performance generally by the widespread use of such systems. Research has reached the stage where reliable anti-lock systems are available but little is known about the way in which riders would use them and how they would react to their presence on a motorcycle. TRRL and Lucas Girling Limited have therefore undertaken a joint project to acquire data from a field trial of seven motorcycles equipped with an anti-lock system developed by Lucas Girling. The machines are in use with a number of Police forces and commercial organisations and are expected to cover relatively high mileages during the three year duration of the trial. This paper represents some of the information obtained at the half-way stage of the trial and gives details of reliability and riders' reaction and comments.

Introduction

Evidence of the difficulties faced by riders in the braking of motorcyles was presented at the 10th ESV Conference.(1) There seems to be little evidence that the part played in accidents by inadequate braking is decreasing to a significant extent. Indeed, recent studies(2)(3) of the behaviour of riders suggest that as far as braking technique is concerned there are serious problems. For example, it was observed that even during emergency braking, over 20 percent of riders used only one brake, predominately the rear. During "normal" braking only about half the riders observed by Sheppard et al. used both brakes. Clearly, if this pattern of braking behaviour is widespread, many motorcyclists are not even attempting to brake effectively. In particular, failure to use the front-brake limits the deceleration available to the vehicle. In an accident-study made in Australia(4) it was suggested that 30 percent of the accidents investigated could have been avoided if the available braking capability of the motorcycle had been used.

The reasons why riders brake "incorrectly" are not clear; training or the lack of it does not appear to be a factor according to Sheppard et al. However, many riders in his study expressed a fear of locking a wheel, particularly the front wheel. This is consistent with the fact that over 50 percent of motorcyclists who were interviewed said that they had skidded at some time during their riding career. Clearly the removal of the fear of locking a wheel would contribute to an improvement in the standards of braking of motorcycles in several ways. First, those accidents attributable directly to the locking of a wheel could be eliminated. Second, and perhaps more important, riders would gain confidence in using the braking-performance of their machines to the maximum. This could bring about a reduction in accidents by reducing stopping-distances.

The widespread use of effective anti-lock brake systems on motorcyles seems to be an obvious way of achieving the result discussed above. Such systems have been the subject of research for more than twenty years but little experience exists of their performance in the field. TRRL and Lucas Girling Limited are jointly conducting a field-trial with an anti-lock brake system developed by Lucas Girling. This has involved the fitting of the system to a number of motorcycles which have been distributed to Police forces and commercial organisations.

This is believed to be the first such trial of anti-lock systems on motorcycles, where the requirement for safe and reliable operation is critical because of the inherent instability of such machines.

History of the Trial

On the basis of experimental work which has taken place during the last twenty years(5)(6), it is believed that a wide-spread use of anti-lock brake systems on motorcycles would confer a number of benefits:

i) capsize or loss of control caused by over-braking produced by panic or misjudgment would be eliminated,
ii) steering-control would be retained during emergency braking,
iii) all riders could brake to the levels attained by the most skilled,
iv) removal of the fear of wheel-locking should encourage riders to employ the braking-performance of which their machines are capable.

However, no information exists about the use in the field of motorcycles fitted with anti-lock brake systems. It is possible that there could be problems related to factors such as reliability and rider-acceptance which are not apparent during research and development. At the outset it was clear that the reliable collection of data would be an essential part of the trial. This requires disciplined and methodical recording of vehicle maintenance, defects and mileage. As a result, the Police were approached with a view to taking part and agreed to do so. In addition, a company which uses motorcycles to deliver urgently-required mail agreed to participate.

At present six machines are in use by various Police forces in the United Kingdom, one by Cycle Courier Ltd. and one has toured a number of European countries for assessment by Government authorities. A ninth machine (R1) is based at TRRL for development and demonstration to participants in the trial. Two machines (ESM 2 and ESM 3) have the anti-lock system fitted as one of their safety features. These are exhibited nationally and internationally but have not to date been issued to external users.

Development has proceeded for a number of years and the first machine was issued to Gwent Constabulary in May 1985. Apart from R5, issued to Cycle Courier Ltd, each machine will be assessed by a number of users, each for a period of at least six months. R5 is intended as a high-mileage machine and will be used by Cycle Courier Ltd until it is life-expired.

The Trial Motorcycles

The motorcycles to be issued to external users were bought new, specifically for the field-trail. In the case of machines R1 to R5, it was necessary to fit cast-alloy wheels and convert the rear brake to disc operation, in order to accommodate the anti-lock system. In other respects all machines for use by the Police are to normal Police Specification, the remainder are standard "civilian" machines.

Three models of motorcycle are involved, all of which are in common use by Police authorities. The Cycle Courier Ltd. machine is similar to those normally used by the company. R1 to R5 are BMW R80/R100 machines; K1 to K5 are BMW K100 machines and N1 to N2 are Norton Interpol II machines.

The anti-lock systems were installed and tested by Lucas Girling Ltd. Although three types of motorcycle are involved, the fitting of the anti-lock units was straightforward. This is because prototype units were used which were designed to be applicable to a wide range of motorcycles.

Trial Procedure

After installation and testing of the anti-lock units, the motorcycle was handed over to the user, who was asked to treat the machine like any other in his fleet. (This was one of the reasons why three models of motorcycle are involved. Users have an "anti-lock" machine which is otherwise similar to the remainder of their vehicles).

It was considered that prospective riders of the trial machines would find it interesting to ride the TRRL-based motorcycle on a test-track, particularly as this machine is equipped with safety-skids which allow wheels to be locked without danger. Whenever possible, riders are given practical experience on a test track when each trial machine is handed over. This takes the form of making heavy applications of the brakes both with and without the anti-lock system operating. In this way riders are able to experience the effects of wheel-locking and the way in which an anti-lock system works to prevent it. Subsequently the value of this period of "tuition" became apparent and will be discussed later.

