PROCEEDINGS VOLUME I

Fifteenth International Technical Conference on the Enhanced Safety of Vehicles

WORLD CONGRESS CENTRE
MELBOURNE, AUSTRALIA
13 -16 MAY 1996

ENHANCED • SAFETY • OF • VEHICLES

Federal Office of Road Safety
Australia

U.S. Department of Transportation
National Highway Traffic Safety Administration

The Fifteenth International Technical Conference on Enhanced Safety of Vehicles

Proceedings Volume 1

U.S. Department
of Transportation

**National Highway
Traffic Safety
Administration**

The Fifteenth International Technical Conference on Enhanced Safety of Vehicles

Sponsored by:
U.S. Department
of Transportation

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Traffic Safety
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Hosted By:
Federal Office of
Road Safety
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Held At:
Melbourne, Australia
13-16 May 1996

Foreword

This report of the proceedings of the Fifteenth International Technical Conference on the Enhanced Safety of Vehicles was prepared by the National Highway Traffic Safety Administration, United States Department of Transportation.

We wish to thank the authors and all those responsible for 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. Papers in each of the technical sessions are numbered sequentially. Due to the withdrawal and/or transfer of papers from one session to another, breaks in the number sequence may occur.

Introduction

The International Experimental Safety of Vehicles (ESV) Program originated under NATO's Committee on the Challenges of Modern Society (CCMS) and was implemented through bilateral agreements between the United States Government and the governments of France, the Federal Republic of Germany, Italy, the United Kingdom, Japan, and Sweden. The participating nations agreed to develop experimental safety vehicles to advance the state-of-the-art in safety engineering and to meet periodically to exchange technical information on their progress. Over time the focus of the Conference has shifted from concentration on the development of experimental safety vehicles to broader issues of motor vehicle safety. In addition the number of our international partners has expanded. The governments of Canada, Australia, the Netherlands, Hungary and Poland are also active participants. In 1991, the name of the Conference was changed to "International Technical Conference on the Enhanced Safety of Vehicles" (ESV) to reflect these broader issues.

To date, fourteen international conferences have been held, each hosted by one of the participating governments. These conferences have drawn participants from government, the worldwide automotive industry, and the motor vehicle research community. The 15th ESV Conference, held in Melbourne, Australia, was attended by the largest number of countries since its inception. Participating countries included Belgium, Canada, Denmark, France, Germany, India, Indonesia, Ireland, Japan, Korea, Malaysia, New Zealand, Poland, Russia, Singapore, Spain, Sweden, the Netherlands, the Philippines, the United Kingdom, the United States, and Australia. International cooperation in motor vehicle safety research continues at the highest level.

The proceedings of each Conference have been published by the United States Government and distributed worldwide. These reports, which detail the safety research efforts underway worldwide, have been recognized as the definitive work on motor vehicle safety research.

We are certain that this outstanding example of international cooperation seeking reductions in motor vehicle deaths and injuries will continue its past success.

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Peter Makeham

Director, Federal Office of Road Safety
Australia

Your Excellency Sir William Deane and Lady Deane, the Honourable John Sharp, Commonwealth Minister for Transport and Regional Development, Mr. Robin Cooper, Parliamentary Secretary for Transport, Roads and Ports in Victoria, Dr. Ricardo Martinez, Administrator of the United States National Highway Traffic Safety Administration, Delegates from around the world.... Ladies and gentlemen....

I am very proud, as Director of the Federal Office of Road Safety, to bid our overseas guests a warm welcome to Australia, and to Melbourne in particular. I also warmly welcome delegates from throughout Australia. Hosting the opening of the 15th ESV is one of the important events of my career.

The Federal Office of Road Safety is committed to leading Australia towards the world's best road safety practice. This can only be achieved by actively interfacing with the world's best practitioners. So it is particularly significant that today, the world's best practitioners in vehicle safety design and technology have come to our doorstep. I know you will be surprised at some of the advances Australia has made in vehicle design and safety. But Australia will continue to look abroad for best practice and work towards harmonisation as a means of achieving greatest benefit for the greatest number of people.

We intend making this a visit you will not forget not only for the fellowship you will enjoy with colleagues from throughout the world -- nor for being able to enjoy the delights of Australia but for being given the opportunity for advancing vehicle safety through your deliberations over the next four days.

I would like to make special mention of the close co-operation and support this conference has received from the State Government of Victoria and from the United States National Highway Traffic Safety Administration - NHTSA - on whose behalf Australia is hosting this conference.

It now gives me great pleasure to invite Mr. Michael Brownlee, Associate Administrator of NHTSA, to join me in welcoming you here today.

Michael B. Brownlee

Associate Administrator, Safety Assurance, National Highway Traffic Safety Administration
United States

Thank you Peter, Your Excellency Sir William Deane and Lady Deane, Mr. Sharp, Mr. Cooper.
It is a privilege for me to co-chair this conference with Mr. Peter Makeham and to be part of this historic event -- the 15th ESV Conference taking place for the first time in the Southern Hemisphere.

This Conference is unique in that it is the only scientific conference that brings together the worlds leading researchers, academia, automotive executives and government leaders for the purpose of sharing up to date research technology in the field of motor vehicle safety.

For the past two years we at the National Highway Traffic Safety Administration have been working with the Federal Office of Road Safety to bring you the most scientifically sound conference possible.

Over the next three days you will be scientifically challenged with over 180 technical presentations being presented by some 100 authors. This challenge, will take place in an atmosphere that is congenial, relaxed and well orchestrated, thanks to our hosts, the Federal Office of Road Safety, the Australian Government and the Australian Automotive Industry.

We are most thankful to the Australian Government for hosting this 15th ESV Conference. I believe I speak for all of us here today when I say that Australian hospitality is only surpassed by their commitment to motor vehicle safety.

Opening Address

Sir William Deane

Governor-General of the Commonwealth of Australia
Australia

Mr. Sharp, Mr. Cooper, Dr. Martinez, Mr. Brownlee, Mr. Makeham, Ladies and Gentlemen.

This 15th International Technical Conference on the Enhanced Safety of Vehicles has been a long time in the planning. It is an important event for this country. Indeed, in a context where, world-wide, more than 500,000 human beings die each year in motor vehicle accidents and where the motor vehicle is intrinsically intertwined in the fabric of society in all countries, it is an event of global importance. Let me, as Governor-General of Australia and speaking for all Australians, tell all who are participating in the conference that we are delighted and honoured that you have decided to gather here in one of our country's great cities. Let me also, without listing particular names, congratulate and thank all those who initially succeeded in having our Federal Office of Road Safety selected as the host and Melbourne designated as the host city and all those who, in the intervening years, have worked so hard to bring the conference to the stage where it is now commencing.

This is the first time the biennial ESV Conference has been held in the southern hemisphere. It is truly an international congress. Those who have gathered here come from many countries spread across Asia, Europe, Africa and the Americas and, of course, from Australia. Between them, they include leaders and experts in all aspects of the life of the motor vehicle: manufacture of components, birth in the assembly line, testing of and trade in the finished product, regulation of its use, its actual use and its ultimate decease whether peacefully in a wrecker's yard or violently in a collision on the road. The principal speakers at the conference are from Japan, Canada, the Netherlands, Sweden, Poland and the United States of America, in addition to Australia.

My own first clear memories of a motor vehicle date back some sixty years, to about 1936 when I was five years old. At the time, my family lived in Canberra to where we had moved from this city. My two sisters were boarders at a school in Yass which is about seventy kilometres from Canberra. The particular memories are of a journey from Canberra to Yass taking my sisters back to school at the end of school holidays. They have a nightmarish quality.

Our family car in 1936 was a Chevrolet. It was one of those cars without a fixed top. If it rained, one would become soaking wet as one struggled to put into place a collapsible top which somehow nestled at the back of the back seat where it invariably became obstinately stuck. It was then necessary to seek to attach the celluloid windows which in fact never fitted. My father called the car "a Chev tourer". Everyone else called it "the old Chev". He had bought it secondhand. It was old and infirm long before it came into our lives. It disliked us all.

On the Canberra side of Yass there was a very steep hill. At least to the Deane family in the old Chev it seemed to be very steep. The particular memory commences as my father confidently approached it: my mother sitting beside him in the front and the three children in the back. The day was a very hot one and the car began to struggle a third of the way up the hill. My mother said: "it's not going to make it." My father ignored her. Two-thirds of the way up, it stopped. The engine spluttered and cut out and the car rolled backwards a little way as my father applied the hand brake. Then, upon instruction, we children alighted and pushed while my father turned the car around to the other side of the road to face towards the bottom of the hill. We then all got on board again and somehow, as the car ran down the hill propelled by its own momentum, my father managed to re-start it. When we reached the bottom of the hill with the engine going, he turned it around and faced the hill again. This time, all the passengers, including my mother, were ordered to disembark and walk up the hill as my father determinedly drove the vehicle to the top. Like Don Quixote, however, he did not learn from the experience.

The next time we made the journey he again attempted to drive the fully laden car up the hill. Again it stalled two-thirds of the way up. And the whole endeavour was repeated. The reason which leads me to recount that overlengthy family story is, I expect, obvious. A conference of experts such as this could, no doubt, identify an innumerable number of dangers to which we and other users of the road were exposed as, by one means or another, we travelled up and down that hill. When I add the facts that the brakes of the old Chev were unreliable and that the road was without centre line or other signs, one has about as good a negative case example for a conference on the enhanced safety of vehicles as one could find. Indeed, the only thing going for the old Chev was that it could not go very fast. But even that was, in itself, a safety hazard in circumstances where about every other vehicle on the road could, even then, travel much faster.

In the years that followed the time of our old Chev, the motor car became progressively faster and, as a consequence, a progressively more dangerous instrument. From the point of view of those inside the car, the danger of greater speed was lessened by greater structural strength and added reliability of engine, brakes and other accessories. On the other hand, the greater structural strength increased, rather than diminished, the danger of greater speed in so far as those outside the vehicle were concerned. What really placed the focus of society upon vehicle safety was, however, the ever growing numbers of deaths and injuries resulting from the ever increasing numbers of vehicles on the roads of this and other developed countries. And, of course, in this and other countries, growing awareness of the lethalness of the cocktail of speed, petrol and alcohol.

Nonetheless, even in the seventies, there were, at least in this country, serious objections raised to the restriction of liberty involved in compulsory measures aimed at enhancing road safety. In particular, I remember that it was, at times, almost impossible to turn on the radio or television or to pick up a newspaper without hearing or reading some protest at the erosion of liberty involved in being required to wear a seat belt or to subject one's self to a compulsory breath test in circumstances where one may not have done anything wrong. Even the most ardent civil libertarian acknowledges, however, that civil liberties must, at times, be adjusted or qualified to meet compelling social needs. It has long been accepted in this country and elsewhere that the dangers of the motor vehicle are such that those who elect to use it or to travel in it can properly be required to submit themselves to measures designed to protect other members of society or themselves.

These days, there is universal recognition of the importance of taking all practical steps to enhance the safety of motor vehicles and their use. As each new safety feature is invented, developed or identified, it is embraced by the public, even if it comes at a price. In the period since the seventies, random breath testing of drivers has become universal in this country. In addition, to name but some of the many new measures or developments, we have had: a new generation of brakes; improved vision in most cars; improved impact design; safer seats; compulsory wearing of helmets for cyclists and motor cycles; improved safety standards for large commercial vehicles, passenger vans and four-wheel drives; driver's airbags, then passenger airbags and now side airbags; higher speeding fines and greater penalties for driving under the influence; greater driver awareness; radar; red light and speed cameras. Those measures and developments have achieved much. Australia's road toll has dropped by over forty percent since 1971, from nearly 3600 to just over 2000, even though the number of vehicles on the road has doubled, from nearly 4.8 million in 1971 to more than 10.9 million in 1995. Much remains, however, to be done. The continuing toll of death and injury must be further reduced by identifying individual and specialised areas for improvement - and one by one addressing them. In that regard, we like others, turn to this conference for instruction, inspiration and guidance.

In the years since the ESV Conferences originated under the Nato Committee on the Challenges of Modern Society, they have become the premier international fora for exchange of ideas in vehicle safety technology. The conferences encompass research and exchange of ideas over the full spectrum of motor vehicle safety issues. They draw participants from government, the worldwide automotive industry and the motor vehicle safety research community. They have contributed much to the enhancement of the safety of motor vehicles and their use. I am confident that this conference will continue the proud tradition set by its predecessors and do much to enhance the safety of the use of motor vehicles throughout the world.

Over the next four days, you will have the opportunity to hear papers on 13 major subject areas and make judgments on how each paper can contribute to your individual work. Because vehicle safety is so important, worldwide, each of you faces the challenge to make sure that the latest thinking and research is disseminated as widely as possible. As I have said, more than half a million people die each year as a result of road crashes worldwide. It is only through applying the best minds of industry and government to the question of reducing that awful number by enhanced safety measures that further significant progress will be made. That task is yours. I wish you well as you work to discharge it.

There is one particular subject to which I would make specific mention. I notice that worldwide harmonisation of vehicle safety standards occupies the major part of this afternoon's program. As you may know, Australia and New Zealand have agreed to start work on a harmonisation regime in respect of new vehicles. This country also supports moves by the Asia-Pacific Economic Community - APEC - to facilitate trade through harmonisation of standards, mutual recognition of conformity and transparency in standards throughout the region. We are also strong supporters of harmonisation at the broader international level through the United Nations. Hopefully, your program this afternoon will help set the path for future developments towards general harmonisation.

Finally, to all who have come from overseas I say, on behalf of all Australians, that I hope your visit to our country is a happy one. I hope that the great professional benefits you will obtain from this Congress are matched by the personal and social benefits as you renew old friendships and make new friends among your international colleagues. I hope that you come to know as much of our country as is possible in the period you are here. And I hope that, when the time comes to leave us and to return to your homes, you carry with you fond memories of Australia and Australians.

I now declare open the 15th International Technical Conference on the Enhanced Safety of Vehicles.

Keynote Addresses

The Honorable John Sharp

Commonwealth Minister, Transport and Regional Development
Australia

Your Excellencies, the Governor-General and Lady Deane, Mr. Cooper, Dr. Martinez, Mr. Brownlee, Mr. Makeham, delegates.

The 15th ESV which we are attending today is, without doubt, the premier world forum for the exchange of ideas in vehicle safety technology -- so today represents a major step forward for Australia in the field of vehicle safety. Motor vehicle safety is vital to Australia's economic performance and social cohesion. We all benefit from enhanced safety. Government and industry must continue to work together to better our present performance.

Australia led the world in the area of occupant protection by mandating the fitting of seat belts, first for front seats followed by rear seats for passenger cars. Its effect was built-on with the progressive introduction and subsequent strengthening of Australian Design Rules governing vehicle design and construction. Seat belt legislation marked the beginning of a spectacular decline in Australia's road toll. Fatalities fell from a high of nearly 3,800 in 1970 when Victoria became the first to mandate the wearing of seat belts -- to the current level of about 2000. Other states soon followed Victoria with very strong community support.

That simple measure saves 1700 lives a year -- the population of a typical Australian country town. In monetary terms, compulsory wearing of seat belts saves more than $1.3 billion every year in direct costs and lost productivity. Outside North America, Australia is the only country to adopt requirements for passenger car side impact protection and we led the world in the 1980s by developing a frontal impact protection requirement for forward control passenger vehicles.

In the 1990s, Australia and Japan adopted injury-based performance requirements modelled on United States standards to protect passenger car occupants in full-frontal crashes. Our bus standards are the most stringent in the world. For two years now, it has been a requirement for all new coaches to be fitted with lap-sash seat belts in all seats. Our system of Australian Design Rules -- or ADRs -- has led the way in bettering Australia's road safety performance. But we are capable of even further improvement?

In addition to bus safety and seat belts, the ADRs also impact on other important economic factors such as vehicle emissions and fuel efficiency. Yet the Labor Government used the ADRs as a market-intrusive mechanism -- as a trade barrier. We are committed to seeing the ADRs used to facilitate Australia's national interest in:

- Safety
- Emissions control
- Energy efficiency; and
- International standards harmonisation

...... without creating barriers to trade.

Finally, the ADRs and the process by which they are implemented must be transparent in order to provide maximum possible information to the consumer. In spite of our efforts to date, road trauma still costs the Australian economy more than $6 billion a year. This is 60 per cent of the cost of all cars manufactured in Australia in any year. Just think of the jobs that cost represents. Human factors play an important role and must also be pursued in reducing this unacceptable cost to the community. To these ends, Australia has adopted a strategic planning approach.

We have set ourselves an objective of reducing road deaths to 10 per 100-thousand population by the year 2001, with similar reductions in injuries. High on this Government's agenda for achieving this goal is our Black Spots road funding program which will see $36 million spent over each of the next three years -- $108 million in total -- to eliminate the worst accident sites on our roads, be they federal, state or local roads.

This Government is also committed to spending $750 million over ten years on improving safety and travelling times on the Pacific Highway -- the coastal link between Sydney and Brisbane, which has been stained with blood. This highway will, at last, be declared a road of national importance. Labor neglected our roads over its term of government. Fuel excise rose from six cents a litre to 35 cents -- $10.3 billion a year -- while real spending on our roads dropped from $650 million to $640 million in real terms -- leaving aside inflation.

As well, our strategy includes encouraging people to understand that road trauma is a community issue, and bringing together the sometimes disparate objectives of the various community groups involved. We now have a commitment from all - outlined in Australia's first ever National Road Safety Action Plan. Broader community involvement is also required on the technical front. We must consult thoroughly with the vehicle manufacturing industry, automobile associations, consumer groups and government to develop greater transparency. This can only benefit the consumer. Safety and secrecy are not compatible.

Mere achievement of an ADR sets a minimum desired standard. Full publication of the process encourages much better performance by all concerned and engenders participation by all parties -- including the consumer. This is happening in Australia where we have vigorous automobile associations in all the states representing the consumers, and a very pro-active chamber of automotive industries representing the auto industry.

Our system of ADRs enhances competition not only in terms of safety; but also by showing benefits to consumers in:

- Production efficiency,
- Reduced emissions, and
- Fuel efficiency,

ADRs can benefit other road users -- not just drivers. Such transparency also helps identify duplication of effort -- a luxury we cannot tolerate. Research too must make best use of resources and ensure that maximum benefit is made from its findings. Australia is moving ahead with its research agenda which includes:

- Participating in an international committee to develop a harmonised offset frontal crash test procedure,
- Mandating more stringent side impact protection requirements; and
- Working on ways to improve pedestrian safety.

In developing vehicle safety standards, we must, because we are not a major producer, take into consideration the global nature of the vehicle industry - more and more manufacturers are designing vehicles destined for a global market. Our Design Rules must be more closely with international standards. More than 60 per cent of the Australian vehicle requirements harmonise with United Nations standards -- but that is not enough. Manufacturers around the world have indicated strong support for harmonised standards to achieve road safety benefits as well as reduced development costs and production lead times.

This approach, of course, benefits consumers as well as manufacturers, and is consistent with Australia's policies on international competitiveness. Harmonisation is one of the areas where I expect progress to be made at this week's conference. Road safety benefits aside, harmonised vehicle standards have the potential to expand marketing opportunities for all countries with an automotive industry - bringing attendant economic benefits.

In the Asia-Pacific region, Australia has played a leading role in the APEC Transportation Working Group Road Transport Harmonisation Project. Our efforts here have been widely supported by the international motor vehicle industry. In relation to European standards, the Australian-European Union Mutual Recognition Agreement, when completed, will allow mutual acceptance of each other's conformance procedures. This, in turn, will help facilitate bilateral trade.

Australia is also collaborating with the United States, Japan, the European Union and South Africa to revise the United Nations Agreement and to encourage its adoption by the wider world community. And - closer to home - Australian and New Zealand authorities are presently negotiating the harmonisation of vehicle standards and procedures within the Trans-Tasman Mutual Recognition Arrangement. This country needs a viable vehicle-building industry and vehicle components industry.

We have proven expertise in designing and building vehicles for Australian conditions and finding export markets for them in other parts of the world which also have to cope with long travel distances and rugged terrain. We need those export dollars -- and the jobs they bring. By the same token, our consumers should have access to the best automobile technology available - be it domestic or imported. Most importantly, whatever vehicle is sold on the Australian market should be the safest available. The Australian Government will be closely watching the outcomes of this conference.

On behalf of the Australian Government I wish you well.

Robin Cooper

Parliamentary Secretary, Transport, Roads and Ports
Australia

Your Excellency the Governor-General, Sir William Deane and Lady Deane, Federal Minister for Transport and Regional Development, the Honorable John Sharp, Distinguished Guests, Ladies and Gentlemen.

It gives me great pleasure to welcome you all to Melbourne on behalf of the Victorian Government. Melbourne is the home of Australia's vehicle manufacturing industry and we are proud of both the city and our achievements in vehicle manufacturing. By hosting the conference in Melbourne we are able to offer some very exciting technical tours including a visit to Holden's proving ground safety development centre, Autolivs new barrier crash and development complex, Toyota's new car plant and Monash University's wind tunnel, crash test facility and driving simulator.

Victoria has an excellent road safety record, one of which we are justifiably proud. Victoria has successfully campaigned to reduce the road toll over the last three decades. Victoria committed itself to reducing the road toll and backed this up with the resources needed to develop a wide range of road safety initiatives to achieve this outcome. The work undertaken by Australia's vehicle manufacturers in the area of vehicle safety has certainly contributed to our enviable road safety record.

To have a conference as important as this one held in Melbourne is a reflection of the world-wide respect Australia's vehicle manufacturers are earning and gives us an opportunity to share the rapid developments in motor vehicle technology which are happening in the Asia Pacific Region. Melbourne is a beautiful city and is graced by extensive parklands, and botanic gardens of world class standards. It is a cosmopolitan city and has a unique balance of graceful old and new buildings. Melbourne is a leader in theatre, sport and fashion and boasts some of the finest restaurants in the country.

I am sure your stay in our city and the surrounding area will be a pleasant one. There is much to see and do whether it is riding our trams around the city, strolling through the gardens or sampling the many different cuisines on offer. I have much pleasure in welcoming you to our city and I am sure you will all enjoy what the city offers.

I would like to invite you all to the Welcome Reception this evening on behalf of the Victorian Minister for Roads and Ports the Honourable Geoff Craige, MLC who unfortunately could not join us this morning for the opening ceremony due to Cabinet commitments.

Ricardo Martinez, M.D.

Administrator, National Highway Traffic Safety Administration
United States

Thank you Peter Makeham, Your Excellency Sir William Deane and Lady Deane, The Honourable John Sharpe and Mr. Cooper, Ladies and Gentlemen.

It is a pleasure to be here in Australia as head of the U.S. Delegation for the 15th ESV Conference. This is my first visit to Australia, and I commend the Australian's for their gracious hospitality. This also is my first ESV Conference. I look forward to attending the technical sessions and meeting with many of you over the next four days to learn of your thoughts on motor vehicle safety for the 21st century. Speaking of firsts, I would be remiss if I did not acknowledge that this is the first time the ESV Conference has been held in the Southern Hemisphere. Peter Makeham and his staff at the Federal Office of Road Safety, His Excellency Sir William Dean, and the Australian Government are to be commended for their leadership and vision. By bringing this vital safety conference to the Southern Hemisphere, they have opened the door to share motor vehicle safety technological advancements with countries that have not had the opportunity to participate in the past.