Information is obtained from the trial in several ways. Each rider is asked to complete an assessment form after having ridden the trial machine for a period of time. (Figure 1). This seeks subjective opinions about various aspects of the anti-lock system and its installation and also invites more general comments. In addition, each participant is visited periodically by a representative of Lucas-Girling Limited or TRRL to monitor progress. Problems or failures which are related to the anti-lock systems are rectified as they occur and a record made of their nature. In March 1987, an informal meeting of all past and current participants was held to discuss the progress of the trial. It is likely that this will be repeated at the conclusion of the trial. Figure 2 shows the history to date of each participating motorcycle.

Results

The rider-evaluation form seeks subjective ratings on a scale of one to ten of various features of the anti-lock brake installation. Figure 3 is a summary of these ratings based on the forms received to date. Figure 4 lists briefly the comments which riders added

Motor-cycle reg'n. no. ____________________

Motor-cycle Anti-Lock - Rider Evaluation sheet

Name .... Age ....

Riding Experience .... Current m/cycle (prior) ....

Test date .... Weather conditions ....

Test route summary ....

Assessment of Anti-Lock

Rating. (F=Front, R=Rear, if different)

Not acceptable. Satisfactory. Very good.

	1	2	3	4	5	6	7	8	9	10
1 Lever pulsing										
2 Fork vibration										
3 Vehicle drive										
4 Vehicle control										
5 Performance - dry road										
6 - wet road										
7 - slippery road										
8 - normal braking										
9 General brake feel										
10 Installation concept - packaging										

General comment:

e.g. Do you like this system? Does it have a future? Do you favour anti-lock for cars, and/or motor-cycles? Would you buy it? How much would you pay? Any comments from above rating chart? Critcisms ? (Attach sheet if necessary)

Figure 1. The rider evaluation sheet

to the assessment forms. These can be divided into "unfavourable" and "favourable." The most common unfavourable remark concerned excessive suspension "dive", particularly with one model of motorcycle. This, of course, is a feature of the design of the motorcycle itself. It is likely that the cyclic action which occurs when the anti-lock system functions will accentuate such characteristics of the vehicle suspension.

A number of riders criticized the appearance of the anti-lock units. At this stage of development it is an essential feature that the system should be applicable to a variety of motorcycles. For this reason the belt-drive arrangement was adopted; in a production version the system could be incorporated into the motorcycle as part of its overall design and styling.

One rider remarked that he was concerned that even though he felt that the system was excellent, it would lead to riders becoming "lazy" because they would not need to consider correct braking technique.

The final form of adverse comment concerned pulsation of the brake controls during anti-lock operation. This is, in fact, a deliberate feature, intended to make the rider aware of the fact that he is overbraking and its absence is an important indication of internal failure of the actuator. In addition, riders learn to assess the degree of slipperyness of road surfaces without risk, because the level pulsation provides an indication.

Favourable comments were less specific than the unfavourable ones. Fifty percent of riders (20) said specifically that they felt anti-lock systems should be a

MACHINE	DATE	MILEAGE	NOTES
R1	Jan. 84	600	Anti-lock installation complete.
			General development, system-tuning for optimum performance.
	Feb. 87	3000	Demonstrations, research for legislative tests, tuition of participants in field-trial.
R2	Mar. 85	-	Anti-lock installation complete.
			Experimental safety motorcycle, ESM2, exhibited nationally and internationally.
	Feb. 87		
R3	Nov. 85	160	Issued to Avon and Somerset Constabulary.
	Mar. 86) May 86)	1000- 2100	Loose steering-head bearings, run-out of front brake-disc (replaced).
	Aug. 86	4000	Issued to Devon and Cornwall Police - satisfactory.
R4	Nov. 85	230	Issued to Thames Valley Police. Loose steering-head bearings at 1000 miles.
	Dec. 86	7000	Issued to Northern Ireland Police Authority.
	Feb. 87	8300	Rear wheel lock at low speed caused by ingress of dirt into flywheel sensor - rectified, now satisfactory.
R5	Mar. 86	810	Issued to Cycle Courier Ltd. Loose steering-head bearings at 6000 miles.
	Dec. 86	28000	Rear anti-lock unit seized because of ingress of dirt into flywheel-shaft bearings - belt failed but normal braking retained.
	Feb. 87	40000	Satisfactory.
K1	Mar. 85	600	Anti-lock installation complete - General development and optimisation tests.
	Aug. 85) Feb. 87)	1500 6000	Evaluation by Government authorities in Holland, France, Germany and UK. - Dirt ingress to front unit sensing mechanism after steam-cleaning.
K2	May 85	150	Issued to Gwent Constabulary - Use restricted by problems with radio interfering with engine-management system.
	Aug. 85	3620	Modified rear anti-lock mounting-bracket fitted.
	Apr. 86	11000	Wear problem with drive pulleys - replaced by steel.
	Jan. 87	21000	Leak in flexible hose in rear brake - replaced. Performance satisfactory.
K3	Aug. 85	500	Issued to Lancashire Constabulary.
	Nov. 85	2700	O-ring leak in rear unit - replaced.
	Dec. 85	4800	Mounting bracket of rear unit redesigned after fixing-bolt became loose. - All similar motorcycles modified.
	May 86	9000	Vehicle destroyed in road-traffic accident - not related to braking.
K4	Aug. 85	180	Issued to Sussex Police - Satisfactory.
	Sept. 86	6000	Issued to Essex Police - Satisfactory.
K6	Feb. 87	100	Ready for issue to Thames Valley Police - replacement of K3.
N1	Mar. 86	1200	Issued to Gwent Constabulary.
	Jun. 86		Issued to Hampshire Constabulary - Satisfactory.
	Nov. 86		Issued to Strathclyde Police.
N2	Aug. 86	13000	Anti-lock installation complete.
			Vehicle to form basis of experimental safety motorcycle ESM-3.