Secretary of Transportation Peña sends his personal greetings for a successful conference and expresses his regret at not being able to attend this important motor vehicle safety conference --- especially since he has made safety his number one priority.

There are many aspects of motor vehicle safety that I might address. Today, I would like to focus my remarks on the future of auto safety research, technologies and injury control, and describe our current and future challenges. First, let me briefly discuss the road that vehicle safety research has traveled.

Dr. William Haddon served as the first Administrator of the National Highway Traffic Safety Administration when it was founded 30 years ago. As a public health physician, he examined the injury problem from the epidemiological perspective of Host, Agent, and Environment. He attacked the crash problem by creating his famous nine cell matrix addressing the three components that affect automobile crashes -the Human, Vehicle, and Environment through the three time phases of crash occurrence -- pre-crash, crash, and post crash events.

Dr. Haddon also pioneered the multi-disciplinary approach that is so obviously represented in this audience. Medical researchers, scientists, and engineers, working together to build a safer world for motorists. Building on this vision, NHTSA recently created a seven trauma hospital medical and engineering research network that will be computer-linked nation-wide. This network uses four established NHTSA programs with the three projects that GM has initiated into a single, geographically diverse research program. The objectives of this unique program are to foster system-wide improvements in the prevention, acute care, and rehabilitation phases of crash injuries. This new network will bring constant and dynamic dialog between the medical, scientific, engineering academic and practitioner communities to investigate and analyze crash injuries in unprecedented collaboration and detail. This will allow faster and smarter triage and treatment, provide early understanding of the effectiveness of vehicle design and increase the learning and effectiveness by all parties involved.

During the past 30 years, we have seen many technological advances in automotive safety. Most of the safety improvements in today's vehicles were made possible through research by governments, private industry, and individuals represented at this conference. Through the ESV program, you were the first to recognize the need to share information and work cooperatively towards a common goal -- reducing the number and severity of motor vehicle fatalities and injuries world-wide. In the United States, many of these advancements have found their way into production and have played a major role in improving safety in the United States. While great strides in safety have been made, we are not finished yet. Over the past 3 years, the U.S. fatality rate has been steady at 2.74 per one hundred million KM. The increase in vehicle miles traveled has resulted in an increase in deaths, now at 41,700. This trend of flat fatality rates is characteristic of many countries. The challenge that we all face is how do we achieve substantial reductions in the fatality rate in the future. Cast against this challenge is that many changes are taking place that make this challenge formidable. To name a few:

- Decreases in funding levels for research
- Government down-sizing
- Shifts in the role of the federal government, with increased emphasis on deregulation of programs
- Decision-making placed at the state and local levels
- Privatization of research and testing facilities, and
- A more global economy.

One important burden is the cost of motor vehicle crashes. According to a recent World Bank Development Report, motor vehicle injuries are one of the leading drains on the world's economy. In the United States alone, the economic cost of motor vehicle deaths, injury and property damage are over $150 billion annually. The medical treatment of motor vehicle injuries costs $17 billion. The economic cost is only one consequence of motor vehicle crashes. People injured in crashes often suffer physical pain and mental and emotional anguish well beyond their economic loss.

It is clear that we can no longer do business as usual and continue to make progress. At the National Highway Traffic Safety Administration, about 2 years ago, we began re-examining our way of doing business. We recognized that we already had made the easy gains in vehicle crashworthiness, increased use of safety belts, and reductions in drunk driving.

Injuries can be prevented, but only if we understand the nature of the injury problem. The importance of data collection and analysis has been at the forefront of addressing motor vehicle safety research. For the past 20 years, NHTSA has been collecting crash data on the vehicle, the occupant, the crash scene, safety belt usage, etc. We have learned, however, that crash data alone are unable to convey the significance of the injury problem in terms of its impact on quality of life. Data linkage is one key to understanding what we should focus on first. By linking the crash, the vehicle and the occupant characteristics to their specific medical and financial outcomes, we can identify the factors that increase risk. To that end, I would like to strongly encourage you to work cooperatively in developing internationally compatible data sets that can be used to identify the critical components of automotive research. These are crucial steps in reaching common understandings of problems and solutions.

In 1993, the U.S. Government Performance and Results Act directed the United States Government agencies to develop strategic plans to establish performance goals in an objective and measurable form. NHTSA was selected to take part in this pilot program. As part of NHTSA's project, we set annual performance goals and program outcomes, and shared them with our partners in the form of a Strategic Plan followed by our Strategic Execution Plan. Several of the goals set forth in this plan directly relate to the objective of this conference. I would like to briefly describe these goals and share with you our efforts to date in meeting them.

International Harmonization

One of NHTSA's international initiatives is a proposal to establish an international harmonized research agenda. In November 1995, I met with the ESV Government Focal Points in Geneva, Switzerland, and presented a six point plan. Since then, we have been working together to develop such an agenda. For the first time progress has been made. A list of priorities has been suggested, a request to identify lead countries has been made, and a preliminary process has been proposed. I now ask each of you to join us to make this goal a reality. With all of our international research efforts to date, we have not arrived at a common set of injury criteria, or developed internationally accepted surrogates to assess injuries in crashes. This points up the fact that, collectively, we need to develop programs to answer these questions. Harmonized research is one approach to addressing this issue. This will be discussed further during the afternoon session on "Opportunities for World-wide Harmonized Safety Regulations." Working together, we can move towards harmonized regulations by providing a solid research foundation.

Another important goal for my agency is to mitigate the consequences of motor vehicle crashes.

Examining our Crashworthiness efforts, we believe that we are on the cutting edge. Working cooperatively with many of the countries represented here, through the European Experimental Vehicles Committee (EEVC), Japan Automobile Manufacturers Association (JAMA), and the Japan Automobile Research Institute (JARI) we have developed an advanced frontal impact dummy. I am pleased to tell you this dummy is being displayed for the first time here at the 15th ESV.

Pursuit of research and development efforts in biomechanics is essential for our continued success in mitigating and preventing automotive impact injuries. Our biomechanical research is also applicable to mitigating impact injuries in other transportation modes and in other fields such as industrial and product safety where impact injuries occur. Accordingly, we are in the process of establishing the first intermodal National Transportation Biomechanics Research Center within the Department of Transportation. In addition to pursuing the traditional biomechanics research, this research center will be computer-based, create new. research tools, work globally and serve as a repository of biomechanics research information. All of this will make our research faster and more efficient. We have strengthened our constituencies -- brought in new stakeholders, and new partners -- in the government, the academic, the medical community, and the research organizations to gain more information. We anticipate that this center may be able to serve as resource for many of you, and a potential partnership with some of you.

There is one more goal which I believe sets the tone for where we are going to reduce the number and severity of crashes.

The wave of the future lends itself to focusing our efforts on collision avoidance -- that is, to reduce the risk of crash occurrence. NHTSA is actively pursuing the use of intelligent transportation system technologies (ITS) towards developing crash avoidance concepts. These systems are intended to prevent roadway departure crashes, rear-end crashes, intersection collisions and lane change and merge crashes. We are actively working to address the problems of driver drowsiness, driver vision deficiencies, and automatic notification to emergency response units when a crash occurs. Our preliminary assessment of ITS benefits is that over 1 million accidents per year could be avoided when the ITS technologies are fully deployed in the fleet.

To complement these research efforts, we are building the tools necessary to understand driver behavior under normal and other, more demanding, driving conditions. Our most recent project is a state-of-the art driving simulator which will allow human factors research --- under a full range of driving conditions without exposing drivers to undue risk. These activities will help in understanding and evaluating countermeasures to reduce the 90 percent of crashes which are attributed to driver error. The simulator will also allow us to measure the crash avoidance effectiveness of some current vehicle standards and be a useful tool in regulatory harmonization both now and in the future. The simulator is expected to be fully operational by early 1999. Congress has endorsed that we work together with international partners on this project. As this will be the world's most advanced driving simulator, I invite to you to take advantage of this important research tool.

In closing, I want to emphasize that in the spirit of international cooperation and harmonization, my agency is committed to working with you, to communicating with you and other interested parties, and to making our resources available so we can provide the motoring public of the 21st century with the safest vehicles possible and the best consumer information. You have made tremendous strides in motor vehicle safety, and I believe you can be the first to make significant advancements towards world-wide harmonization.

Thank you.

Awards Presentations

U.S. Government Awards for Safety Engineering Excellence

Head of U.S. Delegation: Ricardo Martinez, M.D.

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

Australia

Gradimir Zivkovic
Milford Industries

Mr. Zivkovic is being recognized for his outstanding contributions to road safety through innovative engineering research. His contributions addressing safety issues include the development of user-friendly cargo barriers and load guards for station wagons, hatchbacks, and vans. His products are in the forefrcnt of load restraining systems and are marketed internationally. He has also applied innovative techniques to impact testing procedures. Mr. Zivkovic's work serves as encouragement for young engineers to pursue careers in research. For these contributions to international automotive safety, he is being recognized by this award.

Canada

Dainius J. Dalmotas
Transport Canada

Mr. Dalmotas has focused his research in the areas of human biomechanics and vehicle crashworthiness. Mr. Dalmotas' major accomplishments include the implementation of a field accident study documenting the injury experience of occupants restrained by three-point seat belt systems. This work included a joint National Highway Traffic Safety Administration (NHTSA)/Institut National de Recherche sur les Transports et leur Sécurité (INRETS)/Transport Canada study documenting the response of the human thorax when loaded by a shoulder belt, and a joint NHTSA/Transport Canada comparative dummy response study of the US SID, the EuroSID I and the BioSID. Additional work includes the development of the Canadian Belt Fit Test Device (BTD), injury criteria for CMVSS 208 use to regulate the safety performance of occupant restraint systems, and of padding strategies for side impact protection. For his outstanding and sustained contributions to the field of occupant crash protection and of restraint countermeasures, Mr. Dalmotas is deserving of special recognition.

France

Christian Thomas
PSA Peugeot/Citröen/Renault

Mr. Thomas has worked in the field of automotive safety engineering for the past 25 years. He conducted pioneering detailed crash investigations, focusing primarily on occupant protection. He leads the French motor vehicle manufacturers in the field of real-world field accident analysis. Mr. Thomas has made significant contributions in the areas of frontal and side impact protection, effectiveness of occupant restraints, and injury distributions and divergences between experimental data and real-world crash approaches to frontal regulatory tests, car ratings, compatibility in car-to-car collisions, and pedestrian and bus collision patterns. For his many accomplishments in the field of vehicle safety, Mr. Thomas deserves this special recognition.

Germany

Dipl.-Ing Eberhard Faerber
BASt

Mr. Faerber's accomplishments in the field of passive vehicle safety and biomechanics deserve recognition. He has served as advisor of the German Department of Transport in various fields of passive safety especially concerning full scale and component testing, accident analysis, and legal requirements for introduction of seat belts, airbags, and motorcycle helmet wearing. He has made substantial contributions for the establishment of European standards relating to vehicle occupant protection in frontal and side impact, as well as for the draft regulation for pedestrian protection. He has performed a wide range of crash tests and accident simulations for research purposes, consumer information, and the automobile industry. Mr. Faerber has published extensively in the field of passive safety and biomechanics, and is to be commended for his contributions in motor vehicle safety.

Dipl.-Ing Ingo Kallina
Mercedes-Benz AG

Mr. Kallina is recognized worldwide as a leader in bringing together accident investigation, intensive computer analysis and simulation, and crash testing for enhancing passive safety and promoting lightweight designs for conservation of environmental resources. Mr. Kallina's long standing commitment to passive safety and the surrounding environment have succeeded in the introduction of innovative and effective structural designs. Mr. Kallina played a key role in structural improvements to avoid intrusion and lower extremity injuries. He is currently promoting the optimized lightweight design approach to balance economy, ecology, and function of the body structure. In recognition of his many accomplishments in the field of structural and passive safety, Mr. Kallina is deserving of special recognition.

Dipl.-Ing Dietmar Otte
Medical University Hannover

Mr. Otte is being recognized for his development of an interdisciplinary in-depth accident investigation team on statistical basis, his exposition of concepts and simulations for further measurements in car design, safety devices for optimal protection of occupants, pedestrians, users of two-wheelers, and the development and optimization of shock absorbing elements, so called "Protectors" in motorcycle clothes. He is an assistant to national and international standardization committees, as well as a member of both the steering committee of International Research Council in Biokinetics of Impact and the Working Group of European Transport Safety Council in Brussels. Mr. Otte deserves special recognition for his contributions to Motor Vehicle Safety.

Hungary

Sándor Szabó
Automotive Industry, Ltd./Autōkut

Mr. Szabó is being recognized for his significant contribution to the development of laboratory testing methods and facilities concerning the active and passive safety of buses, especially in the field of pneumatic wheel-suspension, hydraulic assisted steering systems, and pneumatic braking systems. For the past 25 years, he has been an active member in the Working Party of Construction of Vehicles, UN-ECE-WP29, which is responsible for well over 80 internationally recognized vehicle safety regulations. Mr. Szabó deserves special recognition for his contributions to motor vehicle safety.

Japan

Dr. Kaneo Hiramatsu
Japan Automobile Research Institute

Over the past 20 years, Dr. Hiramatsu has worked in the field of motor vehicle safety. He has organized and conducted research on many technical issues of motor vehicle crashes. He has contributed to the activities of International Standards. His proposal on vehicle sensitivity to lateral wind disturbance has been adopted as an International Standard. Dr. Hiramatsu's current efforts include promoting human factors research, including advanced safety technology assessment using driving simulators. His efforts are also being directed towards reducing the number and severity of crashes involving older drivers. For these and many other outstanding accomplishments in motor vehicle safety, Dr. Hiramatsu is most deserving of this award.

Saburo Kobayashi
Honda R & D North America

Mr. Kobayashi has made significant contributions in the fields of technological concept creation and the Honda Experimental Safety Vehicle (ESV) Prototype fabrication. While advocating social responsibility, he has played a major role in promoting practical safety technologies and consolidating the base for current research of safety aspects. His research on airbag systems revealed the abnormal G-Wave observed at the chest during tests due to the structure of the dummy, and his research of effects on a standing-child, were highly acclaimed at the ESV Conference. He pioneered the research on the reliability of an airbag system, and has seen it through mass production with high reliability. The system was introduced on the Legend in 1987, the first car in Japan to be installed with an airbag. Under Mr. Kobayashi's leadership, passenger side airbags and seat belt pre-tensioners were installed. Mr. Kobayashi's many contributions in the field of international motor vehicle safety deserve this special recognition.

Kuniaki Osaka
Toyota Motor Corporation

With an expansive career in motor vehicle crashworthiness and crash avoidance research and development, Mr. Osaka is a leader in improving vehicle safety. The Toyota LS400 is an example of his efforts to produce a safe automobile, taking into consideration the various crash modes; full, offset, and oblique frontal impacts, side impacts, rear impacts, and rollover. Mr. Osaka served as Chief Engineer for Toyota's 1995 Advanced Safety Vehicle (ASV) program. This program encompassed both advanced safety and crash avoidance concepts. Mr. Osaka has contributed substantively to enhancing the safety of automobiles and deserves special recognition for his achievements.

Eiichi Yaguchi
Nissan Motor Company, Ltd.

Mr. Yaguchi has been a pioneer in vehicle safety advanced handling and stability research and development. As a result of his efforts, many valuable safety improvements have been achieved in stability and control. He was instrumental in advancing research on traction control systems and driver\vehicle interaction systems. His work led to the development of Nissan's Electronically Controlled torque split 4WD system, the Electronically Controlled Limited Slip Differential System, and the Anti-Slip Traction Control system. Mr. Yaguchi's current research activities include traction control, human factors, vehicle safety performance, and the use of driving simulators for older driver research. Mr. Yaguchi is most deserving of this special recognition.

Poland

Prof. Cezary Szczepaniak
Technical University of Lódz

Dr. Szczepaniak is being recognized for his contribution to automotive safety engineering. The focus of his research has been on traction and braking control systems and transmission fluids for vehicles. The Polish industry produces the Anti-locking Braking System for buses created by Dr. Szczepaniak. He is the recipient of the Collective Prize of the Ministry of Transport and three awards from the Scientific Prize of the Ministry of Higher Education of Poland for his contributions to the development of antilocking braking systems. Dr. Szczepaniak is being recognized for his outstanding performance in this field.

Sweden

Lars Gardell
SCANIA

Mr. Gardell has been the leader of development of the Scania 4 Series heavy trucks launched in October 1995. His outstanding leadership and dedication to motor vehicle safety have resulted in unique safety and handling properties, optimized brake systems including integrated hydraulic retarder, safety steel cab with self-adjusting seat-integrated safety belts, and built-in underride protection. For outstanding leadership of safety engineering in truck development, Mr. Gardell is deserving of this special recognition.

Björn Lundell
Volvo Car Corporation

Mr. Lundell began his career in motor vehicle safety in 1973. His work has covered a diversified field of safety, including research and development, testing, system analysis, and advanced engineering. During his tenure at Chalmers University of Technology, Mr. Lundell concentrated his efforts on head angular accelerations and pedestrian injuries. Since 1978, with the Volvo Corporation, his efforts have centered on occupant restraint systems, crashworthiness analysis of the Volvo 700 and 900 vehicles, and developing a safety strategy for future vehicles. Mr. Lundell and his work are well recognized in the international motor vehicle community. He is Chairperson for the International Standards Organization (ISO) Working Group on child restraint systems, and has undertaken the initiative to start an international Working Group to discuss standardization of passenger airbags, with an emphasis on disconnect features for airbags when using a rear-facing child restraint system in the front passenger seating position. For his valuable contributions to motor vehicle safety, Mr. Lundell is being recognized.

United States

Roger P. Daniel
Ford Motor Company

During 40 years of service to Ford Motor Company, Mr. Daniel has worked continuously in the field of occupant safety and biomechanics. He has participated in advanced safety research and testing, dummy design and development, and vehicle related biomechanical research and applied mechanics. His research has led to advancements in virtually every safety-related function of today's vehicle. He has made enormous contributions to the industry's knowledge base and technology in automotive safety. These accomplishments merit receiving this award for motor vehicle safety engineering excellence.

Dr. Harold J. Mertz
General Motors Corporation

Dr. Mertz is recognized for his leadership and contributions in the development of crash test dummies and injury assessment techniques for evaluation of restraint system performance. His expertise and steadfast efforts have guided the advancement of significant international dummy and human tolerance standards within the Society of Automotive Engineers and the International Standards Organization since 1970. Dr. Mertz has provided expertise and outstanding leadership in the field for more than 25 years. Well known internationally, his work has influenced human performance measurement within General Motors, throughout industry, government, and academia. For his lifetime achievements, Dr. Mertz is deserving of this special recognition.

Special Awards of Appreciation

In recognition of and appreciation for outstanding leadership and extraordinary contributions in the field of motor vehicle safety.

Australia

Harry L. Camkin
Advisor to National Road Trauma Advisory Council

Mr. Camkin has been a leading figure in road safety over the past 20 years. He has been influential in the introduction of Random Breath Testing in New South Wales (NSW) and the introduction of road safety into school and pre-school curricula. Mr. Camkin initiated both Road Safety 2000 (NSW Strategy) in 1990, and the national road safety equation which gave rise to the National Road Safety Strategy. He played a major role in determining the direction of the national model being used for strategic processes. His outstanding contributions and leadership for road safety at the national level merit this special recognition.

Prof. A. Peter Vulcan
Accident Research Centre

Dr. Vulcan has been actively involved in road safety initiatives for the past 25 years, both at a technical and policy level. Research undertaken by Dr. Vulcan and his team at Monash University Accident Research Centre for the Federal Office of Road Safety has played an important part in the introduction of improved occupant protection standards for the Australian motor vehicle fleet. He has been a leading proponent of the scientific approach to road safety, ranging from community-based safety projects, bicycle helmet use, and speed enforcement, and drunk driving crash investigations. He has published extensively in the fields of road safety with landmark evaluations of the effects of legislation, enforcement, and public education. Dr. Vulcan's energy and adeptness in bringing key research results to the attention of politicians, officials, and the community continue to make a significant contribution to road safety. Dr. Vulcan's outstanding contributions to improvement in road safety are deserving of this special award.

Italy

Dipl.-Ing Claudio Lomonaco
Ministry of Transport

Mr. Lomonaco is responsible for the Ministry of Transport's regulatory activities, carried out within the European Union and ECE/UNO, regarding motor vehicle safety. Throughout his career, he has served in numerous chairmanship roles of organizations such as the Group of Rapporteurs on Protective Devices, the Group of Rapporteurs on Crashworthiness, and the Group of Rapporteurs on Passive Safety. In 1992, Mr. Lomonaco was appointed by the U.E. Commission as a member of the High Level Advisors Group for automotive safety. Mr. Lomonaco's safety contributions include passive safety regulations that have been adopted in many ECE countries and in all of the European Union. For his service and dedication to the safety of the motoring public, Mr. Lomonaco deserves special recognition.

Sweden

Claes Tingvall
National Road Administration

Mr. Tingvall has provided valuable contributions in the field of passenger safety enhancement. He was one of the pioneers using paired comparison methods for vehicle safety ratings. Much of his work has been to document the positive effect of rearward facing child restraints. He is a leader in the development of motor vehicle child restraint systems, including protection in side impact collisions. He was one of the initiators of the International Standards Organizations (ISO) work to identify a remedy for motor vehicle child restraint system misuse. For the consumer perspective in vehicle safety research and for outstanding accomplishments in improving vehicle child restraint systems and vehicle safety ratings, Mr. Tingvall is being recognized by this award.

United States

Michael M. Finkelstein
Finkelstein and Associates

Mr. Finkelstein has had a long and distinguished career in furthering the cause of motor vehicle safety. He has demonstrated outstanding leadership in the field of automotive safety not only in the United States, but throughout the world. Mr. Finkelstein had a critical role in implementing automatic crash protection in the U.S. and in identifying motor vehicle crashes as a public health issue. He has championed programs to promote worldwide motor safety for the motoring public, by broadening, developing, and strengthening vehicle safety issues. He is recognized internationally as expert in the field, and has gained the respect of his colleagues, both domestic and foreign. He has been at the forefront on international issues and was instrumental in not only ensuring the continuance of the ESV program, but expanding its role while serving as NHTSA's Associate Administrator for Research and Development. In recognition of his numerous contributions, outstanding leadership, and dedication to motor vehicle safety, Mr. Finkelstein is especially recognized.

Section 2
Government Status Reports

Chairperson: Francis Turpin, United States of America

Commission of the European Community

Herbert Henssler
European Commission

ABSTRACT

This paper reviews the achievements of the European Community in relation to the EC Type-Approval legislation for motor vehicles and its safety-related requirements which occurred since the last ESV-Conference. The new Directives on the protection of car occupants in front and side impacts are presented as well as the intended legislation on the protection of pedestrians in the event of collision with cars. Furthermore, the paper outlines the current activities aiming at enhanced safety of buses and coaches. The paper also refers to the co-operation of the Commission with other international bodies in the field of automobile safety. The importance of the 1958 Agreement of the UN/ECE for international harmonisation of motor vehicle regulations is emphasized and the intention of the EC to adhere to this Agreement is explained.

INTRODUCTION

Mr. Chairman, I wish to thank you on behalf of the European Commission for the invitation to present this report of the regulatory activities of the European Community in the motor vehicle sector.

Many important developments have occurred in the EC since the last ESV Conference in Munich, exactly two years ago, and I am pleased to report on these here at this outstanding forum. The activities of the EC cover, of course, all important aspects of road safety: traffic regulations, driver related regulations and regulations relating to the vehicle construction. As the competence of the department I am representing is limited to the latter aspect, I will focus my presentation on the developments in the area of vehicle safety.

EC TYPE-APPROVAL

On 1 January 1996, EC type-approval became mandatory for all vehicles of the international category MI: that is passenger cars and similar vehicles such as motor caravans. In order to have access to the internal market of the EC, manufacturers of such vehicles must have an EC Whole Vehicle Type-Approval, which can be obtained in any one of the 15 Member States. Purely national approval schemes continue to exist for individual vehicles and vehicles produced in small series, but they do not automatically assure access to the markets of other Member States. (Work is on-going to extend the provisions of the ECTA to all other vehicle categories).

You will certainly be aware that the basic objective of ECTA has been to achieve the internal market of the EC in the vehicle sector, so that motor vehicles and their trailers, once they have been granted such an approval by the administration of one Member State, can be sold and registered without any technical or administrative obstacles in the whole EC. However, the provisions of the Treaty of the EC on which the ECTA and its technical requirements are based, require that a high level of safety, environment and consumer protection must be assured by such regulatory initiatives.

This stipulation and its mandatory nature make the ECTA the ideal basis to introduce, for the whole of the EC, stringent requirements relating to the construction of vehicles, vehicle systems and components which affect safety and environmental protection.

Consequently the Commission follows a process of technical evolution, adapting the existing requirements laid down in so-called Separate

Directives, of which there are 45 at present, to the state of the art.

Equally, the increasing public pressure for improved road safety and environmental protection are a motivation for the Commission to amend existing directives or to establish proposals for new directives.

In the following I will elaborate essentially on the progress made on regulations relating to the safety of vehicle occupants and pedestrians, issues where we have identified considerable need for improving the present situation.

PROTECTION OF OCCUPANTS IN FRONT AND SIDE IMPACTS

The continuous efforts to improve the existing EC legislation have led to a levelling -off of the number of casualties, in spite of a significant increase of the number of vehicles and the annual mileage driven on the roads of the EC.

However this stagnation occurs at a level which is far too high. Therefore, at the end of 1994, as announced at the last ESV Conference, the Commission submitted to the decision-makers of the EC two new Directives aiming at the protection of the occupants of cars and assimilated light motor vehicles in frontal and lateral collisions.

The proposed measures reflect the most recent state of the art, both as far as the representivity with respect to real-world accidents in the EC and the biomechanical aspects are concerned.

The proposal on front impact protection foresees a test of the vehicle to be approved against a fixed, off-set, deformable barrier at 56 km/h. The biomechanical criteria for assessing the protective characteristics of the vehicle relate to the head, neck, thorax, femur and lower leg.

The proposal on lateral impact protection relies on a test with a mobile deformable barrier against the stationary vehicle at 50 km/h. The biomechanical protection criteria relate to the head, thorax and pelvis.

Both proposals are intended to become mandatory, for new types of cars, from 1 October 1998, and for all new cars entering the market and registered in the EC from 1 October 2003 onwards.

Both proposals rely to a large extent on the work of other European, organisations like EEVC and the UN/ECE's GRSP; I will elaborate this aspect somewhat later. It is also worthwhile to note that the important public interest in the EC on road safety matters has resulted in numerous suggestions for improvements when these proposals had been published. In particular the European Parliament has taken an important role in this respect.

Today, these proposals are in the final stage of the decision-making process of the EC. Understandably, priority has been given to the directive relating to lateral impact, because in this field no regulations exist in the EC. I expect this Directive to be finally adopted within a few weeks. The Directive on frontal impact took somewhat more time because of the determination of the Member States to include in the present stage criteria for lower leg injuries, for which an adequate test tool had to be developed. I expect, however, this Directive also will be adopted before the end of the year.

PROTECTION OF PEDESTRIANS AND OTHER UNPROTECTED ROAD USERS

A comprehensive, performance-related regulation, aiming to ensure that car fronts are as "pedestrian friendly" as possible, figures on this year's work programme of the Commission. Here again the EEVC has done the basic work on which this future EC directive will rely. The technical specifications of the tests and the necessary test tools are nearly complete so that a proposal should be possible before the end of the year. It appears, however, that the compliance with such legislation will necessitate significant changes in the construction of cars and that only new car concepts will be able to fully meet the specifications. This will have to be reflected in the implementation of the intended directive, which will have to "phase-in" gradually.

Recognising this fact, the Commission is presently preparing appropriate amendments to the existing directive on external projections in order to allow Member States to ban dangerous devices fitted to car fronts as soon as possible.

BUS AND COACH SAFETY

Public concern raised because of a number of serious accidents over the last years has caused the Commission to consider the safety of buses and coaches as a priority area for its regulatory programme. The current activities cover essentially the two following aspects. A third area, the fire risk of buses and coaches is now covered by EC legislation through the adoption in 1995 of a Directive relating to the flammability of materials used in the interior construction of these vehicles.

BUS AND COACH CONSTRUCTION STANDARDS

A proposal for a new directive of the European Parliament and Council is being prepared, following extensive consultations with the Member States and industry. Once adopted, it will allow the approval of buses and coaches as complete vehicles and of bodywork as separate technical units. As regards safety, it will stipulate the number of exits, for use in emergency evacuation, and include requirements for rollover strength and stability. It is hoped that the final proposal will be submitted to the Parliament and Council in the next few months and to enter into force as soon as possible thereafter.

BUS AND COACH SEAT BELTS AND STRENGTH OF SEATS

On a shorter timescale, the Commission intends to adopt amendments to three existing directives relating to seat belts, seat belt anchorages and the strength of seats in order to introduce the mandatory fitment of seat belts, and in some cases energy absorbing seats, in all seating positions of medium and large passenger vehicles. The proposal specifies the fitting of 3-point belts in minibuses, where the risk of injury in frontal impacts is high, and 2-point belts and energy-absorbing seats in large coaches, where the main risk of injury relates to ejection during roll-over.

AIRBAG WARNING LABEL

The amending directive on seat belts, referred to above, also contains a proposal to require the fitting of a warning label to all passenger cars fitted with a passenger-side airbag. This is to warn against the use of a rearward-facing child restraint in these seats on account of the serious risk of injury to a child in such a seat if the airbag should deploy.

CO-OPERATION WITH OTHER INTERNATIONAL ORGANISATIONS

In the field of vehicle safety the European Commission has established excellent relations with two international organisations: The European Experimental Vehicle Committee (EEVC) and the UN/ECE Working Party 29 and its Group of Rapporteurs on Passive Safety (GRSP).

EEVC traditionally helps the Commission to establish the scientific basis for new safety regulations and acts as a coordinator for research projects which the Commission initiates in this respect, eg, in the past, the European side impact dummy EUROSID and, more recently, the validation of the new European front impact test procedure. In our view the EEVC will have an important role to play not only for future regulatory projects of the EC, but also in relation to the future efforts for worldwide coordinated safety research. We will certainly hear more about that in the presentation of its Chairman and in the following panel.

WP 29, acting in the framework of the 1958 Agreement of the UN/ECE, is at present the key international regulatory body in the area of vehicle safety. Many of the relevant ECE regulations have been carried over into the EC Type-Approval system and are considered to be equivalent in view of giving access to the internal market of the EC. It is, therefore, logic that, as the revision of October 1995 of the 1958 Agreement allows now for "regional economic organisations" to become Contracting Party to the Agreement, the EC intends to adhere as soon as possible in order to establish a legal link between the two organisations. The internal procedure for this purpose has been introduced by the Commission in December 1995, We hope that important motor vehicle producing countries in the world which are not currently Contracting Parties will also join the revised Agreement in the interest of world-wide harmonised regulations and free global trade unimpeded by technical obstacles. We welcome Japan's decision to join and hope this will be finalised soon.

We are also prepared to envisage additional amendments to the Agreement in order to transform WP 29 to a true global platform for establishing international regulations in the terms of the Technical Barriers to Trade agreement of the World Trade Organisation. However, these amendments should be limited to what is necessary to attract the wider membership which we desire whilst not jeopardising the achievements of the Agreement in its present form as far as the international quality of the regulations and the reciprocal recognition of approvals granted in accordance with these regulations are concerned.

The importance of moving towards the globalisation of international regulations was highlighted at the recently convened TABD automobile conference in Washington.

CONCLUSIONS

Subject to the conclusion of the work on frontal and lateral impact and pedestrian protection the EC Type-Approval procedure and its technical requirements are in principle completed for passenger cars. The next future will see the Commission to focus its activities on the extension of this procedure to heavy vehicles and their trailers. This will, in the area of vehicle safety, require a few new Directives and the amendment of quite a number of existing Directives. I hope I can report at the next ESV-Conference on substantial progress made in this respect.

It is my view, however, that the development of essential new regulations as well as the evolution of existing regulations affecting the design and construction of motor vehicles should, in the future, take place to the greatest extent possible on a world-wide basis and no longer nationally or regionally. Not only our industries would benefit from this principle but also their customers in the whole world who would be able to acquire motor vehicles offering a high standard of safety and environmental protection at reasonable cost. I am convinced that this conference is an excellent contribution to this objective.

I thank you for your attention and wish the Conference and its organisers a great success.

The European Experimental Vehicles Committee (EEVC)

Bernd Friedel
Bundesanstalt für Straßenwesen

It is my pleasure to present the Status Report of the European Experimental Vehicles Committee on the progress of our work since we met in Munich in 1994.

FRONTAL COLLISION

We have outlined our activities on the Front Impact Test Procedure on several occasions. Now these activities come to an end after the validation programme is finalized. This test programme was successful and the information gained from these tests was supplemented by tests performed by NHTSA, Transport Canada, the Australian Federal Office of Road Safety, JARI and the Insurance Institute for Highway Safety in the USA.

Recommendations regarding an offset test are now formulated:

- The vehicle should impact a deformable face of an aluminum honeycomb block with an overlap of 40 % with a speed of 56 km/h.
- Hybrid III fiftieth percentile dummies are used and we recommend the following performance parameters:
  - head: HIC_{36}, 80 g/3 ms acceleration exceedence;
  - neck: Duration of shear and tension force and extension bending moment, chest: compression and VC;
  - femur: Peak and duration of longitudinal force, tibia to femur translation at knee level; and
  - tibia: compressive force and Tibia Index steering column residual upward and rearward displacement.

Some of the tests reported have shown a deficiency in the design of the Hybrid III ankle. At full dorsiflexion articulation, the stop on the joint is a metal to metal contact causing a high spike. There is now a task force of EEVC to resolve these problems and also to formulate a certification procedure for the foot and ankle.

Our recommendations have been used as the basis for a draft ECE regulation as well as by the Commission of the European Union with the result that the deformable test will be introduced as an EC directive. A common position of Commission and Council will be achieved probably in springtime 1996. A second reading of the European Parliament and the final decision of the Council are expected in autumn 1996.

All this work was done over many years by working group (No. 11) of EEVC under the chairmanship of Mr. Lowne, who will present the results in detail during this conference.

SIDE COLLISION

As we have pointed out in Munich the EEVC is dealing with the development of a repeatable and meaningful head impact test for the evaluation of occupant head impact protection in side impacts. The test programme is split in several phases: in phase 1 baseline tests are conducted and the affect of angles of impact is studied, in phase 2 the problem of linear versus free flight impactors will be analysed and in phase 3 a range of popular modern cars will be evaluated with the preferred test method. The end of phase 1 of this programme is expected during 1996.

Furthermore progress is made in the development of a test programme that will check the ability of mobile deformable barrier faces to work sensibly and repeatably. This will be so when impacting cars as well as meeting the force-time performance specification against a rigid, flat load cell wall. A third task is the development of a specification for the positioning of the EUROSID dummy.

These matters of protection in side collision are dealt in EEVC Working Group 13 chaired by R. Lowne. A Common Position of the Commission and the Council for a new directive was achieved in November 1995, a second reading by the European Parliament has been achieved in March 1996 and the final decision of the Council is expected in summer 1996. There will be a paper describing the progress of the head impact test programme during the conference.

FRONTAL IMPACT DUMMIES

Working Group 12 was created with respect to the development of frontal impact dummies. This working group is chaired by Dr. Wismans. An EEVC Report "Recommended Requirements for Frontal Impact Dummies" will be available soon.

The set of recommended requirements in this report should be regarded as additions and comments to the NHTSA development program, taking into account European conditions in order to ensure that the design resulting from the AATD development program can be used also outside the USA.

The working group is continuing the contact with NHTSA on the subject of the AATD development programme. EEVC is prepared to evaluate parts of this new dummy but also any other advanced frontal dummy design. Furthermore the question related to facial injuries in frontal accidents for the drivers and the protection of such in injuries is considered carefully by Working Group 12.

PEDESTRIAN PROTECTION

Concerning Pedestrian Protection we have described our activities at the last conference. Also at this conference a written paper of EEVC Working Group 10 is available. The evaluation tests with the impactors are partially finished. The head impactor and the upper legform impactor are ready for use, the legform impactor was recently improved concerning the damping of shear motion simulation in the knee. The European Commission has started the discussion of this item.

FRONT UNDERRUN PROTECTION OF HEAVY GOODS VEHICLES

Working Group 14 of EEVC has worked to determine of the benefit of an energy-absorbing front underrun protection system for trucks compared with rigid devices. The activities of the Working Group have been sponsored by the European Commission. In a first step a representative type of accident was defined. The material did not include the effect of a rigid front underrun device according to ECE Reg. 93. This effect has been studied in the second step in which three tests have been carried out with a rigid device, a velocity of 56 km/h and an overlap of the car of 75 %. With the help of mathematical simulation the sensitivity of the parameters which define the representative accident type like angle, speed, overlap will be investigated and the requirements for an energy absorbing system will be defined which guarantee a predefined injury reduction to car occupants. With this data the previous cost-benefit analysis will be updated. During this conference the activities and results will be reported by the Working Group.

COMPATIBILITY OF CARS

EEVC has created at the end of 1995 a new working group (No. 15) on compatibility. This important problem will be studied in stages during the next few years. Based on a general overview, the relevant issues will be identified in terms of injury reduction and appropriate measures. In this overview lorries as well as vehicle categories Ml and N1 will be considered. In a second stage the significant parameters with respect to the evaluation of compatibility between vehicles of different types within categories M1 and N1 will be evaluated. This analysis is scheduled for three years. After that the question of the development of a relevant test procedure will be discussed.

We are keen to report to you the results of our studies in this difficult area over the next few years.

INTERNATIONAL COOPERATION

The work of EEVC is based on international cooperation. Not only the governments involved in our activities and their experts are mentioned here, we have a very good liaison with the European Commission, which sponsors much of our research activity. For the new 4th Framework Research and Development Programme we submitted our interest in performing studies on biomechanics and compatibility as already mentioned. A final decision is still pending.

EEVC maintain strong relations also with ECE in Geneva. Much of our scientific knowledge in different fields is transferred to their regulatory work. International cooperation has taken place over many years with governments of non European countries. As mentioned in earlier status reports we are working together with the governments of Australia, Japan, Canada and USA. Their advice and technological experience is a very valuable contribution for our work. We are willing to continue this successful way.

OUTLOOK

During this conference we will distribute a new brochure about the activities of the European Experimental Vehicles Committee. In this you find a description of the major tasks we have investigated and a complete list of references. After working now for more than 20 years we are proud of the results we have achieved. This success has many fathers, I just want to mention the expertise of industry, the research field and governments. Due to the success in the past we are convinced that we should continue in the same way to encourage further improvements of the safety of cars. EEVC has the ambition to remain as a strong partner to the European Union in its efforts to improve the vehicle safety by conducting scientific research to address identified safety problems.

United Kingdom

Keith Rodgers
Department of Transport

INTRODUCTION

Thank you. Good afternoon ladies and gentlemen. Our Chief Mechanical Engineer, Mr. Fendick, sends his apologies at not being able to be with us today. I have been asked to deliver his address in his absence. It is therefore my pleasure to present the status report for the United Kingdom as well as explaining some of the trends in national thinking on road vehicle safety.

UK CASUALTY REDUCTION TARGET

The driving force for our deliberations continues to be the high rates of personal injury accidents. We all face the need to reduce the cost to individuals and the State of the carnage on our roads. In the UK we have been successful in reducing fatal and serious injury accidents. Ten years ago the UK set a national target of a one third reduction in road accident casualties by the year 2000; the base line being the average level in 1981 to 1985. In this respect deaths and serious injuries are already below the target despite a 43 per cent increase in traffic. The downside to this, however, is that all casualties are only 2 percent lower than the target - these being dominated by slight injuries.

I believe that better vehicle design combined with changes in public attitudes, and a combination of local and central government actions, have helped. Thought is now being given to a new target post 2000 and the need to address the severity of accidents so that real improvements are not hidden by the overall increase in traffic. But it is important to keep up this momentum and I will return to this later.

STANDARDS SETTING

Of course the main weapon Governments have to influence matters is in the area of standard setting. There are many players in the European Union involved in this; national Governments, the Council of Ministers, the European Commission, European Parliament, research organisations, manufacturers, consumer organisations and not least, the public. In these international negotiations, that largely concern the single market, the UK places a high emphasis on safety.

I am encouraged by developments since the last ESV. In particular, I welcome decisions in the European Union to move straight to effective requirements in the Directives which will cover side and frontal impact. These have been based on the excellent work done by the EEVC. I hope that there will be a similar approach to the Directive on pedestrian safety. The continued role of the EEVC as a focus for the research into aspects of standard setting is vital. It must continue its role of providing strong unbiased scientific and technical input and advice that can form an objective basis for, sometimes difficult, discussions which follow in Europe.

ACCIDENT DATA COLLECTION

The basis for work on improved road and vehicle safety is the collection of data from the accidents on our roads. We continue to collect accident data on all injury accidents on a nation-wide basis, using police reports, which by law must be completed for each injury accident. We also have access to coroners' reports that are compiled on fatal accidents; these in association with police reports are a unique source of information.

To underpin our secondary safety research we also conduct a programme of in-depth accident investigation work. This is carried out on about 1500 crashed vehicles per year, more than 40% of which are involved in fatal or serious accidents. This project is co-sponsored by Ford, Toyota, Rover, Nissan and Honda. It provides these car makers with real, statistically sampled accident and injury causation data, and enables them to feed these data into their design processes. The Department uses these data to monitor the effectiveness of new systems developed by industry and to develop research proposals aimed at further mitigation of injuries. Papers that utilised these data are to be presented to this conference.

I should also like to announce that a symposium on real world crash injury research will be held next May in the UK. It will give Governments, policy makers and safety engineers the opportunity to learn from those researching real world crash injury causation.

CURRENT ATTITUDES

Turning now to current attitudes. The public recognise the importance of vehicle safety and place a high priority in safety issues. This interest has a very direct effect on future regulations and what vehicles are produced. It is important that this interest is fed and strengthened. I welcome this development and the challenges it brings in conducting a well -informed debate, conducted in an open and balanced way. The UK Government would like to see this trend strengthened by access to reliable and objective safety information. Public awareness has also been recognised by car makers who, in recent years, have positively marketed safety in their products.

INFORMING THE CONSUMER

However type approval standards, by their very nature, are only set at base level. It is very important that there are incentives for manufacturers to strive for higher levels of safety. It is therefore our view that user information should supplement existing regulations wherever this can be made available. The consumer needs to have the information necessary to influence his purchase decisions. These are healthy pressures but it is also important that the information is made available in a clear and meaningful way.

To provide some of this information the UK is carrying out research into the feasibility of a UK NCAP scheme. We are conscious of the role and interest in NCAPs internationally and especially their use in the USA, here in Australia and more recently in Japan.

In the current phase of this programme, we are evaluating several safety aspects of cars in the super-mini class. This is to see if the results offer a reliable indicator of relative performance and how this data can best be presented. The test procedures are those developed under the EEVC for the relevant draft EU directives. The method uses tests for frontal, side and pedestrian protection which is a wider range than for existing NCAP programmes. However the front impact speed used is 64 km/h, which is higher than that adopted for the proposed Directive but in line with that used by the Insurance Institute for Highway Safety and Australia. This is fully justified as it is important to test at a level above the minimum regulatory requirement. It also broadens experience of the test procedures provide useful data for the intended review of the frontal impact Directive set for two years after implementation.

However the worry of manufacturers over the consumer being misled or confused are recognised. We are concerned that results issued are both fair to the consumer and the manufacturer. To help in this the manufacturer has full access to the UK tests on their own vehicles. In the development of this assessment procedure the UK is hoping to work with other Governments and user organisations, either in close collaboration or in a looser liaison on purely technical aspects.

RESEARCH

In addition to the NCAP project the UK is also funding an extremely varied range of research. From biomechanics to bull bars (or in local terms roo bars - one of the less welcome Australian Imports to the UK compared to the welcome alcoholic liquid imports). An important area of research which more emphasis is now being placed is that of vulnerable road users.

Much time and effort have gone into improving the fronts of cars in regard to the damage they do when colliding with vulnerable road users. All this benefit can be wasted by the fitting of rigid bullbars. From laboratory testing clearly such devices are extremely aggressive and can greatly increase the severity of injuries. However in moving from the laboratory to the road it is not so easy to prove that bull bars are actually causing deaths and injuries. Not only is it necessary to identify those accidents that involved bull bars but also to prove that the accident severity had been increased. Currently we are working towards that end, but in the meantime it is becoming politically imperative that something be done about bull bars. This is likely to involve the EU in amending the external projections directive.

Towards the end of 1993 there were a succession of serious coach and mini-bus accidents in the UK. Public pressure for the introduction of seat belts in these vehicles became irresistible; this being especially the case for the carriage of school children. We have introduced in the UK new requirements for coach and mini-bus seat belts to be fitted for the protection of children, which will effectively increase the number of seat-belted vehicles available for general use. Whilst this will undoubtedly reduce casualties, there remains a need to carry out further research on seat belts in large passenger vehicles. Currently, European policy tends to favour lap belts in conjunction with energy absorbing seat backs. I understand the Australian approach favours three point belts which offer improved safety for the wearer but, because of the necessity for stronger seats, offer less protection for any unrestrained passenger behind the seat. We have been undertaking research to examine how coach seats can be designed to offer protection in situations where combinations of lap belted, three point belted and unbelted passengers are being carried.

Another area where we are directing effort is in the rapidly developing field of telematics. All of us benefit greatly from the rapid progress of electronics that provide cost-effective improvements, often aimed directly at greater vehicle safety (and reductions in emissions). Further enhancements and new features mean that systems, including safety related systems such as collision avoidance, will continue to develop in complexity.

In the future we see increasing interest in setting standards to protect a wider range of road users. This includes the older drivers and on systems that allow for a range of sizes and shapes of occupants.