Figure 2. History of participating motorcycles

standard fitting on motorcycles. Although cars were not involved in this trial, eight riders volunteered the comment that they too should be equipped with anti-lock brakes as standard. Seventeen riders stated that they would be prepared to pay for an anti-lock installation on a motorcycle. Five of these specified a price between £140 and £400, representing approximately 3 percent and 10 percent of the cost of the motorcycle itself. The remainder did not specify a price that they would pay. Several riders commented that they thought the widespread use of anti-lock brakes on motorcycles would be a major contribution to safety.

It is interesting to note that even of those riders who were not enthusiastic, none was wholly unfavourable in his comments.

Reliability

The motorcyles in the trial have, to date (March 1987), covered a total of 200,000 km (125,000 miles) without a failure causing a locked wheel incident. There have been several instances of difficulties caused by failures associated with standard components of the motorcycle and with the ancillary equipment of the anti-lock systems. Examples of the former are: distored brake discs, looseness of steering-head bearings, worn brake discs. Each of these caused symptoms which riders attributed wrongly to the operation of the anti-lock system.

The most serious problem with installation has concerned the security of the mounting bracket of the rear anti-lock unit on one model of motorcycle. A modification which incorporates additional fixing-screws was made and appears to be satisfactory. The second problem with the installation of the anti-lock units has been as a result of the ingress of water and grit. This is known to have occurred on two motorcycles; K1 while it was touring Europe, and R5, based with Cycle Courier Ltd. In the first case, dirt and water penetrated the flywheel cap of the front unit during cleaning of the machine and caused the flywheel mechanism to stick. It appeared that the cap had been removed and replaced, leading to poor sealing. In the case of R5, the bearing of the flywheel shaft of one unit seized as a result of water ingress and caused the drive-belt to fail, at 45,000 km (28,000 miles). It is known that this machine is steam-cleaned regularly; as a result road-wheel bearings are changed at 5,000 mile intervals by Cycle Courier Ltd. because of problems similar to that which affected the anti-lock unit.

It should be noted that none of the above faults resulted in loss of braking; at worst, the anti-lock capability was lost.

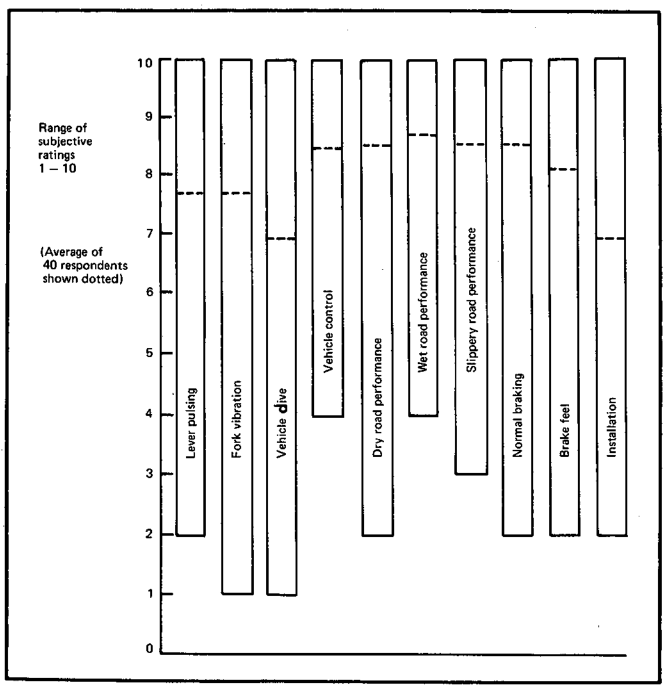

Figure 3. Summary of riders' subjective opinions of ten aspects of the anti-lock system

Assessment of use of anti-lock

A piece of useful information which could be obtained from a trial of this type is the extent to which an anti-lock system is called upon to operate, i.e. what is the proportion of brake-applications in which the system intervenes. Unfortunately it has not been possible to devise a simple and satisfactory means of recording, bearing in mind that the motorcycle provides a difficult environment for instrumentation and that the machines are not available to the trial managers on a day-to-day basis. This is recognised as a failure of the trial to date. Efforts are continuing to find a way of obtaining the information. Similarly, it is accepted that even if this information does become available later in the trial, it will not necessarily provide an indication of the level of anti-lock "use" to be expected of typical riders. This is because the riders involved in the trial are all highly- trained and have considerable experience.

A possible advantage of the delay in collecting use-data is that the "novelty effect" of riders initially experimenting with the system should be absent.

Discussion

The purpose of field-trials of novel equipment is to establish what problems arise as a result of failings of concept or reliability. In this trial the opportunity was taken to seek the subjective opinions of riders. It is accepted that the riders involved are all professional motorcyclists, mostly Police officers, and that their comments and riding-behaviour are not likely to be representative of the motorcyclist population. However, the methodical nature of the systems of maintenance and recording used by the police and similar organisations is invaluable in the collection of data, particularly when close supervision is not possible.

The problems associated with reliability have been discussed above. These have been of an innocuous nature generally and have involved a loss of anti-lock capability as a worst consequence.

Slight changes to some features of the anti-lock units and their associated mountings appears to be all that is necessary to overcome the problems which have arisen to date.