All this work needs to be directed to meet strategic objectives which brings me back to the need for road safety targets.

FUTURE SAFETY TARGETS

The next UK road safety target is currently the subject of debate within the road safety community. Improving vehicle design will continue to offer a significant contribution but achieving this will require the continued development and refinement of existing and new measures.

It is too early to speculate on the size of the next target. But, looking further into the 21st century, it would be wistful, but I hope not impossible, to imagine a scenario where the present levels of fatal and serious casualty levels could be reduced well below current levels. Perhaps we could envisage a return to the level in the late 19th century in the horse drawn era. (In the 1890s there were 1663 fatalities in the UK. The vast majority of which were linked to the horse based transport system, 11 to bicycles or velocipedes and only 7 to "motor vehicles" that is traction engine/steam roller.) It would be a significant achievement to have the mobility of the 21st century but the casualty level of the 19th.

One thing though is clear - the achievement of further reductions in casualty levels will demand an increasing level of sophistication in our strategic view of vehicle safety measures and their implementation. Research will play a key role in this.

Specialists involved with safety whether in industry or research groups continue to raise the level of expertise and it is particularly encouraging to see the degree to which this safety attitude and expertise has become embedded in the culture of the both research and design of vehicles. The success and growth of ESV has been a key element in improving vehicle safety.

It is a pleasure to be able to meet and hear all the experts at this conference. I hope that it will once again renew our efforts for further improvements in vehicle safety.

Germany

K.-H. Lenz
Bundesanstalt für Straßenwesen

It is an honour for me to be able to present the German government's status report for this year's ESV conference.

HIGHWAY CONSTRUCTION AND TRAFFIC ENGINEERING

Approximately 52% of the total kilometrage of motor vehicle travel is covered on federal interstate highways and autobahns. The concentration of motor vehicle traffic on the federal autobahns is particularly high. Although these make up only approximately 1.8% of the length of the entire highway network, they account for around 31% of the total kilometers travelled. These high proportions - and the proportion of heavy vehicle traffic on the autobahns of over 15% is particularly worthy of mention - underline the significance of the federal interstate highways and autobahns, and of their state of maintenance and improvement, for safe handling of traffic.

Because of the rapid increase in traffic on the federal autobahns, which is expected to continue in the future, it is essential that the highways be widened to six or eight lanes and simultaneously renewed and adapted to the current technical standard, in order to increase capacity, and above all to reduce accidents.

The five-year plan for the period 1993 to 1997, with supplements up to the year 2000, provides for extension of the parts of the network having six or more lanes by around 940 km. According to the planning of requirements for the federal interstate highways and autobahns, improvement of a further 1,680 km is planned as a "priority requirement". When these measures have been completed, around 30% of the network will have six or more lanes.

As described at the 14th ESV conference, further substantial investments aimed primarily at improving road safety (reconstruction or pavements, construction of hard shoulders, installation of crash barriers, repair of structures etc.) have been made in particular on the autobahns of the new federal states.

Finally, in order to increase road safety on the federal highways, the danger points (e.g. curves and crests having too small a radius, junctions and intersections with poor sight distance) are being modified one by one, and overloaded by-passes are being eliminated.

In 1994 for example, 28 by-passes (total length 175.5 km), including the sections completed in previous years, were opened to traffic. In total, in 1994 alone, 1,076.6 million DM was spent on construction of by-passes.

The elimination of at-grade railway crossings of the Deutsche Bahn AG (German railways) on federal highways is also aimed primarily at improving road safety. In 1994, around 54 million DM were made available from the resources of the road construction plan for this and other technical safety measures.

Finally, the construction of cycle paths along federal highways also contributes considerably to improving road safety. Thus, in 1994 alone, around 500 km of cycle paths were completed along federal highways.

Use of traffic-actuated traffic control systems is being intensified in order to increase road safety and improve the traffic flow.

In 1994 and 1995, the Federal Minister of Transport allocated a total of around 175 million DM to the federal states for installation of such systems.

The investment program for traffic management on federal autobahns envisages a need for funds amounting to 650 million DM for the period from 1993 to 1997.

At present, around 70 traffic management systems are in operation on the federal autobahns.

Further traffic management systems are to be installed until the end of 1997. The length of the sections equipped with variable speed limit and congestion and fog warning systems will then have been increased by approximately 400 km from over 400 km at present, and the length of the sections controlled by variable message signs by approximately 1,000 km from approximately 1,200 km.

With variable message signs, which were also used in 1994/95, it was possible to divert 20 to 40% of the through traffic (depending on the length of the detour) from overloaded federal autobahn sections via alternative routes of adequate capacity and thus to reduce the risk of congestion and accidents.

On the federal autobahns on which accidents occur particularly frequently, variable message signs were used, when necessary, to warn the drivers on accident risks due to the current traffic and weather conditions (e.g. in the case of congestion and fog). At the same time, speed limits were set or lanes closed.

By this means, it was possible to reduce the number of accidents on these sections by 20 to 30%, and the number of accidents involving injury to persons by up to 50%.

Purposeful, appropriate traffic management is accepted, with the consequence that accident black spots can be made considerably less hazardous and that the traffic flow can be kept smoother at critical points.

Accident black spots on federal highways were made less hazardous by means of highway construction and traffic engineering measures. At the same time, more and more situation-dependent traffic control measures are being provided. In 1993, because of the increasing significance of traffic management systems for improvement of road safety and of the traffic flow on federal highways, the Federal Ministry of Transport established a program for traffic management on federal highways.

The measures are intended to help reduce the negative consequences of the continuously increasing traffic volume on the federal highway network, which is subordinate to the federal autobahns.

The program comprises measures at an overall cost of 125 million DM, of which 100 million DM will be incurred in the period from 1993 to 1997. 60 large facilities and 200 smaller ones are to be installed.

In order to protect vehicle occupants, crash barriers have been developed in past years which, due to their yielding action in the event of an impact, can also deflect heavy vehicles, and, due to the lower deflecting forces, lead to less severe injuries. Since 1960, most of the federal interstate highways in the former federal states have been equipped with crash barriers.

Equipping the autobahns with median barriers, an essential safety measure, was also dealt with as a priority task in the new federal states, and completed at the end of 1992. The cost of equipping the approximately 1,800 km was around 90 million DM. In order to improve road safety, a further 1,038 km of crash barriers were additionally erected on the outer edges of the carriageways in these states between the beginning of 1990 and the end of 1994.

In order to protect motorcyclists and moped-riders, at points where accidents occur particularly frequently, the posts of the crash barriers are being provided with specially developed impact-reducing casings.

HIGHWAY TRAFFIC REGULATIONS

When the 13th decree for amendment of the highway traffic regulations (Straßenverkehrsordnung) became effective on 1 August 1995, an improvement in safety at "dangerous/critical" bus stops of school buses and regular service buses became current legislation.

According to the new regulation, the local highway traffic authorities determine which bus stops shall be classified as "dangerous/critical", and instruct the bus drivers to switch on the hazard warning flashers when the bus approaches such a bus stop. The bus must not be overtaken when the hazard warning flashers are switched on. If it stops, it may be passed only at walking speed; this applies both in town and out of town and also for opposing traffic.

Due to the increase in heavy vehicle traffic, there are not sufficient parking facilities in the rest areas on federal autobahns to permit truck drivers to observe the stipulated rest periods. This applies in particular in the case of longer breaks. The signs indicating rest areas for truck drivers take this into account.

In order to improve the safety of bus traffic, all persons applying for a driving licence providing entitlement to carry passengers in buses must receive theoretical and practical instruction in a driving school as from 1 October 1993. As from I October 1993, the practical test must furthermore also include roads outside built-up areas because accidents involving buses generally occur in non-urban areas.

Investigations have shown that the number of drivers who use the road when under the influence of drugs, also in combination with alcohol is growing. Even if, according to these investigations, use of illicit drugs is less wide-spread than consumption of alcohol among road users, there is still a problem which gives major cause for concern about safety on the road.

According to the provisions of sections 315c and 316 of the German criminal code (Strafgesetzbuch), anyone who drives a vehicle although, as a consequence of consumption of alcoholic beverages or other "intoxicating substances", he is not capable of driving the vehicle safely shall be punished. In contrast to the situation in respect of alcohol, no limiting values have so far been set for drugs at which absolute unfitness to drive is assumed. A person may be convicted for driving under the influence of drugs only if his unfitness to drive can be ascertained and proved. Difficulties are often encountered when relative unfitness to drive is to be ascertained. So far, there is no objectively measurable criterion, analogous to the 0.8 parts per thousand limit, for determining whether an offence has been committed, which applies independently of ascertainment of unfitness to drive. The Federal Government wishes to close this loophole with a draft bill which was delivered to the legislative bodies in autumn 1995. According to this bill, driving a vehicle whilst under the influence of certain intoxicating substances will in future constitute an offence and will be combated with a fine and a driving ban.

The second EU directive on driving licences of July 1991 is to become effective over the whole of the EU on 1 July 1996. The implementation of this directive will mean not only that the previous national driving licence vehicle category system will be brought into line with the international system, but, in addition, that the points system will be set upon a statutory footing and the individual measures will be modified.

Furthermore, the adoption of international regulations on construction and equipment of road vehicles into national law is making a substantial contribution to increasing road safety. In total, over 95 uniform regulations on motor vehicles and their trailers have been passed by the UN Economic Commission for Europe (ECE). Over 75 of these regulations can be applied in Germany. With the cooperation of the Federal Republic of Germany, two ECE draft regulations were passed in respect of frontal and side impact of cars. Both regulations can be applied as of 1995.

The harmonisation of the technical regulations of the member states of the European Union concerning cars is largely completed. In spring 1995, the EU commission put two draft directives concerning frontal and side impact before the Economic and Social Council, the Council and the European Parliament for further consideration. These directives which contain stricter specifications than the ECE regulations (offset barrier, barrier height 300 mm) will be officially passed this year and come into force as from October 1, in 1998 for newly registered passenger cars.

Together with other Federal Ministries, a concept for disposal of old cars is at presently being prepared. This includes disposal of old cars, which are no longer registered and left parked in public street space.

Since 1991, training programs for transport of hazardous materials have been provided in the Federal Republic of Germany. The EU will introduce a comparable scheme with the directive on appointment and professional qualifications of safety advisors on transport of hazardous materials on the road, by rail, and by water. In view of the number of member states which do not have such a scheme, it has been stipulated that the scheme shall become effective on 1 January 2000.

On 1 January 1997, the Council directive on uniform procedures for checks of vehicles carrying hazardous materials in road traffic will become national law in all member states. Then, in all member states, a representative proportion of the vehicles carrying hazardous materials on the road will have to be checked in accordance with uniform inspection criteria. The road checks will be effectively assisted by the possibility of checks in the companies, measures in the case of offences, and improvement of the administrative aid between the member states.

ACCIDENT STATISTICS

The accident trends in the area which constituted the Federal Republic prior to 3 October 1990 and in the area of the five new federal states are not developing uniformly. The two areas will therefore be treated separately in the description below.

In the former area of the Federal Republic, the number of road accidents decreased by around 1.7 million or 12% between 1992 and 1995. In the same period, the total kilometrage travelled decreased by 0.4%.

The number of people killed in road accidents has decreased by almost two thirds since 1970. The number of cars has increased from 17 to 39.5 million in this time, and the total kilometrage travelled has increased by 101%.

In 1992, there were around 7,300 fatalities in road accidents (excluding the new federal states and East Berlin), in 1993 around 6,900, in 1994 6,800. The lowest number since 1953 was achieved in 1995, with around 6,650 fatalities. In 1992, 4,250 car occupants were killed.

The accident statistics in the five new federal states subsequent to the German unification in October 1990 are particularly noteworthy. In the said area, around 3,330 fatalities were registered in 1992, around 3,020 in 1993, and around 3,010 in 1994. In 1995, the number of fatalities fell to around 3,000.

ACCIDENT RESEARCH

As at the previous conferences, the activities of the Federal Government, the automobile industry and the motor insurance companies should be mentioned here.

The Federal Ministry for Education, Science, Research and Technology has for many years been promoting research and development projects aimed at improving active and passive safety in road traffic. Transport of hazardous materials on the road is a research area which has been particularly well promoted in recent years.

At one of the previous ESV conferences, a report was given on the development of the TOPAS safety tanker trailer and on the aims and content of the subsequent THESEUS project (development of tankers with highest achievable level of safety by means of experimental accident simulation), which has now been successfully completed after four years and received almost 10 million DM in financial support.

Important German research institutions and partners in industry have carried out systematic, reproducible investigations relating to the active and passive safety of the tankers, their components and their safety devices.

The starting point of the investigations was the detailed analysis of 231 accidents involving tanker trailers. The safety level was realistically analysed and evaluated in 36 crash tests and 12 overturning tests with various tank designs. In addition, computational models describing both the material strain of the tanks used today in the Federal Republic of Germany, and the handling of the various tanker types, were developed and checked in driving dynamics simulations and accident simulations. Analogies to static overturning tests, which showed the overturning limit to be at an angle of approximately 25°, were also derived. By this means, the overturning limit of every tanker can be determined relatively easily on the tilting bridge.

The following measures ought to be able to contribute to a significant improvement in the safety of transport of hazardous materials, even with the cost benefit ratio also being taken into account:

- Further lowering of the centre of gravity of the vehicle in order to increase the overturning limit of the vehicle (already realised in the TOPAS research project).
- Development of overturning warning devices which, on the basis of the steering angle, the driving speed, the loading condition etc., can determine when there is a danger of overturning, and will make the driver aware of the danger.
- Optimised dome covers and larger-dimensioned fastening elements which will resist a strain of 30 g and 4 bar, as determined in the project, for a period of 50 ms.
- Optimised piping and fittings which provide a higher level of tightness of the valves.

On the basis of cost-benefit considerations, headway warning devices were also recommended which, taking account of all information concerning the vehicle and its surroundings, are intended to assist the driver in maintaining the necessary safe headway from vehicles ahead.

The data compiled are so comprehensive that it is not possible to give an account of details here. The data are, however, available to interested parties both as a report and as a video film. The results of the project can contribute to further improvement of the design of new tankers and to improvement of the safety of transport of hazardous materials on highways. This, however, requires an international or, in this case, EU-wide discussion on implementation of the results in order to avoid competitive disadvantages to the transport economy of individual European states. This process has now begun in the EU.

The Federal Ministry of Transport has continued its comprehensive research efforts. The Federal Highway Research Institute is playing a substantial part in this work.

The on-the-spot surveys at accident sites have been continued. Every year, approximately 1,000 accidents are recorded in detail and assessed. There is at present an extensive database containing 12,000 accidents involving 16,000 injured persons. 20,000 vehicles are documented. These data were used diversely, inter alia for the European work on frontal and side protection, and for work by the EEVC on pedestrian safety. These local accident surveys are incorporated in the Commission's 4th research project on child restraint systems, motorcycle helmet development and standardisation of accident surveys.

In Germany, the shock absorbers of cars are checked in the regular technical inspection as part of the general inspection in accordance with section 29 of the motor vehicle construction and use regulations (Straßenverkehrszulassungsordnung). They are essentially subjected to a visual check in which their tightness and external integrity are assessed.

The aim of one investigation was to answer the question whether it is necessary to perform a more meaningful check of the shock absorbers in the regular technical inspection of motor vehicles, since objective measurement of absorber performance is problematic and it has so far been impossible to quantify the benefits of such a measure.

Because of the uncertainties regarding the influence of defective shock absorbers on accident occurrence, and because of the defect rate determined, it has not so far been possible to prepare a final assessment of the introduction of an obligatory shock absorber test on a test stand as part of the regular technical inspection.

In order to avoid accidents which can be attributed to poor perception of an approaching motor vehicle, a project was carried out the aim of which was to improve recognisability of motor vehicles by appropriate switching the running lights. For this purpose, a switching device for the lighting system was developed which automatically actuated the headlights by means of detectors for the light intensity and by linkage with a logic circuit. The switching behaviour of the device as a function of the brightness of the surroundings was investigated in road tests. It can be stated as the result that an automatic, brightness-controlled switching device with the selected switching threshold of 3,000 lx, and the switching delays, reduces the ON period of the lighting to around 1/5 compared with the daytime running lights. Two essential improvements in road safety are achieved with the device, which, depending on the brightness of the surroundings, switches the headlights of a car: in a particular area all headlights will be switched on or off - there will be no more mixed driving with some lights switched on and some switched off, and the recognisability of motor vehicles will be improved in critical illumination situations. The introduction of such a switching device needs an EU-wide regulation.

In an expert meeting held at the Federal Highway Research Institute which concerned the road safety of truck-trailers (vehicle combinations comprising a truck plus trailer) in combination with various coupling devices and permissible lengths, the experts proposed that the permissible overall length of truck-trailers be increased from the present 18.35 m to 18.75 m, the permissible maximum loading-area length of 15.65 m being maintained. The vehicle width is increased from 2.50 to 2.55 m. This proposal was accepted at the EU ministerial council meeting in September 1995; it is now also being adopted into national law.

Accident-data recorders were installed in the vehicles of three fleets which had high accident frequencies. Accident data and further relevant information are being surveyed for the period before, after and during the field test. An investigation is to be carried out by analysis of these data, as to whether, and to what extent, installation of an accident-data recorder can lead to changes in the pattern of accidents or to, avoidance of accidents (possible preventive effect).

Children are being transported to an increasing extent in bicycle trailers the operation of which has until now been regulated only inadequately. The trailer designs, some of which are very simple, often lack safety devices for protection of the children being transported. In a pilot study, the Federal Highway Research Institute had safety aspects of the trailers, available on the market up to autumn 1993 evaluated, and both the theoretical principles of the dynamics of moving trailers and the legal framework conditions presented. In addition, a technical committee produced two information sheets; one is directed at the users, the other contains information of a fundamental nature for technical inspection institutions.

When designed correctly, active systems for controlling traction slip can increase driving stability in critical situations. The aim of one research project was to develop an objective test method for assessment of the influence of traction control systems on the dynamic behaviour of cars. The condition for suitability of an objective test method for assessment of the handling is that there is a high level of agreement of the results with the subjective assessment by the test driver. Because of their agreement with subjective judgements, certain characteristic values of the yaw velocity and the lateral acceleration have proven to be suitable for objective assessment of the handling. It became apparent that the active safety of cars can be considerably improved by complex traction control systems. It was, however, also found that simple strategies are not always capable of meeting these requirements. The influence exerted on the driving stability by brake intervention makes it clear that the driving safety can be impaired considerably in systems which are not optimally designed, or in the event of malfunction. The need for an objective testing method for investigating the influence of the system was established here, especially as the influence exerted on system quality by parameters such as loading or the tyres fitted to the vehicle was considerable.

In future, gas-discharge lamps should also be used instead of incandescent lamps in motor vehicle main headlights. These are high-pressure discharge lamps which differ from halogen incandescent lamps used so far in terms of the spectral distribution of the light emitted, the higher luminance and the higher luminous flux - whilst having a lower power consumption.

The gas discharge headlamps must not exceed the intensity of luminosity of conventional headlamps. They have to be fitted with an automatic headlight-range adjustment device and a headlamp cleaning system.

The Federal Highway Research Institute had the effects of equipping motorcycles with antilock systems (ALS) on riding safety investigated. In order to take account of the man-machine interface, which is important in particular for real traffic situations and for motorcycle braking, riding experiments were performed by a group of motorcyclists in public traffic and on a closed test section. In order to make the results comparable, the test persons rode the same, modem motorcycle on the same specified test section. Very experienced and inexperienced motorcyclists took part in the experiments. In the experiments on the closed test section, the experienced test riders achieved slightly better deceleration in straight-ahead braking on a dry pavement without ALS than with ALS (cannot be transferred to real traffic situation). On wet pavements and in curves, the deceleration without ALS is poorer than that with ALS. The ALS-controlled full braking attempts carried out in curves (radius of curve = 50 m, maximum speed at start of braking V_0 = 45 km/h) were completed with only minor course deviations and without an accident. The ALS of the motorcycle used in the test runs permits stable full braking in curves with considerable transverse acceleration. ALS permits less experienced motorcyclists in particular to decelerate hard in critical braking situations while maintaining stability.

In the Federal Republic of Germany, the regular technical inspection of all vehicles is controlled by regulations. Essential elements are, for all vehicles, the general inspection, and, for certain vehicles (trucks, buses), the additional intermediate inspection and the special brake inspection.

The EU-wide introduction of an intermediate inspection (in accordance with the "German model"), which covers all parts of the vehicle of importance for road safety, is rejected by other member states as being too costly. A discussion among experts at the Federal Highway Research Institute was intended to clarify what can be altered in the intermediate inspection without a loss in safety standards being incurred.

It was proposed as a reasonable measure that the first intermediate inspection for new vehicles take place considerably later. This proposal has now been fixed in the motor vehicle construction and use regulations (STVZO). The special brake inspection and the intermediate inspection are to be combined later in a so-called safety inspection.

At present, an extensive project analysing the effectiveness of the exhaust gas test is being carried out.

Since 1 December 1993, the commercial vehicles and cars with diesel engines, the low-emission motor vehicles with spark-ignition engines, and the vehicles with three-way catalytic converters on the road have also been included in the exhaust gas test. As part of a research project, a first assessment of the effectiveness of this measure in respect of exhaust gas emission behaviour will be undertaken.

By a suitable method, the exhaust gases of vehicles which have been found to be abnormal in the exhaust gas test are to be analysed in order to determine the levels of the legally limited pollutant components both before and after fault finding/repair. The vehicles to be inspected must be approximately representative of the vehicle population, and the selection should take account of the abnormality rates of the individual points tested in the exhaust gas test. The vehicles must be adjusted or repaired in the most realistic manner possible with the objective of rectifying only faults which would be regarded as abnormal in the exhaust gas test.

German vehicle manufacturers in particular are utilising the results of the European PROMETHEUS research project to the effect that important elements for active driving-dynamics control for cars have been prepared for series production and will be available on the market in the future. These systems are intended to make a contribution to active safety by assisting the driver in controlling the longitudinal or transverse dynamics, for example for headway control, driving stability control or tracking stability. The automatic control involves significant intervention in the engine management and the braking and steering systems with the aid of electronic components, in some cases without the direct participation of the driver.

The developments in the field of driving-dynamics control have become so complex that it is not possible for the makers of regulations to assess such systems on the basis of the existing legal provisions governing approval. The question as to what form of approval they require must be looked into separately. The aim of one, research project is to deal with the subject in an appropriate manner for fast preparation of the necessary regulations, account being taken of the ideas of other parties involved in order to guarantee the intended forceful influencing of road safety.

The Federal Government has a very keen interest in improving the safety of coach traffic. In a comprehensive report by the Federal Highway Research Institute, more than 40 measures for improving the safety of coach traffic are recommended, and proposals are made for over 20 subject complexes in which the knowledge gained so far must be deepened with further research.

The measures implemented so far include, inter alia,

- The regulation on equipping buses with speed limiters which must be set to a speed of 100 km/h.
- The regulation stating that driving licences providing entitlement to carry passengers in buses may be granted only to persons who have received theoretical and practical instruction in a driving school (in accordance with the "bus driver training directive").

Key points in the implementation of further measures are, inter alia,

- Revision of the EC directive on the conditions to be fulfilled before a person may run a bus company.
- Regulations which permit effective monitoring of the driving and resting times in road traffic, inter alia a new monitoring device in the area of the EU.
- Renewed initiatives of the Federal Government with the aim of incorporating the regulations on increasing the passive safety of coaches
  * increase in the strength of the superstructure (ECE regulation no. 66)
  * increases in the strength of the seats and inclusion of padding, in particular in the backrests (ECE regulation no. 80)
  * equipping of all seats with lap belts or three-point belts
  into the EC guidelines for buses, which are to be prepared.
- Further research into specific activities of bus safety in the "Sicherheitsforschung Straßenverkehr 1995/96" (Research into safety in highway traffic 1995/96) program.

It can be seen from the examples mentioned that the majority of the desired regulations require agreement at a European level.

The Federal Highway Research Institute was involved to a considerable extent in the basic research work for a draft directive of lateral collisions at the request of the Federal Ministry of Transport. At present, a component test method is being developed in order to enhance the protection against head injuries.

In September 1995, the 3rd revision of ECE-R 44 (child restraint systems) came into force. This revised version was prepared on the basis of the most recent findings of accident research and was adapted to the design of modern vehicles.

In a research project of the Federal Highway Research Institute, a method was developed at the TU Berlin which investigates the passive safety of cars on the basis of accident analysis and statistical biomechanics and with the aid of crash tests, and evaluates the said passive safety by means of an algorithm. Subsequently, the degree of fulfillment for the physical loading of occupants, which was measured in four different impact tests, is determined by means of the protection criteria for the respective body parts or measuring points. This degree of fulfillment is weighted by means of a relevance structure which was prepared on the basis of the data of the Medizinische Hochschule Hannover (Hannover Medical University) and reflects the significance of the body-part related injury in a real accident. By means of this assessment method, both a safety index for the vehicle as a whole and part indices for the level of passive safety in different types of collisions, or seating position- and body-related safety numbers, can be determined. The method, which is at present being validated in a large research combine with participation of associations, test institutes and the motor industry, offers, on the basis of accident analysis combined with crash tests, the possibility of assessing cars new on the market with regard to their behaviour in real accidents. The validation of the method is under the responsibility of the Federal Highway Research Institute and is performed in three phases with a total of 12 vehicle models from three mass classes.

The Federal Highway Research Institute has developed a restraint system which is intended to protect wheelchair users who, whilst sitting in their wheelchairs, are transported in a special car for transporting disabled persons. When this restraint system was being developed, account was taken above all of the results and experience gained in an earlier investigation by the Federal Highway Research Institute by means of impact tests with systems of this type already available on the market. The knowledge regarding optimum belt geometry had so far been utilised only to a small degree in the commercially available systems. Impact tests with the new restraint system have given good results. The corresponding standard is at present being revised with the assistance of the Federal Highway Research Institute.

An investigation into brushing impact of motorcycle helmets was to discuss whether it is advisable to incorporate the test method for brushing impact of a motorcycle helmet against the road surface into ECE-R 22. For this, it was necessary to clarify whether the test method is suitable for measuring rotational forces on the test head, and whether the accident situations gave rise to a need for such a test method.

The measurement results show that the connection between the injurious rotational acceleration and the measured tangential force cannot be verified without any doubt. It should thus be demanded that the rotational acceleration rather than the tangential force be considered as a criterion for injury. This, however, would place very a high demand on the measuring equipment in the laboratory. From the point of view of accident research, it can be said that such impact situations, in particular with impact speeds below 40 km/h, which can be realised in the laboratory, lead only to moderate head injuries (AIS 2 or lower). The high cost of the test method would thus appear not to be justified.

Since 1993, the EC commission has provided financial assistance for the EEVC working group's investigations for preparation of a draft regulation for a new legal frontal crash test method. Within this project technical specifications have been developed, such as test configuration, test velocity and biomechanical criteria.

Furthermore, the Federal Highway Research Institute took part in research work on pedestrian protection. The research group EEVC/WG 10 conducted studies and presented suggestions which include requirements for bumpers, engine hood fronts and the hoods themselves; prototypes of the impactors needed for the tests are already available. The subject "Pedestrian protection" has been introduced into the future Work Program as a matter of priority by the European Commission.

In addition to its own research projects, which also include investigations for the Federal Ministry of Transport and for the commission of the EU, the Federal Highway Research Institute also performs crash tests for third parties such as the technical inspection agencies and the motor-vehicle and supplying industries. The primary motivation for this is not economic but rather concerns the gain in knowledge and experience which makes it possible to give qualified advice to the Federal Ministry of Transport. The tests performed for the ADAC on various subjects: small cars, spacious limousines, new frontal crash test method, vehicle-vehicle crash tests; tests for the motor-vehicle industry: side impact tests, pedestrian tests in accordance with EEVC WG10 test method; supplying industry: component tests in respect to individual parts of vehicles, are particularly emphasised here.

The Federal Highway Research Institute is having the research project on protection criteria for the human head carried out by the Berlin Technical University (Prof. Appel) and the University of Heidelberg (Prof. Mattern).

The aim of the project is, with the aid of the human head model developed, to determine injury parameters by means of medical analyses and model computations. The relevance of these parameters is to be checked in laboratory experiments in order that limiting values for use in legal standards be determined. The protection criteria refer to vehicle occupants (frontal and side collision, with and without head impact) and to pedestrians.

At the University of Heidelberg, an investigation is being conducted into the injury mechanics and the loading capacity of the foot for the Forschungsvereinigung Automobiltechnik e.V. (Association for Research into Automotive Engineering). From the experimental results gained so far, it is clear that the lower extremities of today's dummies are poorly suited for assessment of injury risks to human legs. The current dummy lower leg would have to be revised fundamentally before the said injury risks could be assessed.

Following conclusion of the investigations in June 1996, results concerning the foot injuries received by persons at specified impact speeds will be available, as will, as a consequence, possible injury predictors.

In recent years, the emergency service in the Federal Republic of Germany has developed into a nationally and internationally recognised system.

According to the extrapolations, which are being prepared continuously, the emergency service answered 8.5 million calls in 1995. 60% of these calls were for ambulances (urgent and non-urgent) and 40% for rescue operations (with and without emergency doctor). This means that, on average, one in every 10 inhabitants made use of the emergency service and that rescue operations were required by 43 out of every 1000 inhabitants.

Around one in eight (11.9%) rescue operations was due to a road accident. The percentage of road accidents in the total number of rescue operations has decreased continually over the years. 20 years ago, it was around 27.2%.

In their accident research, the German motor insurance companies (VdS) have systematically extended the databases relating to accidents involving injury to persons.

On the basis of 15,000 car accidents registered in Vehicle Safety 90", material is now available on approximately 1,600 serious car-car accidents and accidents involving one car alone (MAIS 3+).

The motorcycle file was extended to 600 accidents with serious injuries. It is the foundation for intensive investigations into the risk of injury to motorcyclists and has been compiled in accordance with the newest definitions of the ISO standard. The first stage of the CEN protective clothing standard (prEN 1621), which contains realistic requirements to be met by the protectors, has been completed.

As regards active safety, motorcycle accidents were investigated with regard to critical situations and avoidance strategies. In their training and further training, motorcyclists be made more aware of typical accident sequences.

As before, accident risks of young car drivers are a key area of investigation. Measures for improving practical training of drivers, and a model for a second phase of driver training, were developed.

Investigations of 5,000 accidents in a project conducted jointly with the University of Würzburg showed that even a low alcohol concentration more than doubles the accident risk of young drivers, which is high anyway; when the blood alcohol concentration is above 80 mg/100 ml, the risk is 58 times higher than when no alcohol has been consumed.

Work is being done on a file relating to car-pedestrian accidents. By the end of 1996, this material should comprise approximately 800 accidents and form a basis for decisions on future safety criteria within the framework of ECE/EU directives.

In the case of commercial vehicles, work in respect of active safety (contour markings) and passive safety was continued. Within the framework of the EEVC, in a project in which five countries are participating, requirements to be met by an energy-absorbing front underride protector are being derived from real accidents and crash tests. This work is done on behalf of the Federal Highway Research Institute. Previous car-truck experiments show that, if the truck is fitted with a rigid front underride protector, the load limits of the car are reached at a relative speed of 56 km/h. It would be desirable to extend the protection range to approximately 75 km/h by means of energy-absorbing front underride protectors.

On the basis of the results of real accidents, a working group which is to prepare an improved draft regulation for rear underride protectors has been formed in Germany. In particular, the resistance of the rear underride protector of the lorry in the event of one-sided loading must be improved, and a better protective affect must be achieved by reduction of the ground clearance.

Eight countries are now participating in the ISO "Side Impact Studies with Children in Cars" working group coordinated by the VdS. 140 cases with serious injuries to restrained children (MAIS 2/3+) were collated.

Basic parameters for lateral crash tests of child seats in cars were examined and reproduced by institutes in a series of sled tests with different test methods. The foundations for the draft of an ISO standard are to be submitted at the end of 1996/beginning of 1997.

In a research project of the Federal Highway Research Institute, 600 accidents involving restrained children in cars were evaluated, and observations made concerning misuse of child-protection systems in 250 cars. Current child-protection systems could be more effective if approximately 50% of them were not incorrectly used to at least a considerable degree, which depends to a large extent on the type of protective system. Backward orientated child seats must not be mounted on front passenger seats fitted with airbags.

This incorrect operation rate was reduced to below 10% with the so-called "ISOFIX system". This was shown by a VdS study with an ISOFIX prototype which was installed in a vehicle by 150 users. In future, when child-protection systems are being developed, the same importance should be attached to avoidance of incorrect use as to the objective testing of the safety criteria in the approval tests.

In the accident research of the VdS, which was performed together with the ADAC, around 300 accidents involving cars equipped with airbags were assessed. In a large proportion, airbags are still triggered at low impact speeds of approximately 15 km/h, even though the protective effect of the seat belt alone would be quite adequate. The VdS proposes a triggering threshold corresponding to a wall impact at 25 to 30 km/h. Moreover, "intelligent" (smart) front-passenger airbags, which sense when the seat is occupied, should be used in future. If all cars were equipped with air bags, unnecessary triggering of airbags, including cases where there is no front passenger in the car, would lead to unnecessary repair costs amounting to approximately 100 million DM.

Investigation of rear-end collisions and the resulting injuries is a further key area of the VdS. New field investigations have shown that two thirds of the headrests are incorrectly adjusted and that the range over which they can be adjusted is often too small.

In a research project conducted jointly with the

Technical University of Graz, rear-end collisions were simulated in the speed range up to 12 km/h in 40 sled tests with voluntary test candidates. The experiments proved that vehicles with poor seat quality and unsatisfactory headrests also lead more frequently to cervical vertebrae injuries in real accidents.

Due to the serious economic significance of cervical vertebrae injuries, research projects relating to such injuries, their inclusion in international safety standards, and their medical assessment criteria, should be improved.

The effects of driver assistance systems on active safety are a further key area. In a joint project with the motor industry, the Technical University of Darmstadt and the Dresden University, it is being examined, on the basis of the accident material of the motor insurers, in what proportion of real accidents driver assistance systems would have influenced the accident, and what demands should therefore be made, also in respect of the problem of risk compensation, on driver assistance systems.

The contributions of the German motor-vehicle industry can be found in the various technical seminars of this conference.

We shall follow the presentations in the next few days with great interest. We also wish this year's ESV conference every success.

France

Jean Pierre Medevielle
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INTRODUCTION

This report concerning France emphasises the work carried out over the last three years on road safety and research into road safety, in particular that having repercussions on the safety of vehicles.

Many actions are no longer conceived of at national level alone, whether they concern regulations and/or standardisation, in which France contributes to the works of ESC, ISO, EEVC, ECE, COST and the European Union, or whether they concern road safety policy as set out by the respective jurisdictions of the European Union and its member states or regional and local authorities.

THE PROGRESSION OF ROAD SAFETY IN FRANCE

Since the last ESV conference, figures have improved, with the number of dead within 6 days continuing to fall below the level of 9,000 annual deaths since 1993 (8,533 in 1994 and 8,412 in 1995, which is the best figure since 1954). Likewise for serious injuries over the last two years (39,257 serious injuries in 1995), with slight injuries suffering from a certain stagnation (142,146 slight injuries in 1995) as with the number of bodily injuries (132,987 in 1995).

On the European level, for the number of deaths in relation to the population, France ranks tenth out of the fifteen member countries and ranks eighth for the number of killed in relation to the number of vehicles.

Rates concerning the wearing of restraints in the front seats have progressed.

The fall in the number of motorcyclists and pedestrians killed continues and is close to 1,000 pedestrians killed per year.

ROAD SAFETY POLICY

A large number of measures were taken during the period from 1991 to 1993, and the policy was continued in 1994-1995 and at the beginning of 1996.

Besides the establishment of a new Penal Code which increases the severity of penalties concerning road safety, and the designation of "endangering the life of another person" as a crime to be tried before the courts, the application of a driving license by points was strengthened in order to encourage:

- Car drivers to wear their seat restraints (-1 point).
- Moped riders to wear their helmets (-1 point).

To reinforce the fight against alcoholism at the wheel, the proportion of alcohol constituting a misdemeanour was lowered to 0.7 g/l in July 1994 and then to 0.5 g/l in September 1995.

Furthermore, the technical inspection of light vehicles was reinforced at the beginning of 1996. This concerned the frequency (the first inspection during the fourth year of the vehicle's lifetime then renewed every 3 years) and the number of functions subjected to obligatory repair (16 additional points).

Lastly, France actively participates in the preparation of future international and European regulations within the UN Economic Commission for Europe and in the directives of the European Union.

THE PROGRESSION OF RESEARCH AND DEVELOPMENT

The modifications made to the treaties of the European Union have reinforced its jurisdiction with respect to research and development and transport safety. The French partners are involved in projects concerning road safety within the framework of research and development in "Transport", "Materials Technology", "Standards and Tests" and "The Use of Telematics in Transport" which are constituent programmes of the 4th European Union Research and Development Framework Programme. However, the place given to road safety in these programmes remains modest. Evaluations of demonstrations of Carminat (an in-vehicle information system) have been carried out.

The PREDIT Vehicle and Road Safety Programme, carried out jointly by PSA and INRETS, has progressed, as you will be able to see during this conference.

Besides these works, we have continued in the four directions announced in 1994:

- The introduction of new methods concerned with providing solutions to the problem of serious injuries in addition to that of fatalities with attention being given to relationships between ethics and science (e.g., new epidemiological methods applied to safety, injury mechanisms, simulation-modelling-demonstration-observation).
- The transfer of road safety concepts to other modes of transport and vice-versa (e.g., secondary safety applied to guided transport, the introduction of cindynics and anthropotechnics in road vehicles).
- Multidisciplinary work on the trio infrastructure-vehicle-driver-passenger, not forgetting the problem of our ageing society.
- The compatibility between vehicles using the same infrastructure.

Lastly, France has decided to launch a new research and development programme concerning transport from 1996 to 2000. In this new programme, research will focus on "Safety and Ergonomy", with particular attention being given to all causes of accidents, human as well as technological. It will concern the following areas:

- Accidentology and biomechanical studies.
- The modelling of collision phenomena.
- Imminent hazard warning systems and the localization of accidents.
- Training and safety tools.
- Vehicle design and infrastructures.

CONCLUSION

You will be presented with a large number of French works during this conference, from the manufacturers, the equipment suppliers and the research institutes, and I think that they will demonstrate the challenge France intends to take up with respect to road safety.

Japan

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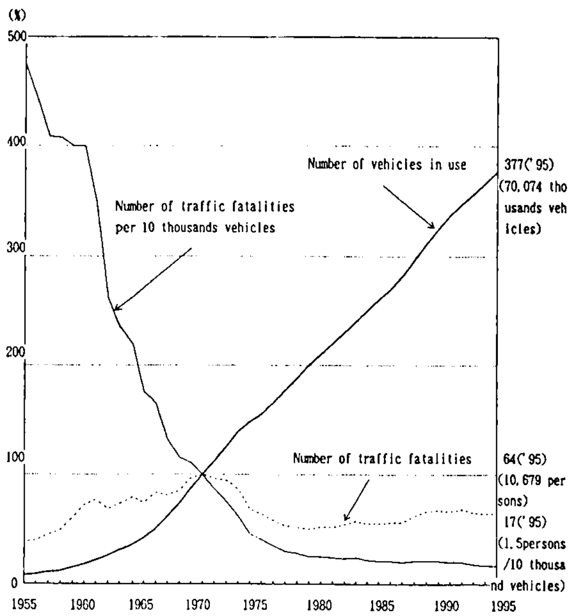
Ministry of Transport

ABSTRACT

This paper will report the current status of the motor vehicle safety measures that have been undertaken in Japan and look at Japan's approach to safety in the future.

TRAFFIC ACCIDENTS AT PRESENT

There were 10,679 fatalities in traffic accidents in Japan in 1995, which marked an increase of 0.3% over the figure for the previous year. What is serious is that the annual number of traffic fatalities has exceeded ten thousand for eight years running, ever since 1988. Nonetheless, the number of traffic fatalities per ten thousand motor vehicle has been declining slightly because the number of vehicles in use has been increasing gradually year by year. (Refer to Figure 1.)



Note:

1. The number of motor vehicles in use is according to MOT materials. Figures on the number of vehicles in use are as of the end of year.
2. The number of traffic fatalities is according to National Police Agency. Figures on the number of traffic fatalities are as of the end of year.

Figure 1. Trends Traffic Benchmark Indices (1970=100)

Several characteristics of these traffic accidents stand out. The number of fatalities in the young segment aged 16-24 years and in the elderly segment aged 65 and over totaled 5,656 in 1995, thereby accounting for more than half of all traffic fatalities. (Refer to Figure 2.)

An examination of the circumstances of the accidents reveals that the largest number of fatalities occurred to occupants of motor vehicles, which accounted for 42.6% of all fatalities. Moreover, this percentage is showing a tendency to rise. Broken down by circumstance, fatalities to pedestrians rank second, but this is showing a tendency to decline. Fatalities to vehicle occupants and pedestrians combined account for 70.6% of total traffic fatalities. (Refer to Figure 3.)

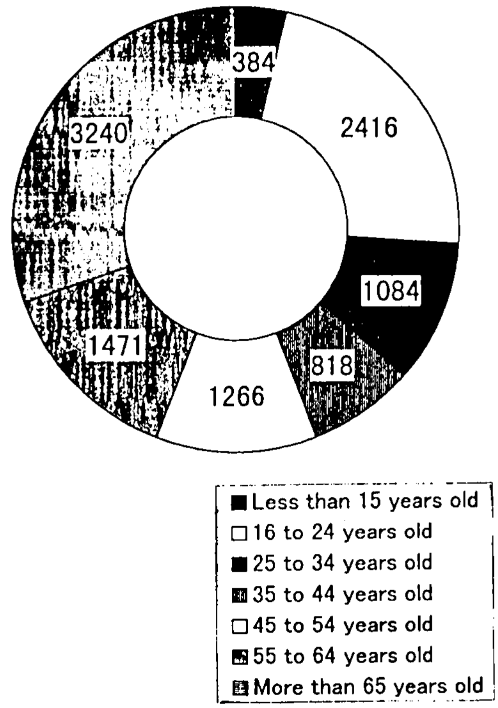
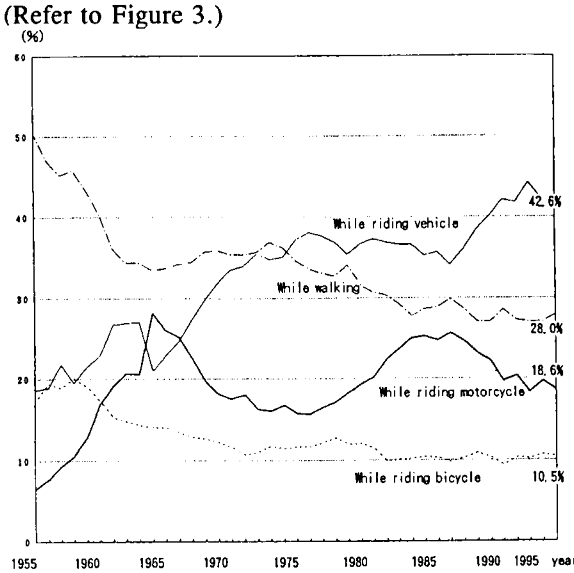


Figure 2. Number of Traffic Fatalities Broken Down by Age Segment (Unit: Persons)

An examination of the circumstances of the accidents reveals that the largest number of fatalities occurred to occupants of motor vehicles, which accounted for 42.6% of all fatalities. Moreover, this percentage is showing a tendency to rise. Broken down by circumstance, fatalities to pedestrians rank second, but this is showing a tendency to decline. Fatalities to vehicle occupants and pedestrians combined account for 70.6% of total traffic fatalities. (Refer to Figure 3.)



Note:

1. According to National Police Agency materials.
2. Prior to 1971, Okinawa Prefecture is not included.

Figure 3. Trends in Traffic Fatalities Broken Down by the Circumstances of the Accidents

The young segment aged 16-24 years accounts for about 30% of the traffic fatalities to motor vehicle occupants, while the elderly segment aged 65 and over account for about 55.5% of the fatalities to pedestrians. Obviously, these two segments account for a high proportion of traffic fatalities. (Refer to Figures 4 and 5.)

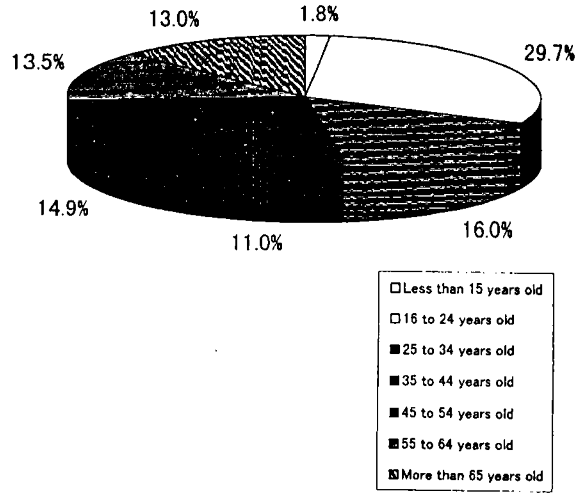


Figure 4. Proportion of Traffic Fatalities While Riding Vehicles Broken Down by Age Segment

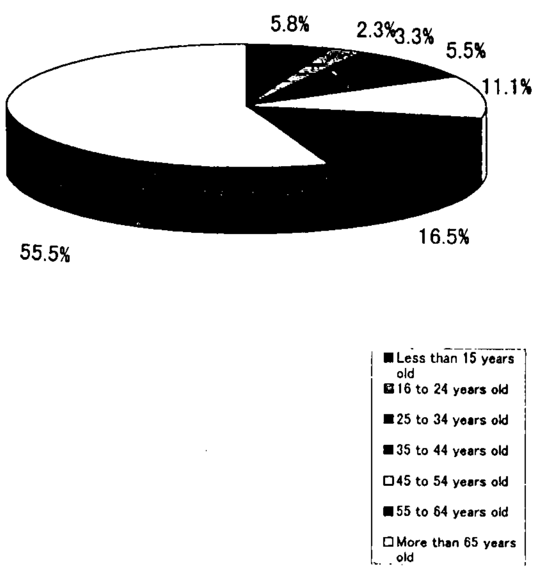


Figure 5. Proportion of Traffic Fatalities While Walking Broken Down by Age Segment

Furthermore, an examination of the proportion of traffic fatalities broken down between nighttime and daytime reveals that 55.4% occurred at night. (Refer to Figure 6.)