Riders have, in general, been favourable in their reaction to the advantages which anti-lock can confer. This is encouraging when the fact that these riders are trained professionals is considered. Many appeared to be set against the anti-lock system at the outset, expressing the view that "we don't need it," for example. This emphasizes the importance of educating potential riders regarding the working of the system and what it is intended to do. It is significant that riders from one Police force who had not been able to take part in a test track riding-session prior to riding the field trial machine were the most vehement in their condemnation of the system. It was unfortunate that their machine was one which had problems with distorted brake-discs. The effects of this were attrib-

1. System should be fitted to all new motorcycles. (20 riders)
2. System should be fitted to all new cars. (7 riders)
3. At all times the machine remained under control.
4. Would purchase an anti-lock system for own motorcycle. (17 riders)
5. A great advantage in rider safety. (7 riders)
6. Effectiveness for outweighs initial cost.
7. Excellent in the wet.
8. System prevented accident on wet road in London.
9. Would expect to see such a system as standard on large machines in future.
10. An advanced rider would never be in a position to need anti-lock.) made by same rider
11. When the anti-lock system was caused to operate it did prevent a major rear-wheel skid.) made by same rider
12. I can find no criticism of the system at all.
13. To be able to retain full control on slippery road under heavy braking is reassuring.
14. Rear anti-lock operation is jerky.
15. If needed only once in a lifetime could be a life-saver.
16. Good apart from vehicle dive.
17. System needs to look less clumsy.
18. System looks unattractive and gave unacceptable "grab and notchiness" (Result of disc run-out)
19. System quite good but would have benefitted from fitment of anti-dive forks.
20. Steering-head bearings needed tightening 3 times in 1500 miles.
21. For Police use not really necessary because of high standard of training.
22. A good idea for members of the public but could give a false sense of security.
23. Light suspension of vehicle gave unacceptable effect with anti-lock.
24. Amazing to experience the difference after replacement of defective brake discs.
25. Gave the confidence to ride hard and brake hard.
26. System excellent but will make normal riders lazy.
27. Overall a good system worth improving with a package which is neater.
28. System allows full use of front brake and will give good deceleration.
29. Too easy to bring rear anti-lock system into operation because of load-transfer.

Figure 4. Summary of main comments made by riders

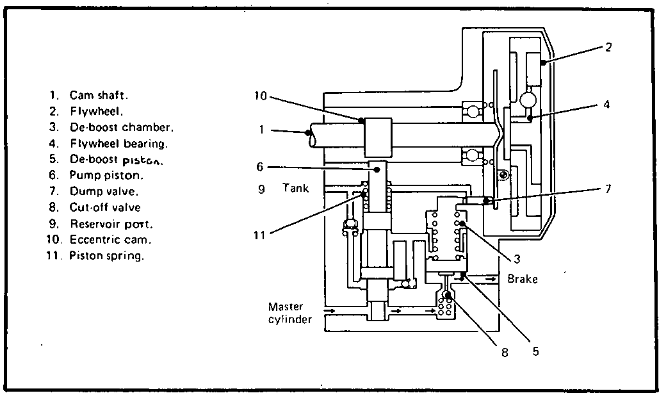

Figure 5. Sectional view of anti-lock brake unit

uted to the operation of the anti-lock system, probably because riders had not experienced the system working without these external influences. A retrospective test-track session was arranged for these riders, who then reversed their opinions. The need for particularly carefully-worded instruction material is apparent when such systems become available widely. Similarly problems and doubts have been expressed following a field trial with anti-lock fitted to passenger cars.(7)

Several problems and areas of adverse comment by riders have arisen as a result of defects in components of the motorcycle itself. Similarly, incompatibility between the operation of the anti-lock system and characteristics of the motorcycle has been a problem. These have arisen because the motorcycle and the anti-lock system have not been designed as an entity. This is inevitable at this stage; in a production form these failings would be eliminated by design.

Conclusions

1. The results to date of this trial have shown that the anti-lock system on the motorcycles has always worked correctly. Problems have concerned failings in ancillary components.
2. Some motorcycles seem to be more suitable than others for the installation of anti-lock. Careful choice of suspension characteristics to prevent exaggerated movement during cycling of the anti-lock system appears to be essential. Similarly the design of such components as steering-head bearings should be such that the cycling of anti-lock systems does not lead to the need for frequent adjustment. It was reported that with one model of motorcycle used in the trial, the use of police radio equipment interfered with vehicle electronic systems such as fuel-injection. This, perhaps, reinforces the decision to select mechanical rather than electronic anti-lock for motorcycle use.
3. It appeared to be essential to educate riders about what to expect of anti-lock systems. This has important implications when anti-lock becomes available commercially on a widespread basis.
4. The trial has been important in convincing interested parties of the viability of anti-lock under the arduous conditions of a motorcycle in the field. It has lead to interest being shown by major motorcycle-manufacturers and by potential users. For example, Essex Police have a policy of using anti-lock equipped vehicles throughout; the trial has convinced them that it is possible to extend this to their motorcycle fleet.

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Appendix

The Anti-Lock System

The system fitted to the motorcycle used in this trial was developed and manufactured by Lucas Girling Ltd. specifically for motorcycles. Although electronically-controlled systems were already manufactured by the company for use in cars and commercial vehicles, it was decided at the outset that the motorcycle system should be mechanically based. This was considered suitable for the following main reasons:

i) low cost,
ii) compact, self-contained unit,
iii) not affected by electrical system of vehicle or external electromagnetic interference,
iv) simple adjustment and servicing,
v) reliability in harsh environment of motorcycle.

A single hydro-mechanical assembly is interposed in the hydraulic line between the master cyclinder and the brake caliper, usually adjacent to the road-wheel (Figure 5). (An anti-lock unit is necessary for each road wheel which requires control.) A shaft within the unit is driven directly at a fixed ratio from the road wheel. In the case of the machines used in this trial, this drive is accomplished by means of a toothed belt. Two functions are performed by the rotating shaft:

i) to drive a cam-operated pump which provides hydraulic pressure for brake reapplication.
ii) to carry a small flywheel which senses road-wheel deceleration and acts as a speed reference.

Operation of the System

During braking which occurs at a level below the limit of tyre/road adhesion, brake-fluid passes uninterrupted through the anti-lock unit and applies pressure to the caliper in the normal way.

If braking is excessive, the flywheel overruns its shaft and causes the dump valve (7) to be opened, allowing fluid-pressure at (3) to fall. A pressure differential then exists between (5) and (3) and causes the deboost piston (5) to retract. The supply from the master cyclinder is isolated by the closure of the cut-off valve (8). These events result in a controlled reduction of brake-pressure and allow the road-wheel to return to a safe condition.

The fall in pressure at (3) causes the pump (6) to be forced into contact with a cam on the flywheel shaft. Fluid is circulated through (3), (7) and a reservoir port (9) back to the pump. When the road wheel has recovered, dump valve (7) closes and causes the pump (6) to re-pressurise the chamber (3) and move the piston (5) to reapply the brakes. This sequence of events continues until the vehicle comes to rest or the rider reduces the brake-force.

A more detailed description of the system has been published (CART 1985).

Safety Considerations of Motorcycle Lighting At Night (Written only paper)

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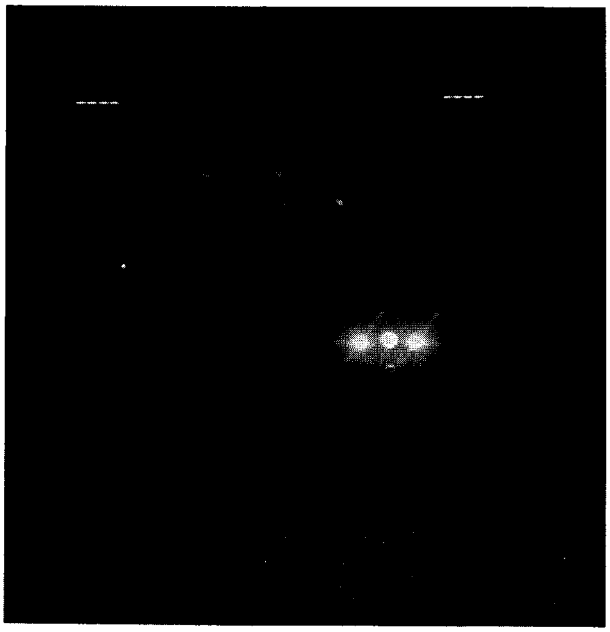
Abstract

Studies have shown that a large proportion of motorcycle accidents involve another road user's failure to see an approaching motorcycle. At night some of these accidents are associated with the misinterpretation of the visual cues given by motorcycles. These problems might be alleviated by the use of appropriate lighting displayed at the front of the motorcycle. Experiments have been conducted in darkness to discover what type of lighting-arrangements assist road users to see and make correct judgments about motorcycles in traffic conditions. Results showed that the detectability of motorcycles is related to the intensity and beam-pattern of the headlamp. Lighting which helped to define the form of the motorcycle, used in addition to the headlamp, aided identification in traffic. Although the use of daytime running lamps, in various forms, was found to be of no benefit at night it should not be inferred that specifications for improving lighting for day and night use are necessarily incompatible.

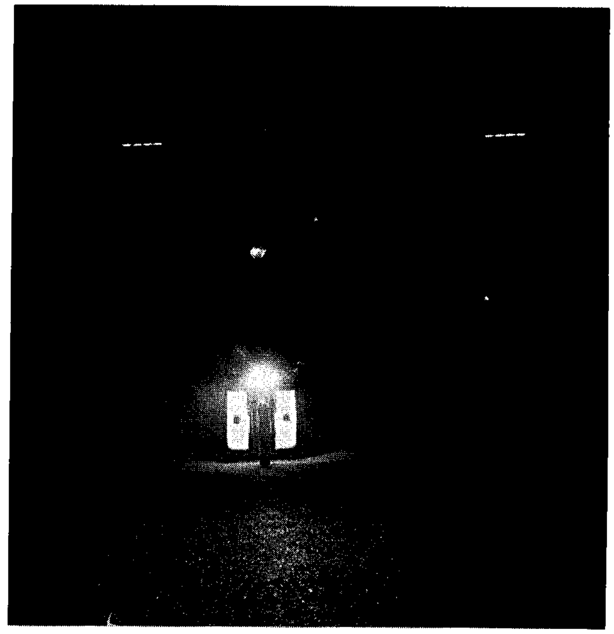
The functions of lighting on motorcycles

Lighting equipment on the front of motorcycles has two main purposes: to indicate the presence of the motorcycle to other road-users and, in darkness, as a source of illumination to enable riders to see their way and avoid obstacles.

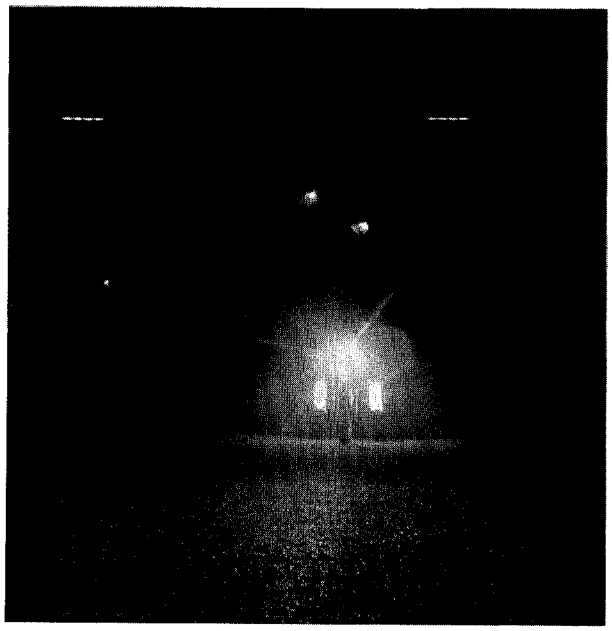
The principle source of illumination at night is the headlamp. In well-lit streets at night and, of course,



Running lamps.