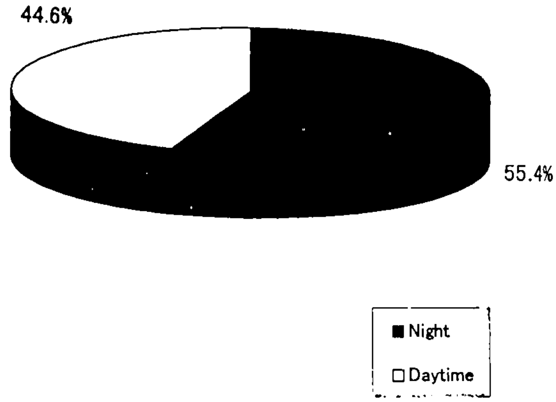


Figure 6. Proportion of Fatal Accidents by Day and Night

In the same vein, an examination of the proportion of traffic fatalities broken down by circumstance shows that 46.7% occurred in vehicle-to-vehicle accidents, 28.3% in vehicle to pedestrian accidents, and 24.3% in single-vehicle accidents. So, the largest number of fatalities occurred in vehicle-to-vehicle accidents. (Refer to Figure 7.)

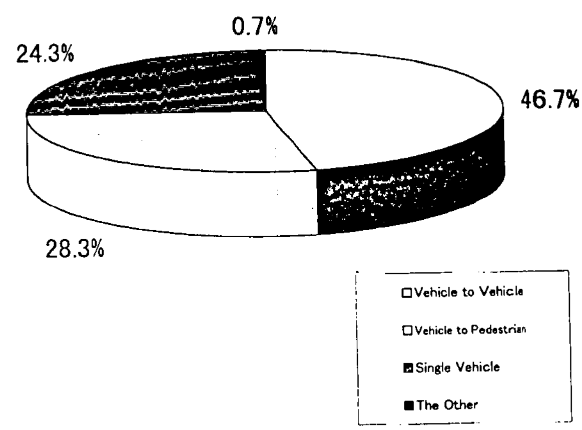


Figure 7. Proportion of Fatal Accidents Broken Down by the Circumstances of the Accidents

THE INVESTIGATION OF ACCIDENTS

Obviously, traffic accidents are the result of complex interrelationships of factors involving people, motor vehicles, and roads. To reduce accidents, therefore, countermeasures will need to address each of these factors — people, motor vehicles, and roads — and at the same time it will be increasingly important to take account of the effective interplay between the various countermeasures. It was for this reason that the Ministry of Transport, the

National Police Agency, and the Ministry of Construction jointly established "Institute for Traffic Accident Research and Data Analysis" in March 1992, with the objectives of consolidating and putting into usable order the wide range of data that have been accumulated and used by various organizations and of supplying basic data that will facilitate the establishment of effective traffic safety measures.

The Institute carries out macro-investigations and surveys by conducting analyses of accidents based on huge volumes of data obtained from a wide range of sources, and also carries out detailed micro-investigations conducted by specialists at the scene of accidents. To facilitate these investigations, a survey office was established in 1993 in the city of Tsukuba, on the outskirts of Tokyo, where specialists have been permanently stationed to conduct investigations of traffic accidents.

THE IMPROVEMENT OF MOTOR VEHICLE SAFETY

The Council for Transport Technology, which is an advisory organ to the Ministry of Transport, recommended in 1972, in 1980, and in 1992 its first, second, and third sets of targets for the expansion and strengthening of motor vehicle safety standards. These recommendations covered accident-avoidance measures, damage-decreasing measures, damage anti-expansion measures, and other items, and provided guidance on methods of implementing regulations and studies concerning each item as well as specific time frames. The various regulations implemented so far based on the Council's third recommendations and related research will be explained below.

Regulations Implemented to Date

Traffic accidents in Japan in recent years reflect certain pertinent factors — the increase in the speed of traffic as a result of the spread of the expressway network, the increase in activities carried on at night due to diversifying lifestyles, and the progressive aging of the Japanese population. To address accidents involving such factors, Japan has expanded and strengthened its safety standards as indicated in Chart 1.

Scheduled Expansions of Safety Regulations

The Ministry of Transport plans to expand and strengthen safety standards in the near future as indicated in Chart 2.

The Promotion of Safety Studies

In addition to working to expand such regulations, the Ministry of Trasnport intends to carry out studies to improve vehicle safety, of which a certain portion are already in progress. The relevant items are indicated in Chart 3.

AUTOMOBILE SAFETY INFORMATION

In addition to such efforts to improve vehicle safety through regulation, MOT has begun trying to improve vehicle safety by another means -- by influencing users' choices through the provision of safety information. Since fiscal 1991, National Organization for Automotive Safety and Victims' Aid has, under MOT guidance, begun deliberation of what items of safety information to provide to users and methods of actual vehicle testing. It has been doing this as part of an experimental initiative on the provision of safety information.

Data on the safety performance of motor vehicles, including safety in collisions, etc., was provided in March 1996 on a trial basis to vehicle users in an easy-to-understand format entitled "Automobile Safety Information." The data presented took account of the results to date of the deliberations and revealed the real names of the vehicle models tested. The Organization intends to intensify its deliberations to embrace items for comparative testing and methods of evaluation in an effort to continue providing valuable information.

ADVANCED TECHNOLOGY

Japan has since fiscal 1991 been implementing a five-year plan based on cooperation between the government, private, and academic sectors. By making practical application of advanced electronics technology, this plan seeks to realize the Advanced Safety Vehicle (ASV) of the next century, a vehicle that incorporates highly sophisticated capabilities. Demonstration runs have been held of 19 ASV test models. These models feature the achievements of five years of intensive research, including automatic collision detection and prevention systems, drowsy/nonattentive driver warning systems and practical navigation systems. The next five-year plan commenced from fiscal 1996 and will include considerations from the perspectives of harmonizing and integrating the vehicles with the infrastructure and of the human interface, in addition to research on the motor vehicle itself. The aim will be to promote

Chart 1.
Measures Concerning The Regulation Completed So Far

Item	Applicable model	Contents being carried out	Implementation Date
Damage Decreasing Measures:			
Frontal collision	Passenger vehicles	New regulations for impact absorption capacity of body for passenger cars were established, by requiring frontal collisiotest on sample vehicles.	April 1, 1994 for new type-approved vehicles [January 1, 1996 for other domestic vehicles; April 1, 1999 for imports]
Three point seat belts at rear seat	Passenger vehicles Multi-purpose vehicles and small-sized trucks	Installation of three-point seat belts equipped on outer sides of rear seats were required.	April 1, 1994 [April 1, 1995 for imports]
Warning device for seat belts left unfastened	Passenger vehicles Multi-purpose vehicles and small-sized trucks	Installation of warning devices for seat belts left unfastened were required.	
Seat belt with pretensioner	All vehicle models (except two-wheeled motor vehicles)	Heat resistance test and cold resistance test were added as the test requirements of seat belt with pretensioner. Requirements was established not doing mistaken operation in these tests.	April 1, 1995 for new type-approved vehicles
Child seats pretensioner	All vehicle models (except two-wheeled motor vehicles)	Requirements, dynamic test , etc., were established for seat built-in type child seats.	April 1, 1995 for new type-approved vehicles
Accident Avoidance Measures:			
Braking performance	Passenger vehicles	Standards were established concerning breaking performance at high-speed, operational stability during braking and anti-fade capacity.	April 1, 1994 for new type-approved vehicles [January 1, 1996 for other domestic vehicles; April 1, 1999 for imports]
Large-sized rear reflecting plates	Medium-sized trucks	The vehicles for which installation of large-sized rear reflecting plates are required were expanded.	September 1, 1995 [September 1, 1996 for vehicles in use]
Field of vision in bad weather, such as rainy day	Passenger vehicles Multi-purpose vehicles	Quantitative standards on wipers (wiping cycle and areas) and on defrosting system performance were established.	April 1, 1994
Illumination of stoplamps when auxiliary brake (retarder) is in operation	Trucks & Buses	Vehicles equipped with electromagnetic-type retarder were required to have stop lamps turned on while the retarders are in operation.	April 1, 1994 [April 1, 1995 for vehicles in use]
Operating force of brake	Passenger vehicles	Operating force of brake were reduced to attain easier operation of brakes.	April 1, 1994 for new type-approved vehicles [January 1, 1996 for other domestic vehicles; April 1, 1999 for imports]
Anti-lock brake system (ABS)	Large-sized trailers	The vehicles for which installation of ABS are required were expanded to include ordinary large-sized trailers.	September 1, 1995
Double accelerator return spring	All vehicle models (except two-wheeled motor vehicles)	Installation of double accelerator return spring were required to ensure the return of the accelerator pedal in case of breakdown of one spring	April 1, 1994
Lower beams of headlamps	All vehicle models	To maintain and assure performance of lower beams of headlamps, concrete requirements shall be set forth as to the direction of main photometric axis and the minimum requirement of luminouce intensity.	February 1, 1996
Damage Anti-expansion Measures:			
Prevention of fuel leakage in rear-end collision	Passenger vehicles Multi-purpose vehicles and small-sized trucks (except mini-sized vehicles)	The regulations concerning prevention of fuel leakage in rear-end collisions were strengthened.	April 1, 1994 for new type-approved vehicles [January 1, 1996 for other domestic vehicles; April 1, 1999 for imports]
Flame-resisting materials of interiors	All vehicle models (except two-wheeled motor vehicles)	Use of flame-resisting materials for vehicle interiors was required.	April 1, 1994
Adoption for new materials	All vehicle models (except two-wheeled motor vehicles)	Requirements were established for the case using the organic glass at the area except windshield glass, using the glass-plastic at the area including windshield glass (including for the case of laminated glass) Requirement were chemical resistance test and burning resistance test, etc.,.	September 30, 1994

Chart 2.
Expansion of the Regulation Planned in the Future

Item	Applicable model	Contents
Accident Avoidance Measures:		
Field of vision in bad weather, such as rainy day	Trucks & Buses	Standards shall be set forth quantitatively as to wiping performance (cycle requirements and wiping areas) and on defrosting system performance.
Field of vision of motor vehicles having high vehicles height	Multi-purpose vehicles	On multi-purpose motor vehicles, the range of direct or indirect field of vision at immediate front side and at immediate left side of the motor vehicles that is necessary from driver's seat shall be set forth quantitatively.
Conspicuousness of two-wheeled motor vehicles	two-wheeled motor vehicle	Regulation shall be stipulated as to obligatory adoption of such circuitry that will automatically turn ON headlamps of two-wheeled motor vehicles when engine is operating.
Braking performance	Multi-purpose vehicles, buses, truck and two-wheeled motor vehicle	Requirements shall be stipulated to assure (1) braking performance during high-speed running; (2) vehicle direction stability in braking (3) fade-withstanding characteristics
Operating force of brake	Multi-purpose vehicles, buses, truck and two-wheeled motor vehicle	Efforts shall be made to reduce the operating force of brakes from the viewpoint of attaining easier operation of brakes.
Damage Decreasing Measures:		
Frontal collision	Multi-purpose vehicles and small -sized trucks	Efforts shall be made to improve impact-absorving performance of motor vehicle as a whole,including the construction of vehicles body, by making it obligation to enforce frontal collision test by means of sample vehicles.
Lateral collision	Passenger vehicles multi-purpose vehicles and small small-sized trucks	Efforts shall be made to improve impact-absorving performance of motor vehicle as a whole,including the construction of vehicles body, by making it obligation to enforce lateral collision test by means of sample vehicles.
Rear underrun protection device	medium-sized trucks	Vehicle models where the installation of rear underrun protection device is obligatory shall be expanded.
Damage Anti-expansion Measures:		
Prevention of fuel leakage in rear-end collision	Mini-sized vehicles	Requirements for prevention of fuel leakage in the event that the rear end of motor vehicles is collided by other motor vehicles shall be strengthened.

Chart 3.
Outline of the Study Items

Item	Applicable model	Contents
Accident Avoidance Measures:		
Cornering lamps	All vehicle models (except two-wheeled motor vehicles)	Study to the effectiveness of cornering lamps to ensure visibility during night-time/mountain driving and at the time of right/left turns.
Driving posture of two-wheeled motor vehicle	two-wheeled motor vehicle	Study to positional relationship among the seat, handle grip and foot rest.
How the whole lighting and light signalling devices should be	All vehicle models	Comprehensive study to improve visibility of vehicle signals and to clarify signal communication by lamps
Display devices for distance between vehicles	passenger vehicles and multi-purpose vehicles	Study to standards for visual display devices such as navigation systems.
Resistance to electromagneticg waves	All vehicle models	Study to durability standards and assessment methods against such outside factors as electromagnetic waves and static electricity on electronic devices in the on-vehicle status.
Fail-safe functions of electronic devices		Study to fail-safe system for electronic devices.
Maximum speed and maximum output		Comprehensive review of maximum speed and output.
Steering wheels having a small diameter	passenger vehicles	Study to relationship between the modification ofsmall-sized steering wheel and steering stability.
Loading weight meters	Trucks	Study to precision improvement for loading weightmeters.
Running stability	Trucks and double deck buses, etc.	Study to driving stability of vehicles with high center of gravity.
Damage Decreasing Measures:		
Improvement of seat belts	All vehicle models (except two-wheeled motor vehicles)	Study to installation position of belt anchorage and on improving belt pressure.
Front construction ,etc.		Study to structure of the front part of vehicles to protect pedestrians at time of collision.
Front underrun protector	Medium- and large-sized trucks	Study to installation of front underrun protectors.
Damage Anti-expansion Measures:		
Prevention of ignition	All vehicle models	Study to prevention of electric circuit damage at time of collision.
Seat belts cutter	All vehicle models (except two-wheeled motor vehicles)	Study to necessity of seat belt cutters.
Window glass breakage hummer		Study to necessity of Window glass breakage hummers.
Openning/closing of door in collision		Study to the requirement concerning Openning/closing of door in collision.
How warning system should be	All vehicle models	Study to systems that appropriately warn of danger to the vehicle and the driver.
Information communication with outside vehicle		Study to the effective exchange of information with the outside of vehicle.
Adoption for alternative energy vehicles		Study to standard for alternative energy vehicles.
How confirmation of function of onboard electronic equipment should be		Study on standardization of performance and structure for self-assessing devices.

the development of technology related to the safety of the ASV.

ASV, which is predicated on the assumption that the motor vehicle is being operated by the driver, uses a variety of sensors to determine the driver's condition and the running environment of the motor vehicle. It is expected to warn the driver when danger occurs and to have the capability of automatically stopping the vehicle if necessary. For example, if the driver should doze of while driving, the envisioned system will sound an audio alarm and perhaps even have the systems technology to emit odors and vibrate the seat in an effort to wake the dozing driver. Also under consideration are systems technologies that will be able to judge when to stop the vehicle and when to take evasive action if the driver should fail to take appropriate action to counter obstacles on the road.

At the same time, a wide range of research and development work is proceeding on Intelligent Transport Systems (ITS). Through systems that integrate people, roads, and vehicles, applying advanced information technology including ASV, ITS has several objectives: vastly improving traffic safety; greatly improving transportation efficiency and comfort; and making significant contribution to environmental concerns by facilitating the flow of traffic and reducing congestion.

INTERNATIONAL HARMONIZATION

MOT believes that each country's motor vehicle standards must reflect that country's social customs, the traffic accident situation, and the traffic environment. With the rapidly advancing internationalization of cars as products, MOT feels that it is particularly necessary to devote great effort to the harmonization of international standards in order to ensure smooth international distribution and, specifically for Japan to facilitate the import of motor vehicles.

It is for these reasons that Japan is stepping up its participation in the international standards harmonization activities of the United Nations Economic Commission for Europe, Working Party on the Construction of Vehicles (ECE/WP29). In addition, Japan is promoting international harmonization through the activities of the Japan Automobile Standards Internationalization Center (JASIC), which was established in 1987. Japan also announced its intention to participate in the UN-ECE (United Nations Economic Commission for Europe) 1958 Agreement. This is an agreement on the mutual recognition for motor vehicle equipment. Moreover, Japan plans to actively step up its efforts toward the international harmonization of motor vehicle standards in the future.

It should be noted that Japan has taken the lead over Europe and the United States in regard to the formulation of internationally harmonized standards. Examples include standards for passenger car brake performance, the installation position of motor vehicle lamps, and the distribution characteristics by of head lamps.

Canada

Harvey Layden
Transport Canada

Canada is a large country—about 9.2 million square kilometers in area—with a small population—about 29,000,000 people—concentrated mainly near the southern border. Its economy relies heavily on a large network of highways, about 900,000 kilometres in length.

Over the past 25 years, traffic deaths have decreased by more than 46%. During this same period, the number of licensed drivers and registered vehicles has more than doubled. While this reduction in fatalities can be attributed to a number of factors, one of the most important is the increased use of seat belts. Statistics show that, since 1975, seat belt use in Canada has increased from about 20% to over 90%. As mentioned at an ESV Conference, in 1989, Canada set itself the goal of 95% seat belt usage by the year 1995. This goal has been achieved, and even surpassed in some areas of the country.

Last time, we also reported our intention to revamp Canada's occupant protection requirements. Consistent with our position that seat belts provide essential primary protection, Canada will be making seat belts mandatory in all vehicles--which is not explicitly stated in our current regulation. Air bags will remain optional because they offer supplemental protection only.

As part of this initiative, Canada recently published more stringent requirements governing head and chest protection. We have proposed replacing the presently used Head Injury Criterion with an 80-g head acceleration limit for light-duty vehicles equipped with seat belts only. For vehicles equipped with both seat belts and air bags, manufacturers will be allowed to meet either the new head acceleration limit or a reduced HIC of 700, as calculated over a shorter maximum time interval.

With regard to chest protection, we are also replacing the present testing criterion by chest compression. A provisional limit of 65 mm has been set for light-duty vehicles in order to give vehicle manufacturers time to make the necessary changes. Commencing September 1999, the limit will be lowered to 50 mm.

Originally, Canada also intended to introduce new requirements governing seat belt fit. A standard was proposed, based on the Belt-Fit Test Device, which measures how well a seat belt is likely to fit a human occupant. However, the automotive industry requested that additional research be done, and a joint government-industry project was set up. This research program, which will establish the best means of minimizing the risk of belt-induced abdominal injury, should be completed by early 1998. It is expected that a belt fit standard based on the Belt-Fit Test Device, or an equivalent, will be in place effective September 2001.

In addition to improving occupant protection in frontal collisions, we are also engaged in developing dynamic side impact protection criteria. We have conducted extensive research evaluating the testing protocol of the United States and that proposed by the Economic Commission for Europe. While the U.S. requirements are appropriate to the Canadian context in many ways, we are concerned about the adequacy of its mandated test dummy. We have not as yet determined what our requirements will be and intend to work together with industry to develop a standard that will provide motorists with the best side impact protection that is available.

In addition to these regulatory initiatives, we are also involved in four research projects of note. One study looks into the factors that contribute to heavy truck accidents; another is studying the injuries being suffered by belted drivers from deploying air bags; a third focuses on the injuries sustained in moderately severe side impact collisions; and another, which is being conducted by our ergonomics division, will examine behavioural adaption to Intelligent Transport Systems

As part of an endeavour to improve the safety standards governing heavy trucks, we are systematically analyzing national police-reported data on heavy truck collisions. We are also conducting a series of related and detailed studies in regions with different topographies and highway design practices. A Heavy Truck Safety Pilot Study, which is being conducted in the province of New Brunswick, has examined 40 cases to date. It has revealed an unexpectedly high incidence of presumptive driver fatigue, disrupted circadian rhythm, and environmental factors in rollovers. A heavy truck safety study is also being conducted by a federal-provincial, multi-disciplinary team in the province of British Columbia, with a similar study being developed in Saskatchewan. While not statistically representative, such studies provide details that are essential for effective regulation.

Like many of you, we are very concerned about the increasing incidence of air-bag-induced injuries. Many minor to moderate injuries are being suffered by drivers due to low deployment thresholds, a topic on which Dainius Dalmotas will be reporting in a later session. The most serious of these injuries, some of which are fatal, occur to small female drivers. In order to better understand the dynamics involved in these incidents, we have recently conducted a series of ten 48 km/hr full frontal barrier tests using 5th percentile female dummies in forward seating positions.

These tests will be followed by a further series, also using small female ATDs, but at a lower speed and against the fixed offset deformable barrier recently proposed by EEVC Working Group 11. The test protocol approximates the conditions in which small female drivers are sustaining disproportionately severe injuries. The data from these two test series will be used to develop supplementary performance requirements for occupant protection systems. Canada has been fortunate in that we have not seen the same incidence of child injuries and deaths as reported in the United States, a fact we attribute to the importance our population places on the use of seat belts.

As part of our continuing research effort to develop an effective side impact protection standard, we are conducting a detailed study of moderately severe real-world side impact collisions. This project focuses on the specific crush profile of the struck vehicle and the injuries sustained by nearside occupants. These data will enable us to correlate actual injury mechanisms to the test data collected from instrumented test dummies in staged collisions, which will help in defining appropriate injury criteria for the purposes of regulation.

Unlike the previously mentioned research projects, our study of behavioural adaptation to Intelligent Transport Systems, does not relate directly to the development of safety standards. This research project will examine whether behavioural adaptation occurs to a Fatigue Warning System and whether adaptation, if it occurs, is affected by personality type. About fourteen tired participants, half of whom have a high sensation seeking score and half of whom have a low score, will complete a test track drive with a Fatigue Warning System. Several types of measures will be recorded, and the results will be compared with those of a control group in order to determine whether drivers with high and low sensation seeking scores show differences in levels of fatigue, driving performance, and adaptation to the Fatigue Warning System.

In closing, I would like to warmly welcome you to join us in Canada for the 1998 ESV Conference. It will be held in Windsor, Ontario—Detroit's twin city—in early June. We look forward to seeing you there.

The Netherlands

Gerard Meekel
Department of Road Transport

INTRODUCTION

In the Netherlands the trend in the number of fatalities and injuries, as a consequence of traffic accidents, has been decreasing since 1972. In this year the number of fatalities had its maximum at 3264.

Relevant figures for 1994 and 1995 are as follows:

	1994	1995	+/-
*fatalities	1,298	1,334	+ 2.8%
*injured	49,146	50,970	+ 3.7%
of which in hospital	11,735	11,688	- 0.4%

Another interesting set of data is the relation between fatalities/injuries and road categories:

On highwagens	: 13%
On 50 and/or 80 kmh roads	: 80%
Rest	: 7%

On a yearly basis ca. 2000 goods vehicles are involved in accidents; 60% of these vehicles are of the category "light duty vehicles" and this percentage is gradually increasing. Because of the roadtax-system in the Netherlands it is quite well possible that an increasing percentage of these cars are used not as a light duty vehicle, but as a sportive "off the road" vehicle belonging to the afore mentioned vehicle category!