Illuminated legshields.



Striplights.

Figure 1. The special lighting arrangements

in daylight the role of the headlamp is primarily that of indicating the vehicle's presence.

Considerations of vehicle lighting as a means of providing illumination are well-documented, for example(1)(2)(3). This paper is concerned with those aspects of motorcycle lighting which contribute to the indication of the vehicle's presence at night. The relative advantages of standard and novel forms of lighting for motorcycles are discussed.

Nighttime accidents and lighting

It is now accepted that a major factor in accidents which occur in daylight is the failure of road-users to see motorcycles in time to avoid a collision. A similar problem exists at night. Evidence of this is provided by detailed studies of accidents(4)(5)(6).

These studies indicate that in about one third of all collisions between motorcycles and other vehicles, the driver of the other vehicle claimed to have not seen the motorcycle prior to the accident. Using 1985 data it is estimated that approximately 15,000 accidents are of this type each year in Britain. Of these, approximately 3,000 occur during the hours of darkness. The accidents occur mainly at junctions, in urban areas with well-lit streets and involve manoeuvres in which other vehicles infringe the motorcycle's right-of-way.

Even though the use of motorcycles is much less at night than it is during daylight, it was estimated in the studies mentioned above that about one third of all accidents which involved a motorcycle occurred during the hours of darkness.

The evidence from accident studies suggests that, as in daylight, a significant proportion of motorcycle accidents is associated with some kind of perceptual error on the part of another vehicle's driver. But, unlike the daytime problem where it appears that drivers overlook the motorcycle, the difficulty of seeing motorcycles at night is complicated by errors in identification and interpretation as well as in simple detection. This indicates that there are several different ways in which good motorcycle-lighting should assist other road-users at night.

i) detection—it should be easy for other road-users to detect motorcycle lights against the background of other lights at night, both those on vehicles and from other sources such as shop windows and signs.
ii) identification—it should be possible to recognise the light as signifying a motorcycle; by virtue of their size and performance, motorcycles behave differently from other vehicles. Hence, misinterpretation of a motorcycle light as belonging to another type of vehicle can have dangerous consequences.
iii) judgment of location and speed—these judgments are more difficult to make about all vehicles at night than in daylight because visual cues are more restricted. The design of lighting must not add to difficulties in interpretation; if anything it should aim to assist these judgments. Currently in Britain, and in most other countries, motorcycles are required to display only a single headlamp during darkness. This arrangement has a number of disadvantages with regard to the important functions described above:

in detection: most other vehicles have two headlamps, many of which are also larger and more powerful than those fitted to motorcycles. This makes motorcycles comparatively difficult to detect.

in identification: motorcycles have only one headlamp and are thus likely to suffer misinterpretation because of fewer visual cues.

in speed and location judgment: there are special problems for motorcycles because the major visual cue of changing angular-separation between fixed points is not provided by a single lamp.

Hence it appears that the existing front lighting arrangement for motorcycles is not performing its function of providing the information required by other road-users at night. As a consequence, attempts have been made to investigate how existing lighting

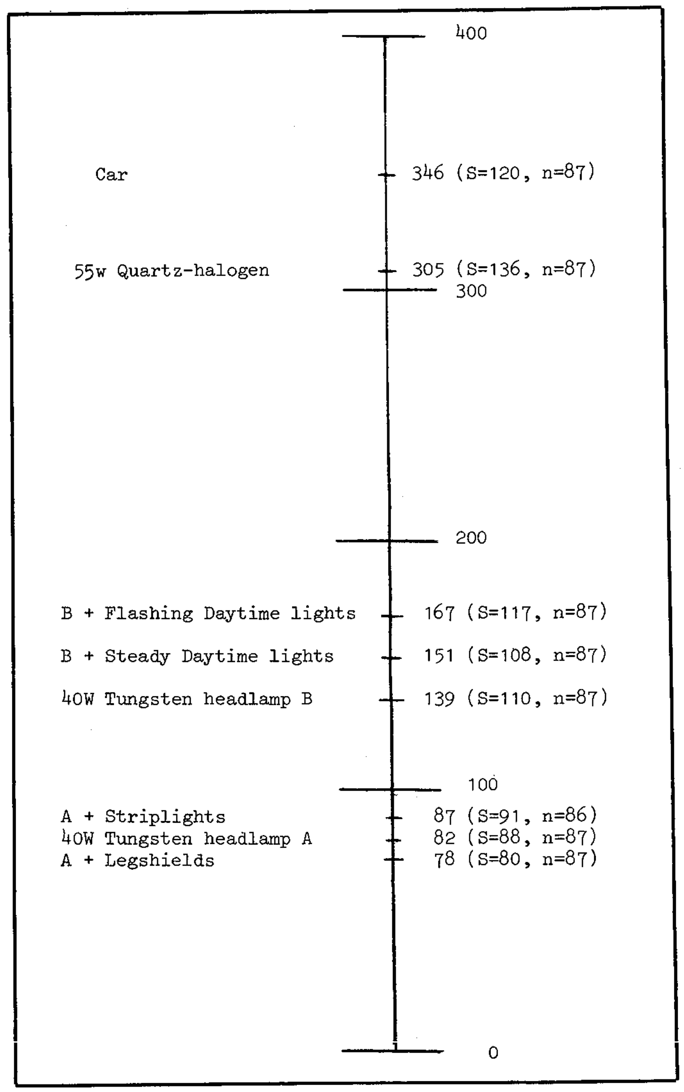


Figure 2. Results of peripheral-detection experiment Mean detection distances (m) from subjects' position (2 Motorcycles, A and B, were used with nominally-similar 40 watt headlamps). (S is standard deviation, n is number of observations)

might be modified, or supplemented, to make it more effective in meeting these requirements.

Experimental studies in lit streets at night

Various lighting arrangements were selected and tested to examine the ways in which they might contribute to the ease and accuracy with which observers perceived motorcycles.

Single headlamps of various powers (up to 55W quartz-halogen) and size (up to 180mm in diameter) were tested, both in steady-state and modulated form (at 3-4 Hz). These were also tested coloured by a yellow filter intended as a means of providing a unique signal to indicate a motorcycle. Various lamps were used in conjunction with the single headlamp in order to provide both additional illumination and a pattern of lights peculiar to motorcycles (Figure 1). These included the existing amber front direction-indicator lamps, wired so that both were permanently illuminated; pairs of handlebar-mounted daytime running lamps, both white and with yellow filters, steady-state and flashing; a pair of 300mm long strip-lights mounted vertically, one alongside each fork leg. The final arrangement used with the existing headlamp was a pair of white legshields, illuminated by a pair of running lamps. This and the strip-light arrangement were selected because they were believed to convey information about the form of the motorcycle to assist identification as well as providing additional light.

These lighting arrangements were tested in a series of experiments, each concerned with one aspect of the visual problem.

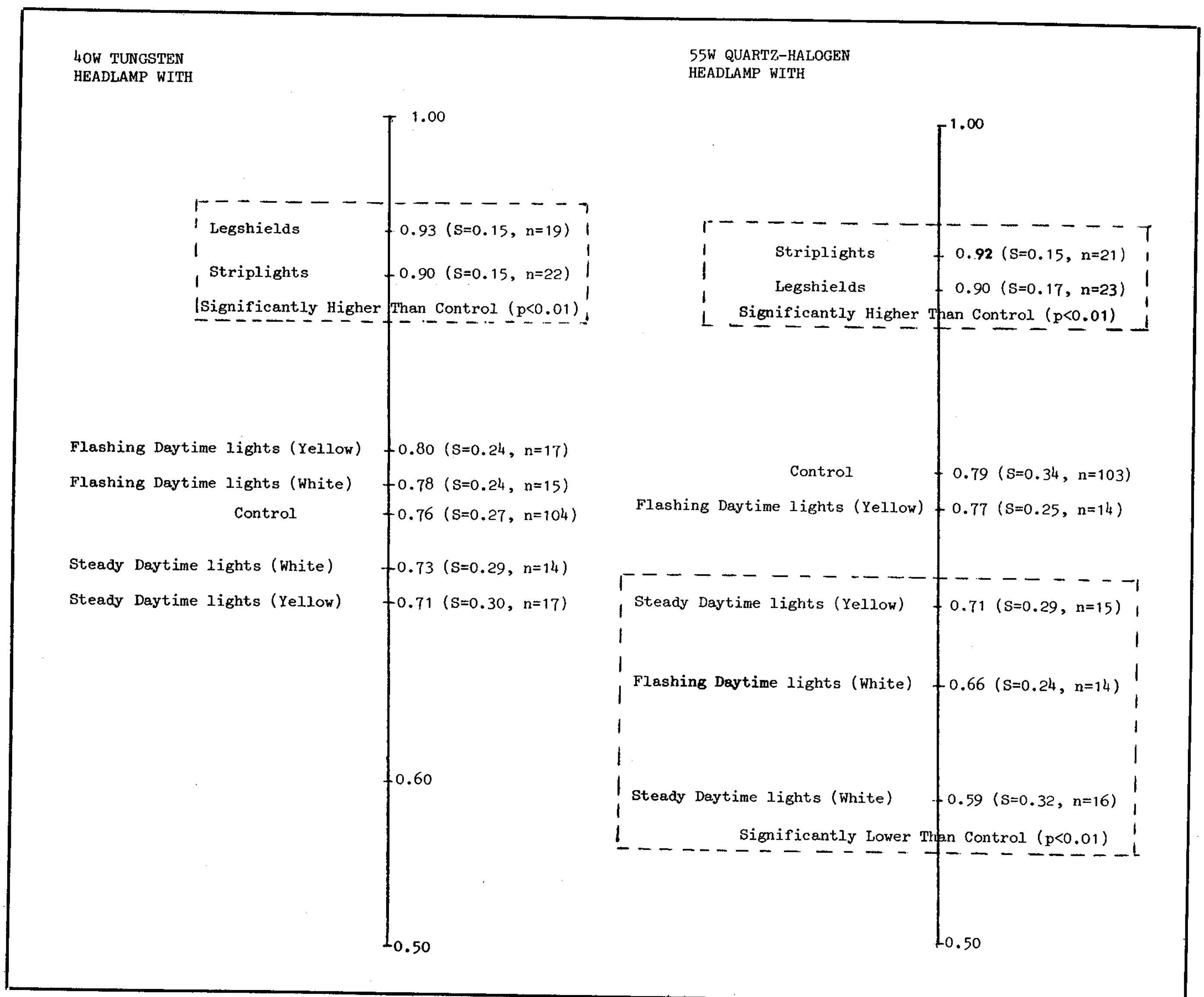


Figure 3. Results of identification experiment Mean proportion of correct identification of motorcycle. (S is standard deviation, n is number of observations)

The first was a peripheral-detection test. Observers were given a task to occupy their central visual-field and asked to indicate when they were aware of a vehicle which approached at an angle of 60° to their line-of-sight. The distances at which the motorcycle was detected were used as the measure of the detectability of the lighting-arrangements. A car was used to provide a control condition.