Taking into account

- the more or less constant, still too great, number of fatalities and injuries in road traffic,
- changes in percentages of fatalities and injuries amongst different categories of road users, categories of vehicles or categories of roads,
- difficulties in spending an increasing percentage of available financial resources for research and measures on all possible aspects for improvements of road traffic problematics,

the Ministry of Transport, Public Works and Water Management of the Netherlands is developing a different approach. This approach should lead to a better cost/benefit ratio of the financial resources spent to solve the problems which are encountered because of the increasing traffic density in general.

Such problems are related to:

- Accessibility (of e.g. town centres), traffic jams.
- Sustainability (emissions).
- Safety (accidents, injuries, fatalities).

One of the causes in this quantity of problems is the vehicle and its related problems and possible solutions for these. A new approach for a vehicle related policy for the near future is now under development in order to improve cost/benefit rations of measures. This, however, does not mean that no attention at all is paid anymore to aspects falling outside these cost/benefit ratios.

NEW APPROACH

Instead of developing activities on whatever item related to possible improvements in road traffic safety the first thing to establish is definition of problem. From this definition of problem it should be made clear what the goals will be to solve the above mentioned problem, in other words: what do you want. To attain these goals it is necessary to have the right instruments: what can you do, and to have sufficient support: what is allowed to do. Resulting from these instruments and support a list of priorities can be defined. In order to follow such list of priorities it is necessary to define a working plan and a plan for the execution on one hand, on the other hand it is necessary to establish the need of knowledge, leading to needed research.

Concerning improvements in road traffic problematics the above mentioned structure is in the process of being elaborated and applied for the governmental policy concerning the vehicle. Several national studies have indicated the problems, for vehicles, being accessibility and sustainability. For accessibility important aspects are e.g. the use of the highway network and increase of transport efficiency for sustainability these are e.g. road traffic safety, gaseous emissions, noise emissions, transport of dangerous goods, use of energy, recycling of vehicle materials.

In this relation with sustainability the reduction of e.g. $CO2$ can be performed e.g. by vehicle engines with less energy consumption, by engine down sizing or by speed limiters. It can of course be reduced also by other means: driving behaviour, car price policy, $CO2$ storing, switching to transport by rail and/or water.

The general idea in vehicle policy is 3 times zero (zero congestion, zero fatalities and zero emissions). Specific goals for attaining this general idea as much as possible are needed and have to be determined. Preliminary research has been performed recently in order to establish cost/benefits of several measures which could be taken and which will have promising or nonpromising results on a short term or long term basis. External actors influencing the support for some of the measures are determined and their relation towards a positive or negative support. From this a list of priorities in possible activities can be drafted.

Results So Far

A survey of roughly 60 categories of vehicle related measures has been established which can lead to improvements in accessibility, safety and environment. An evaluation has been made by a group of experts about the chances of implementation of these 60 measures. The chances are based upon the possibilities for their realization and the time gap until realization can be effectuated: short or long term. With the help of two computer models (commercially available model "Expert Choice" and another model special developed for this) an analysis of the available information is made. From this analysis the most promising measures which result in good cost/benefit ratios have been determined as well for the short term, as for the long term.

The most promising measures for the <u>short term</u> are related to:

- Active control of vehicle features.
- Improvement of vehicle conspicuity.
- Front underrun protection and side guards for heavy goods vehicles.
- Improvements in the integration of car and public transport.

Somewhat less but still promising are:

- Optimalization of tyre technology.
- In-car improvements for vehicle control and driving behaviour.
- Improvements in safety aspects for interiors of buses.
- Vehicle identification.

The most promising measures for the <u>long term</u> are related to:

- Collision avoidance systems.
- In-car information systems.
- Intelligent speed limiters.
- Automated highway vehicle guiding systems.

Examples of measures with a <u>low score</u> in the computer analyses are:

- Electric vehicles.
- Improvements of stability of motorcycles.
- Periodical inspection.
- Electronic road pricing.
- Integration of car and bicycle.
- Hybride vehicles.
- Vehicles with fuelcells.
- Forewarning systems (for dangerous situations).
- Autonomic intelligent cruise control (car-following systems).
- Automated people movers.

It must be mentioned that these conclusions on most-not-promising measures and most-not-promising measures are preliminary. Further elaboration and refinements are to be carried out and will be done during next six months.

CONCLUSION

Activities in improving road traffic problematics can be done by several means. One of these is on the basis of a sound vehicle policy. Taking into account a 3 times zero policy, (zero congestion, zero fatalities and zero emissions) a logic order of steps has to be taken starting form determining the problem through setting the goals, instruments and support resulting in priorities and a plan of activities. In this manner future priorities can be listed which, independent of external influences, always result in a low cost/benefit ratio. Such an approach is considered to be necessary in times where financial resources are decreasing. Of course existing activities should not be stopped immediately, but should be considered carefully in terms of cost/benefits.

Sweden

Kåre Rumar
Swedish National Road Administration

ABSTRACT

Initially an account is given of the road accident and injury development in Sweden during the last years. The number of fatalities is steadily going down since 1990 while the injury and the accident situation does not look quite that favourable. In the Fall of 1994 SNRA, who is the responsible for the road safety situation in Sweden, published a new National Road Safety Programme which was very well received and is presently having large impact on the road safety work in the country. A key concept in the new programme is a result management system that governs the road safety work. The latest development of the programme is the 0-vision. This means that the road safety work should be guided by the same type of philosophy as e.g. flight safety or industrial safety - the goal is that nobody is killed in road accidents.

ACCIDENTS, INJURIES AND FATALITIES

In the 80:ies, when the number of persons killed in traffic each year was around 800, the Swedish Parliament declared that the number of injured and killed persons as well as the risk of being injured and killed in traffic should continuously be decreased for all groups of road users. The risk decrease should be larger for the unprotected road users and for the children. The authorities quantified these goals by stating that the number of persons killed in the year 2000 should not exceed 600 -- a decrease of about 25 percent. That goal was reached already in 1994 when 589 persons were killed in road traffic. The goal for injured persons was however not reached.

In 1995 the number of killed persons decreased further by 3 percent. The number of seriously injured persons decreased by 6 percent compared to 1994. The parliamentary goals from the 80:ies mentioned above were reached for all goals except the risk to be injured in road traffic, which is roughly constant. The improved road safety situation during the 90:ies is dramatic especially for children and young persons with a reduction of about 50 percent in the number of killed persons.

The main reasons for this reduction are hard to pinpoint. It is probably fair to say that a large part is due to external factors like the general economic recession during the first half of the 90:ies. This recession had many important effects on traffic. The main one from safety point of view is that the young drivers, who have large risks, have reduce their driving by about 35 percent during the last five years. Another effect of the recession is that traffic with heavy vehicles, which also constitute high risks for other road users, has been reduced. Other reduction factors are more effective police surveillance, safer vehicles, safer roads.

About 65 percent of the killed and the severely injured in road traffic are car occupants. The most severe accident types for passenger cars are single vehicle accidents, head on collisions and truck collisions. In spite of all ambitious Swedish efforts alcohol still plays an important role in fatal accidents especially in single vehicle accidents where over half of the killed persons are influenced by alcohol.

THE NEW ROAD SAFETY PROGRAMME

In 1994 the authority responsible for road safety in Sweden -- the Swedish National Road Administration (SNRA) -- together with the Swedish Police and the communities worked out a new National Road Safety Programme.

The road safety problems pointed out in that new programme are with one exception identical with the problems of earlier road safety programmes. The only new problem is however given highest priority:

The perceived value of increased road safety is too low among decision makers as well as the general public.

This low value has several negative effects on road safety:

- The public does not demand road safety measures.
- The decision makers do not allocate money to road safety measures.
- The road users do not use the potential road safety measures available (belts, helmets, speed restrictions etc).
- The road users have a low priority to defensive traffic behaviour.

Another new and important quality of the programme is a result based management system for the follow-up and control of the road safety work. By means of this system it is possible continuously to check whether the work has the safety effects intended and to modify the content or direction of work if the effects are unsatisfactory.

The result management system is based on a number of priority problem areas. For each one of these problem areas the present situation (value) and the target value for a specified year is specified. E.g the present usage of seat belts is totally about 85 percent, Target value for the year 2000 is 95 percent. Present value of drunken drivers in traffic is about 1 promille. Target value for the year 2000 is a reduction of 27 percent.

THE SWEDISH NON-FATALITY VISION

A third new important quality of the national road safety programme is a vision of the future road traffic system from safety point of view. A vision is a powerful instrument to create new values, initiatives, creative force, participation among people. Visions have for many years with good results been used by industry. If all actors in a sector share a common vision about the goal towards which everybody is striving it is not necessary to instruct everybody what to do. Each individual or organisation can use his own role, position, capacity, resources etc. to facilitate reaching the common goal.

In 1995 the so called 0-vision was presented. The motivation is that almost 5 percent of the total population is killed or disabled in road traffic accidents. Road accidents thereby constitute a considerable public health problem. The basic thoughts in the 0-vision are:

- Road traffic accidents should of course be avoided as far as possible. It is however not possible totally to avoid accidents.
- When we have accidents the violence against the human body should not exceed the biomechanical tolerances of the body.
- When we have injuries the rescue, treatment and rehabilitation should be as effective as possible.

The central idea in this vision is that authorities and individuals shall share the responsibility in the road safety work. The individual shall follow the rules agreed upon (speed limits, use of alcohol etc). But human errors will always occur anyway. When they occur the authorities are responsible for the system in the respect that the violence against the body will not exceed the human tolerances.

This non-fatality vision has received a very strong support in Sweden. National politicians as well as local decision makers and road safety workers have finally got a simple, easy to understand, easy to communicate and straight forward goal that unite them in their efforts. The goal is clear. The speed towards the goal is a political question.

ROAD SAFETY AND THE GOVERNMENT

As stated above the interest for road safety on the political level has increased during the last years. One of the reasons is that due to economic problems there is not enough money available for an extensive road building of e.g. motor ways. The road building programme in Sweden has been radically cut. It is then natural to look for other less expensive ways to increase road safety. The non-fatality vision described above indicates some other possibilities.

Another interesting change in the way the government treats road safety problems is that previously legislative measures were leading the road safety work (seat belts, helmets, speed limits etc). In the last years no specific road safety legislation has been implemented. Road safety work is to a large extent left to the market and the voluntary organisations. However it seems to be working fairly well anyway. As an example can be mentioned the very high penetration of air bags in new cars in Sweden during the last years.

The consumer driven market forces are powerful and comparatively quick. They are important complements to the regulative process. Therefore a scientifically based safety index for cars is an essential development that should receive more interest.

To improve the cooperation between the government and the automobile industry the government has created a vehicle research programme. The topics of primary interest within this programme are:

- Find and develope the road transport telematic functions that will have large positive impact on road safety.
- Develope intelligent protection systems for vehicles.

VEHICLE TOPICS OF CURRENT INTEREST

Neck injuries is a specific injury problem that so far has not received enough interest. Researchers, industry and consumers share this opinion. Between 10 and 15 percent of all traffic injuries are of this type. It leads to disability for about 2 percent of the population. It is characterized by being a low violence problem. It seems to be of primary interest to find test methods that lead to the best solutions.

Due to the increased interest in reducing the environmental impact of road traffic the government is again interested in downsizing cars. It is therefore important to establish a scientifically based method to stimulate a vehicle development that leads to an optimal balance between environment and safety. Economic incentive systems should be designed to stimulate a development in the desirable direction.

Presently the crash performance of cars and road side furniture (poles, barriers etc) are in large developed separately. Since the protection effectiveness is dependent on the interaction between the two sides it is in order to achieve a more optimal protection for the car occupants in crashes important that the crash performance is developed as a system. The two sides are very much dependent of each other.

Seat belt wearing is generally high in Sweden -- in passenger cars about 90 percent. However studies show that the wearing among drivers involved in crashes is much lower. About half of the drivers killed in crashes did not wear seat belts. It is therefore important to develope an intelligent interlock system.

The airbag still poses problems for the rear facing front positioned child seats. It is not enough to inform people about this problem by leaflets or stickers. This has to be solved in a technical way. ISO-fix, which should be developed for the rear facing front child seats, seems to offer a potential solution on this problem.

The non-fatality road safety vision not only shows the way for road safety work in general. It also offers a kind of a new paradigm for road safety research. Let us use it!

Poland

Wojciech Przybylski
Instytut Transportu Samochodowego

INTRODUCTION

This is for me the second time I have participated in the ESV Conference. I would like to make few remarks on a very important traffic safety factor, human behaviour. Two years ago I had the pleasure to inform you of our approach to vehicle construction features and connected legislation. This activity has progressed continuously, thus making our technical legislation closer to EU solutions. Now let me describe the main findings and activity in Poland concerning the matters related to human factors. First I would like to quote our accident statistics; in 1995 it appears that casual factor of road traffic accidents in Poland for driver behaviour reached 75%. Taking this figure into account it is clear that we have a great deal of work to be done in preparing our inhabitants to modern traffic conditions. The effective control of the problem is not easy taking into account that in Poland responsibility is divided between several authorities and organisations. From a theoretical point of view the most effective, due to the largest public, is legislative action, the other activities are either narrowly directed or temporary performed and thus less effective. Now I would like to describe shortly the Polish activity in the field of human behaviour.

LEGISLATIVE ACTION

I want to stress that not only road code is important for the Polish legislative action, education, law, and popularisation of traffic safety knowledge play a very important role in wide information to the public. Our road code is based on the 1968 Vienna Convention rules and thus has prescriptions uniform to the other European countries. Currently our Parliament is working on the final approval of new traffic rules. Two of the most important are:

- The lowering of speed in build up areas to 50 kph.
- Prolonging the period of obligatory use of daytime running lights from four to five months during the year.

With regard to educational action we published the requirements to conduct traffic training for children beginning with kindergarten and primary schools. There is no requirement for secondary schools. Drivers training is regulated by Ministry of Transport Ordinance and has full authority throughout local agencies.

Taking into account trends in our accident statistics it appears necessary to develop activities directed towards human behaviour, with the greatest emphasis on appropriate new scientific programs. The findings of these programs will be concluded in the next three years. We expect to receive some new tools for better control of the road safety. The following represent several activities which we believe will pay us back in the future.

EDUCATION

Our activity in the field of education concentrates on three main elements:

- Traffic education of children and youngsters.
- Driver training and post-exam improvements.
- Popularisation and informative activity.

Traffic Education in Kindergarten

Organised activities directed to small children started on a large scale in 1973. In the course of playing, taking walks, excursions and special lessons small children are informed of the traffic rules and necessity of being in compliance with these rules. The scope of education is limited to few main rules and information on most important road signs and signals with special emphasis given to the danger connected with playing in the road proximity.

Traffic Education in Primary and Secondary School

This activity also started in 1973, under the same legislation. Children in age 7 to 11 (I to IV standard) are taught the basic road traffic rules, and signs which should in practice be verified during exam after which the bicycle license is given to the child. From 1995 September the 1st the Minister of Education expanded traffic education for older group standards up to the VIII. The educational program is being prepared by Motor Transport Institute. The main idea of the enhanced program is to make children aware of the different roles played by different road users. Through simulation they are taught to understand each others' interests and advantages, from wide cooperation on the road. In addition to the school program there are different kinds of activities led by many organisations. The most spectacular appears to be:

- National contest on the Road Traffic Safety Knowledge.
- Police actions titled: "Safe Road to School" and "Safe Holidays".
- Different meetings and lectures for children and parents.
- Cooperation with different media (newspapers, radio and TV).

The last listed activity also concerns youngsters, but for this age group there is no official educational enforcement. In most secondary schools pupils could perform drivers training organised by school or on their own, after which they could enter for the drivers exam in one of officially accepted examinations centres.

Driver Training and Examination

Our driver training and examination program is based on the 1975 Geneva European Agreement (APC Agreement). We participate in the International Driving Tess Committee (CIECA). The exams have a theoretical part (choice test) and a practical part (manoeuvres and normal traffic). Since June 1993, Poland has started a drivers record program. Under this program the names of the drivers making road offences are listed together with their penalty score (from 1 point to 10 points depending on the severity of the offence). In the case where during one year the score exceeds 20 points the driver is obliged to pass further exam. The post-exam driver training currently is not very well developed, but it could be one of future promoted activity.

POPULARISATION AND INFORMATIVE ACTIVITY

This activity is performed by press, radio and TV, mostly on voluntary basis by journalists, thus sometimes many mistakes are made from theoretical points of view on road traffic safety. This action is, however, positively evaluated as a whole and concentrates on:

- Accident statistics (number of accidents and victims, real description of most spectacular accidents).
- Road traffic and weather condition information.
- Information on traffic contests, police actions, direction social events, new road safety equipment, drinking and driving problems and many other.
- Promotion of safety behaviour of drivers and pedestrians.

It appears important to find the tool which could be useful for evaluating this kind of action but it seems not to be found in near future.

SOME INFORMATION ON ACCIDENT TRENDS IN POLAND IN 1994-1995

The basic characteristics of our accident statistics has not been changed during last two years, however, some changes could be treated as more or less steady. According to analysis performed by the Centre of Road Traffic Safety of my Institute it appears that:

- The share of pedestrians killed and injured in accidents is falling down.
- The share of passenger car occupants killed or injured is rising.
- The relative rate of accident severity has fallen slightly.

The annexed figures show the latest analysis result.

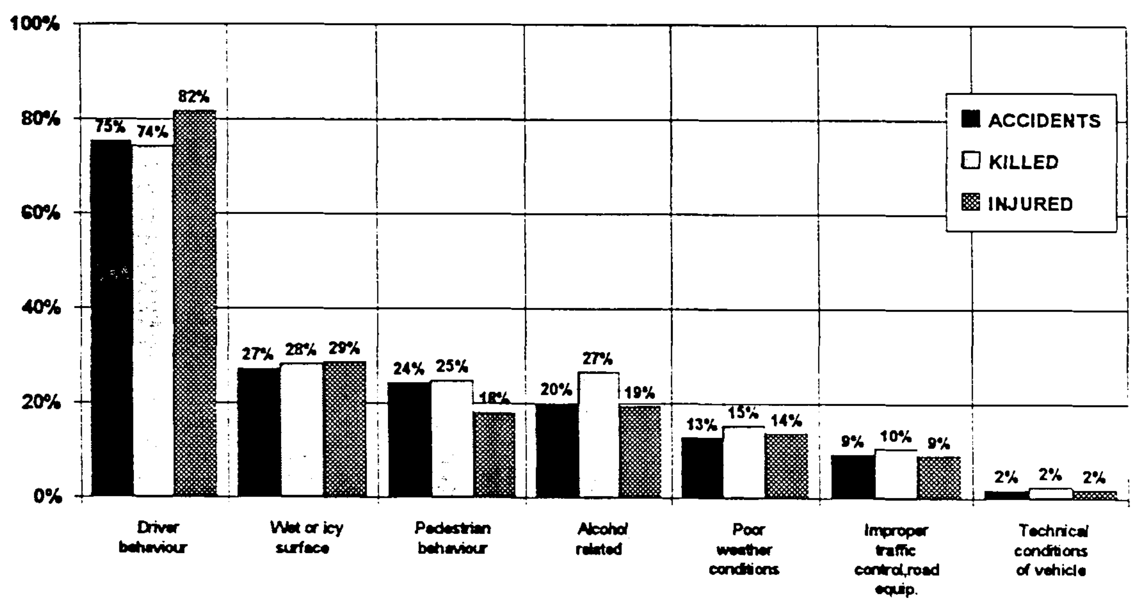


Figure 1. Causal Factors of Road Traffic Accidents in Poland in 1995

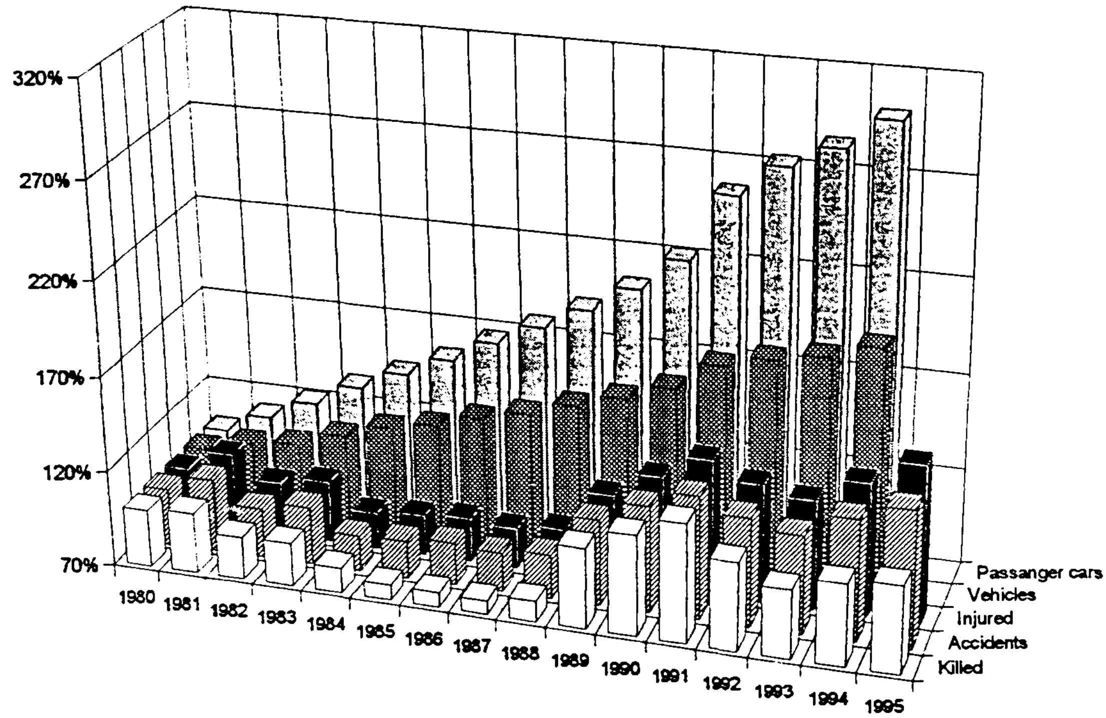


Figure 2. Trends in Road Traffic Accidents in Poland (1980=100%)

UNITED STATES

William A. Boehly (presented by Michael B. Brownlee)
National Highway Traffic Safety Administration

The National Highway Traffic Safety Administration (NHTSA) is challenged to initiate research and generate ideas to ensure that critical motor vehicle research continues well into the next century. It is a slow process that is critical to maintaining the pursuit of a safe and secure transportation system. The prospect of making transportation safer and saving lives through research is a formidable one, and one that merits our best efforts. The following pages report NHTSA's progress and achievements that have taken place toward these goals since the last ESV conference.

CRASH ENVIRONMENT

NHTSA's National Center for Statistics and Analysis reports that from 1992 to 1995, deaths on US highways increased from 39,250 to 41,700. This fatality increase is, in large measure, due to increased travel.

The fatality rate per 100 million vehicle miles travelled in 1971 was 4.5. By 1993, it fell to 1.7 and has remained at this level through 1995. Similarly, the fatality rate per 100,000 population decreased from 25.4 in 1971 to 15.62 in 1994.