The second experiment was concerned with vehicle identification. Observers were given brief glimpses of groups of approaching traffic constituted in different ways, sometimes including the test motorcycle as the leading vehicle. They were asked to identify the leading vehicle. The test measure was the proportion of correct responses.

The third series of trials examined judgment of speed and location in two experiments. In both, the test motorcycle travelled towards observers at a range of predetermined speeds between 25 and 60 mile/h. In one experiment the motorcycle was obscured from the observers' view at a distance of 50m, and they were asked to judge the time for the motorcycle to reach them, without the aid of any further information, either visual or audible. In the second experiment, observers were asked to estimate approach-speeds.

Results from all these trials may be summarized, as follows:

i) Peripheral detection. (Figure 2). Most of the experimental arrangements were not as detectable as the car (control condition). It was evident that to improve the detectability of motorcycles it is necessary to increase the amount of light reaching other drivers' eyes. To achieve this the most simple and effective way is probably to increase the intensity of the motorcycle's headlamp rather than apply additional lighting. However, it is not as simple as just increasing the power output. The lamp's beam pattern is as important in determining the amount of light which reaches an observer's eye. For instance, it has been shown that a relatively low-powered lamp with an ill-defined beam-pattern can produce more light at an observer's eye than a 55 watt quartz-halogen lamp with a well-controlled beam. Stroud et al(1986). An ill-defined beam-pattern has serious disadvantages in respect of illuminating the road ahead for the rider. More light at an observer's eye should be achieved by using a high-intensity lamp with a well-defined beam.

ii) Identification of motorcycles in traffic (Figure 3). Motorcycles with illuminated legshields or striplights in addition to a headlamp were identified correctly in approaching traffic significantly more often than motorcycles using a headlamp only. Running lamps of both colours (white and yellow) and flashing white lights, each of which, in conjunction with the 55 watt quartz-halogen headlamp, performed significantly worse than the headlamp alone.
The effect of the illuminated legshields and the striplights was beneficial with either a standard 40w headlamp or a 55w quartz-halogen headlamp. Either arrangement used in combination with the latter headlamp would have the additional benefit of enchancing an observer's peripheral-detection performance.
The illuminated legshields and striplights performed equally well. From a practical viewpoint the legshields and the lamps used to illuminate them have the advantage that they are standard motorcycle accessories.

iii) Speed judgement (Figure 4). Experiments on speed judgement confirmed the findings of other research about peoples' tendency to underestimate high speeds. This was independent of vehicle type and lighting. The speed of the control motorcycle was underestimated to the greatest extent but this was generally not statistically-significant. In some circumstances, motorcycles using daytime running lamps (white or yellow, steady or flashing) had their speed estimated more accurately than those with only the control condition. However, this was not consistently the case.

On the whole, the results of these trials suggest that there would be advantages to be gained by increasing the light-output from motorcycle headlamps and from illuminating the form of the motorcycle. In the trials, the latter was achieved by illuminating legshields or by providing additional lamps which described the motorcycles' shape. These measures appear to offer improvements in both the detectability and the identification of motorcycles.

Introduction of dim/dip lighting

There is an interesting post-script to the work described above which could affect the needs of motorcycle lighting at night. A change was made to the U.K. lighting regulations which required vehicles sold after October 1986 to be fitted with the means to dim the dipped-beam of their headlamps to an intensity of approximately 10 percent of normal. This is combined with changes in vehicle-wiring which make it impossible to drive with only parking lights in use and drivers are encouraged to use the dimmed beams in well-lit areas at night. The purpose of the change in regulations is two-fold. First, it prevents vehicles being driven with only parking lights in use. Second, there should be a reduction in the problems which are presumed to arise as a result of glare produced by normal dipped headlamps when they are used in well-lit streets. Motorcycles are not included in the new regulation. TRRL with ICE has investigated the implications of allowing motorcycles to continue to be ridden at night, using a normal dipped-beam. (7). The experimental findings suggest that motorcycles should gain an advantage over other vehicles in terms of detection and judgement of their speed. This is because motorcycles will retain the intensity of their normal dipped beams in conditions in which other vehicles may use dimmed dip, i.e., in lit built-up areas. It is in these that the majority of multi-vehicle collisions occur. Detection of the motorcycle is essential before any other judgement can be made.

Conclusions

From the foregoing consideration of motorcycle lighting in well-lit conditions at night, the following conclusion can be drawn:

There appear to be benefits to be gained from motorcycles using a large and powerful headlamp at night. With currently-available designs of headlamp this means a lamp of at least 180mm in diameter and of 40w power, with a well-defined beam-pattern. The use of a headlamp of at least these dimensions has been shown to have advantages in daylight also.

The use of some types of accessory lamps has been found to have advantages. Unfortunately, those which are recommended for use in daylight—i.e., front directed daytime running lamps—are not beneficial at night. In fact, they have been shown to have a detrimental effect on correct identification of motorcycles in traffic. This conflict should not, however, be regarded as indicating that lighting-systems for day and night use are inherently incompatible. Those forms of additional lighting which were found to be advantageous at night (those which provided information about the shape of the motorcycle) have not been assessed in daylight. Moreover, one of these, the illuminated fairing, used a pair of running lights although in a different orientation from their daytime use. These findings should be regarded as a stimulus to the design of forms of lighting which can meet the night time requirements of motorcycles and be compatible with the needs of daytime conspicuity.

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