Passenger car fatalities remained at 54 percent of the total in 1994, the same as for 1992, and continue to constitute the largest proportion of U.S. traffic fatalities. Light truck fatalities in 1994 were 21.8 percent of total fatalities, an increase from 20.6 percent in 1992. Motorcyclist fatalities in 1994 were 5.7 percent of total fatalities, a decline from 6.1 percent in 1992. Nonoccupant fatalities in 1994 were approximately 16 percent of total fatalities, down slightly from 16.2 percent in 1992.

The total number of police-reported crashes in the United States in 1994 was estimated by the National Accident Sampling System's (NASS) General Estimates System (GES) to be 6.492 million, an increase of 8.2 percent from 6 million in 1992.

DATA COLLECTION AND ANALYSIS

NHTSA's National Center for Statistics and Analysis has made several important changes to the agency's data collection programs which will benefit not only NHTSA, but the entire highway safety research community, nationally and internationally.

NHTSA's crash data collection system is composed of several components serving various needs. The Fatal Accident Reporting System (FARS) is a census of all fatal crashes occurring on public roads in the United States. NASS is a yearly collection of data from a statistically representative sample of crashes occurring in the United States. NASS is composed of a Crashworthiness Data System (CDS) and a General Estimates System (GES). These two primary crash data collection systems, FARS and NASS, are complemented by state crash data files compiled from police accident reports for a number of states.

FARS continues to collect occupant, vehicle and crash circumstance data on all fatal crashes. Since 1995, FARS data has been used to assess the effectiveness of numerous programs, including those that increase belt use and reduce drunk driving. FARS data is also used to evaluate the performance of various occupant restraint systems, including air bags. In 1996, FARS intends to make the most current data file available for analysis on the World Wide Web through the Internet.

In 1995, additional detailed information on air bag performance was added to the NASS CDS. Collection on precrash environmental data in support of the Intelligent Transportation System (ITS) was added to the NASS CDS and GES. The NASS CDS sampling plan was modified recently to increase the number of crashes involving air bag-equipped vehicles in support of the agency's evaluation of air bag system performance and regulatory initiatives. In the summer of 1994, a 3-year Pedestrian Crash Data Study was implemented in NASS CDS to update pedestrian injury patterns in impacts with late model year passenger vehicles. In the spring of 1996, a 2-year Unsafe Driver Actions Special Study will start in NASS CDS to identify specific action(s) taken by the driver of the vehicles that initiated the crash sequence and further identify the cause of that action.

NHTSA's State Data Program's objective is to obtain accident data files from select states and process those files into databases usable by NHTSA analysts. Each year, the accident data files from 17 states are obtained, documented, and converted into Statistical Analysis System (SAS) format. A specialized crashworthiness database is created from the raw state data files.

Through the State Data Program, NHTSA also encourages states to improve the usability of their data for highway safety analysis. One component of the State Data Program is the Critical Automated Data Reporting Elements for Highway Safety Analysis (CADRE) program. NHTSA initiated CADRE in 1990 to create national uniformity in a small number of variables that are essential to the use of police-reported accident data for highway traffic safety analyses. A CADRE assessment is being conducted by the National Association of Governor's Highway Safety Representatives (NAGHSR). The assessment will determine what CADRE data elements have been adopted, what elements have been dropped, and future plans for including CADRE data elements in the states' automated data files. The assessment will also examine the procedure(s) for adopting CADRE data elements and for including them in the automated file. The assessments will determine how CADRE data are used for problem identification of traffic safety issues and how these data are used to manage and evaluate implemented traffic safety activities. With the repeal of the national maximum speed limit compliance program, use of CADRE will very likely be of major importance in determining the impact of speed limit increases.

Another component of the State Data Program has NHTSA working with states to encourage the linking of medical outcome databases with crash data. Linking crash data to medical care and financial data provides a source of information not available through any of NHTSA's other databases. This program is the Crash Outcome Data Evaluation System (CODES). Through grants to seven states, occupant-specific statewide police-reported crash data were linked to injury data collected by emergency medical services, emergency departments, hospitals, and rehabilitation and long-term care centers to produce a database for the analysis of the benefits of safety belt and motorcycle helmet use. These data provided the basis for a report to Congress which showed that:

1.) Safety belts are highly effective in reducing morbidity (the occurrence of any injury) and mortality, and also cause a downward shift in the severity of injuries.
2.) The average inpatient charge for unbelted passenger vehicle drivers receiving inpatient care for a crash injury was more than 55 percent greater than the average charge for those that were belted, $13,937 and $9,004 respectively.
3.) Motorcycle helmet effectiveness ranged from 9 percent in preventing any kind of injury to 35 percent in preventing a fatality. The average

inpatient charge for motorcycle crash victims receiving inpatient care was $14,377 for those whose used helmets and $15,578 for those who did not, an 8 percent increase in charge for those electing to not wear a helmet.

With the recent emphasis on public health and cost containment issues, these linked databases provide important information on the injuries and costs of various components of the traffic crash environment. However, while linked data have the potential to provide needed information to the traffic safety community, obstacles to linking in most states remain and experience in utilizing these data is limited.

NHTSA's National Center for Statistics and Analysis produces a variety of reports, using data collected from FARS, NASS CDS and GES, and the State Data Program. These reports provide information on crashes at the national level and the characteristics of these crashes. For example, each year, a comprehensive compilation of motor vehicle crash data from FARS and NASS GES is published in Traffic Safety Facts, along with information on specific issues, e.g., alcohol, occupant protection, motorcycles, large trucks, pedestrians, pedalcyclists, school buses, older populations, young drivers, children, and speed. In addition to these annual publications, the National Center for Statistics and Analysis publishes the results of analytical work in various technical reports and research notes. Recent technical reports have addressed areas such as the effectiveness of antilock braking systems, estimating lives saved by restraint use, and evaluating the benefits of primary enforcement of safety belt use laws.

In addition to the annual publications and technical reports produced by NHTSA's National Center for Statistics and Analysis, almost 9,000 requests for statistical data were answered last year. These requests for information are received from governmental agencies, the general public, the U.S. Congress, citizen advocacy organizations, the automotive industry manufacturers, and international safety organizations. Meanwhile, as part of the NHTSA National Center for Statistics and Analysis' continuing focus on customer service, a wide range of traffic safety statistics and research information offered in our publications is now available on the Internet. By providing information via the Internet, NHTSA seeks to furnish the public and others with even greater access to important statistical information on traffic crashes in the United States.

CRASH AVOIDANCE RESEARCH

Intelligent Transportation Systems

Within DOT, NHTSA plays a leading role for the vehicle safety component of the departmental ITS program. The primary goal of the NHTSA program is to work with a variety of partners to facilitate and encourage the development and deployment of effective collision avoidance systems.

There is a narrow margin between driver responses that result in collisions and those that do not. Instances where a driver does not take appropriate collision avoidance action, are opportunities for intervention by driver augmentation systems. Such systems fill the gap between "actual" and "needed" action. To effectively prevent collisions, they must interact with the driver in a timely and effective way. They must also be designed to address specific collision circumstances. NHTSA is establishing the functional requirements for various collision avoidance safety systems in performance terms. This should foster the development and use of a wide array of technologies for reducing or compensating for driver errors and limitations.

The development of ITS collision avoidance countermeasures is a new science and one that requires innovative research tools and analytical techniques. These research tools are vital to understanding and documenting the safety benefits and potential liabilities associated with new countermeasures and to define requirements associated with their design and implementation. As part of fulfilling its role, NHTSA has initiated research and development of analysis tools necessary to evaluate crash avoidance concepts and is creating a knowledge base of driver-vehicle performance and behavior needed to support safety system development.

To facilitate product development and early deployment of ITS-based, safety-enhancing systems, the agency is supporting industry initiatives by working cooperatively to accelerate development. NHTSA is also working with a variety of partners to assess the performance, reliability, maintainability, failure modes/consequences, driver acceptance costs, and market readiness of promising systems under real-world operating conditions.

Since the Munich ESV Conference, NHTSA has moved forward in all aspects of the program. The details of this progress are provided in a companion paper at this conference. Thus, only some features of the progress are discussed here.

Perhaps the most significant step forward is the development of preliminary performance specifications for systems that address a number of collision types. While preliminary in nature, they represent a new era in the science of collision avoidance. These specifications are based on a very thorough analysis of accident data. This analysis provided a statistical base for problem size, as well as strong engineering insights into precrash maneuvers and vehicle dynamics. This analysis was followed by a thorough consideration of concepts for systems that could intervene during the moments between an event that would precipitate a collision and the collision itself, thus assisting the driver in avoiding a collision that would otherwise occur. The performance specifications identify important performance characteristics of the sensor, such as range and azimuthal field of view, and suggest values for each characteristic. They also include descriptions of algorithms for making the decision on the need for driver action. These algorithms are generally expressed as relationships between range and range-rate to other vehicles or lane edges. The specifications for the driver interface are quite general at this time and will need more study before full descriptions can be developed.

Several research tools are in the process of being developed. Two of these tools will provide the capability to gather data about driver behavior during normal driving. The other research tools are still in the process of being developed and will be available for use in the next two to three years.

Two other projects worth special mention are part of the NHTSA effort to work closely with the motor vehicle industry to move toward commercially viable collision avoidance systems. One of these projects is with a consortium led by Delco Electronics. The purpose of this project is to look at hurdles to the production of effective collision avoidance systems, such as the cost of building sensors, and take steps to overcome them. The second project is an operational test of an intelligent cruise control system (ICC). This is being done with a partnership led by the University of Michigan Transportation Research Institute. The system uses an infrared sensor developed by Leica. Results from this operational test will provide a detailed description of how drivers use this ICC and how effective it is in helping them drive more safely.

As we move forward toward availability and deployment of effective collision avoidance systems, it is important that all parties be able to assess the benefits of systems as they are being developed. NHTSA recently initiated an effort to develop methodologies and apply them to this task. The first step in this process was the creation of a task force within NHTSA to review all available test data from experiments where drivers had the opportunity to drive with, and without, the assistance of a collision avoidance system. The results of this review will be available in a few months. The work of the task force will be followed by an assessment of how to improve the process of gathering and analyzing data from test of concepts and prototypes to refine estimates of the benefits of collision avoidance countermeasures. These new techniques will then be incorporated into the overall NHTSA ITS program.

Human Factors

The human factors research program supports NHTSA's efforts to foster the development of "intelligent" collision avoidance systems, as well as the agency's ongoing programs aimed at more traditional safety improvement in driver-vehicle system performance.

This year will see the fabrication of one of the tools mentioned earlier, known as the Data Acquisition System for Crash Avoidance Research (DASCAR). This will be a valuable tool for the simultaneous measurement and recording of driver and vehicle behavior during long-term exposure to actual on-road driving experiences. A standardized protocol for collection and archiving of data collected will also be developed.

DASCAR simulator and on-road studies will measure drivers' responses to situations where a collision is imminent, and will investigate the effects of various driver feedback cues to determine why the anticipated benefits of antilock brake systems (ABS) for passenger cars are in some cases not being fully realized. The study will also consider the possible effects of risk compensation by drivers.

The investigation of possible adverse safety implications of in-vehicle use of cellular telephones and other in-vehicle systems will continue. As use of cellular phone and other in-vehicle devices increase dramatically, there is concern that this may lead to decreased attention to the driving tasks at hand. Studies will also look at the issue of driver workload in general, particularly as new ITS technologies come on-line.

Research to address the problem of drowsy drivers will continue with the development of systems to detect driver drowsiness by monitoring eye movement and/or lane tracking. These systems could provide warnings to drivers as their driving performances degrade.

Several programs will look at possible

countermeasures to address the problems of night vision and glare. These problems are particularly acute among the population of older drivers.

Heavy Vehicles

As ITS technologies emerge, there is an increasing need for better and more powerful communication and signaling between truck tractors and the trailers they tow. Two cooperative agreements are underway to investigate enhanced methods of powering and communication between tractors and trailers. The studies examine new methods such as multiplexing and the use of fiber optics, in addition to developing a standardized communications protocol.

A system is being developed to warn heavy tractor-trailer drivers as the vehicle approaches the rollover threshold for safe operation. This system could be linked to other systems such as an automatic, wheel-by-wheel braking system, or a roadway geometry information system to prevent rollovers in situations where a driver enters a tight curve too fast. A rearward amplification suppression system tied to electronic braking will also be developed to improve the safe cornering capability of multiple-trailer combinations by enhancing their lateral and roll stability.

Work will continue on investigating improvements to the occupant protection aspects of heavy truck cabs, and to determine the feasibility of modifications to the front-end structure of heavy trucks to reduce their aggressivity in their interaction with lighter vehicles.

National Advanced Driving Simulator

At the last ESV Conference, NHTSA reported two contracts had been awarded to two independent simulator development firms for competing preliminary engineering designs for the National Advanced Driving Simulator (NADS), and that following a thorough technical and cost evaluation of each design, it would select one contractor to proceed with the follow-on construction of the simulator. On February 15, 1996, NHTSA awarded a $34.1 million contract to TRW Transportation Systems of Sunnyvale, California, for the detailed design, fabrication, integration, test, and installation of the NADS. The NADS development team consists of TRW as the prime integration contractor, MTS Inc., of Eden Prairie, Minnesota, as the motion system subcontractor, Evans & Sutherland Inc. of Salt Lake City, Utah, as the visual system subcontractor, I*Sim Inc., also of Salt Lake City, Utah, as the audio system subcontractor, and Dynamics Research Inc., of Torrence, California, as the vehicle cab systems subcontractor. The NADS will be installed in a dedicated facility building which will be constructed in the Oakdale Research Park at the University of Iowa in Iowa City. The construction phase of the NADS development will require 39 months to complete, making the facility operational in the spring of 1999. The simulator will be used to conduct a wide-ranging research program in human factors, driver-vehicle interaction, accident causation and reconstruction, ITS development, medical and health conditions of elderly drivers, highway systems engineering, advanced vehicle-based safety systems, and young and novice driver training.

CRASHWORTHINESS RESEARCH

Frontal Crash Protection

Even after full implementation of driver and passenger air bags as required by FMVSS No. 208, it has been estimated that frontal impacts will account for up to 8,000 fatalities and 120,000 AIS $\geq$ 2 (i.e., moderate to critical) injuries in light vehicles.

A detailed definition of the remaining safety problems in frontal impacts is underway. Research will investigate real-world crash environments and project occupant injuries that will occur for an all air bag fleet. This includes summarizing the human loading and injury tolerances for occupants restrained by air bags in frontal crashes.

This program focuses on the intrusion-type injuries and fatalities and the costly lower extremity injuries observed in crashes involving air bag-equipped vehicles. We believe an offset frontal test best represents the real-world crashes that produce the intrusion-related injuries and fatalities, and the severe lower extremity injuries. We have been working to develop an offset frontal test, and we have been considering what new injury modes, injury criteria, and test surrogates might be necessary for this type of test.

Most recently, the offset crash testing has focused on a left oblique type impact for the target vehicle. The nominal test condition is a 30 degree left oblique impact with about 50 to 60 percent overlap on the target vehicle and a closing velocity of about 110 kmph (68 mph). A 50th-percentile Hybrid III dummy was used as the test surrogate in the driver's seating position. As reported at the last ESV Conference, an initial test in this configuration caused a mid-size vehicle to exceed the FMVSS No. 208 chest and femur criteria. Testing in this mode has continued in order to develop a more simplified, consistent procedure using a moving-deformable-barrier (MDB) as the bullet vehicle. Comparison testing has been conducted with car-to-car, MDB-to-car (both moving), and MDB-to-car (stationary car) to assess whether the latter test is an adequate substituted for the car-to-car test. Initial assessments indicate that most injury measures and structural responses are fairly similar in the three test configurations. In addition, a left oblique/offset test was conducted with more overlap (-80 percent) to determine whether the 50 to 60 percent overlap was also representative of impacts with near full engagement when the impact is oblique. The test results showed good agreement with both structural and injury responses. The future plan is to conduct a series of left oblique/offset, MDB-to-stationary car tests for evaluating current "fleet" performance and to compare with the type of test proposed for the future European frontal requirement.

Sled testing was also conducted using 5th- and 95th-percentile dummies, to determine the effects of occupant size on frontal crash protection. These tests evaluated the presence or absence of the lap/shoulder belt in conjunction with the air bag in the frontal crash tests.

Since the last ESV Conference, accident data on cars with occupants restrained by air bags have greatly increased. A review of these data indicate that the air bag appears to be doing the job for protecting the head and thoracic regions of the occupant. However, arm injuries appear to be more frequent for drivers with air bags. There are concerns, however, about including undesirable levels of neck loads and moments when the lap/shoulder belt is not used in conjunction with the air bag. Also, lower extremities appear to be significantly affected by the absence of lap/shoulder belt use. Although these types of injuries are generally non-life threatening, they can often be physically disabling and result in long-term care and high medical costs.

Recently, cases of injuries/fatalities have been reported which appear to be the result of aggressive air bag deployment. These injuries/fatalities appear to be mainly to out-of-position (OOP) occupants, and specifically, small stature and/or older drivers and child passengers, either unrestrained or in rear-facing child safety seats. The frontal program is addressing these problems through accident analyses of these cases to determine the magnitude of the problem and the type of injuries caused by deploying air bags, and by a test program to evaluate current and future air bag designs with OOP occupants. The test program has been initiated but with no reportable results to date.

Rollover Research

Nearly 9,000 fatalities and 100,000 injuries annually are attributable to rollover accidents. NHTSA is conducting a research program to improve rollover crash survival by preventing occupant ejection and by mitigating the severity of impact that an occupant experiences during rollover. In the area of occupant impact mitigation, NHTSA is investigating improved interior padding, improved restraints, and improved roof support. Research in ejection prevention is centered on improving door latches and ejection resistant glazing.

Occupant ejection through side doors accounts for several thousand fatalities and many more serious injuries each year. A test procedure is being developed to evaluate the potential for door openings caused by occupant impact. A static FMVSS No. 214 impactor is being applied to the interior side of the door to evaluate the potential for linkage activation. A research program to develop a secondary latch as a countermeasure has been completed. Tests of a preliminary design have demonstrated that an auxiliary latch can augment the existing latch and provide significant strength increases when tested using longitudinal and lateral full door procedures.

The improved glazing research program is developing a test to reduce ejection through side glazing. Alternative glazings and their mounting systems are being tested in experimental crash situations to assess their performance in ejection reduction and occupant safety. Crash data are being used in the test development. The capacity for hazing and scratching, and their public acceptance, is also being evaluated.

Analytical models are being developed to increase the understanding of the injury potential to occupants during rollovers. These models are being used to simulate crash tests and real-world crashes to better understand injury mechanisms. The models also are being used to explore the feasibility of potential rollover countermeasures.

Vehicle Aggressivity and Fleet Compatibility

A research program for vehicle aggressivity and compatibility has been initiated since the last ESV Conference, as well as first efforts to investigate the problem of vehicle aggressivity in two-vehicle traffic accidents. An examination of US accident statistics shows a striking incompatibility between light truck and van (LTVs) and passenger car crash performance. In using a preliminary aggressivity metric described in a paper presented at this conference, LTVs as a class have been observed to be twice as aggressive as passenger cars. This mismatch in crash performance has serious consequences for the traffic safety environment as approximately half of all passenger vehicles sold in the United States are LTVs. The effect of this tremendous degree of fleet incompatibility is not measured directly by frontal-barrier crash tests and will be the focus of future NHTSA research.

Side Impact Research

Side impact accidents of light vehicles, i.e., passenger cars and LTVs result in around 10,400 fatalities and over 603,000 injuries each year. This corresponds to about 30 percent of vehicles involved in tow away crashes. Since the last ESV Conference, the dynamic FMVSS No. 214, Side Impact Protection, which establishes minimum requirements for thoracic and pelvic protection for the near-side occupant in side impact crashes of passenger cars, has been extended to include LTVs. This will provide LTV occupants the same minimum level of protection as passenger cars in intersection type crashes. With near term side thorax and pelvic impact research concluding, work is continuing in defining the remaining side impact safety problem in analysis and in testing.

Accident analyses have been initiated to establish a better understanding of the side impact conditions of both single and multiple vehicle accidents. Results to date indicate that side impacts into fixed objects result in around 20 percent of all side impact fatalities. Over 80 percent of narrow objects impacted are trees and fixed poles. As to occupant injuries, the head and neck constitute 46 percent of most severely injured body regions, while the upper extremities constitute around 24 percent.

Based on the accident analyses, a joint program with the Federal Highway Administration to explore side impact protection of light vehicles in impacts with narrow objects is underway. Two passenger car into rigid pole tests have been performed to assess test procedures, test conditions, and dummy performance. This test data is also being used in finite element model validation studies. The impacts were midway through the driver side door at 32.2 kmph (20 mph) as indicated by the general accident analysis. A side impact dummy (SID) retrofitted with a Hybrid III head and neck assembly was used for one of the tests for improved biofidelic response.

Advanced structural modeling is ongoing to evaluate intrusion and dummy loading in side crashes, and to assess potential occupant protection countermeasures. A finite element model of a passenger car has been developed and validated in an FMVSS No. 214 impact configuration. The model is being exercised in vehicle into vehicle and vehicle into rigid pole side impacts. A finite element model of the SID has also been developed. An advanced rigid body passenger car model with the dummy has also been developed for use in parametric studies and in support of crash testing.

Biomechanics Research

The objectives of NHTSA's Biomechanics Program remain (1) the advancement of the scientific understanding of the mechanistic process that produces impact trauma, and (2) the translation of that understanding into analytical and physical testing capabilities that can guide and evaluate the safety concepts designed to prevent and mitigate impact injuries. These objectives are accomplished by pursuing efforts in four distinct areas of concentration. These efforts and their purposes are:

- Highway Traffic Injury Studies - Conduct detailed medical and engineering examinations of crashes, documenting the resulting injuries, treatment, costs, and consequences of the crashes, and applying this data to high priority safety issues.
- Human Injury Simulation and Analysis - Develop and apply mathematic modeling techniques to predict the extent and severity of human injuries in differing crash circumstances.
- Impact Injury Research - Conduct laboratory impacts to human surrogates and quantify the forces, motions, and the extent and severity of injuries to the human body.
- Crash Test Dummy Component Development - Develop test equipment and dummy components to measure crash forces and evaluate trauma risk in automotive crashes.

NHTSA's Highway Traffic Injury Studies continue to yield important results that are contributing to a better understanding of injuries and how we can prevent them. Our studies have shown that the success of the air bag in preventing deaths produces survivors with severe injuries to the pelvis and lower extremities. To promote increased interest in research on this topic, NHTSA organized and held a conference on Pelvic and Lower Extremity Injuries in Washington, DC, in December of 1995. This drew substantial participation from government, academia, and the private sector. The conference brought together a multidisciplinary group of physicians, engineers, and researchers from the United States and other countries, including the United Kingdom, Sweden, France, Germany, The Netherlands, Italy, Japan, and Australia. The conference was extremely successful in giving participants access to the growing body of research in this area, in identifying gaps and needs for future research efforts, and in developing alliances to accelerate research activities.

Another important new initiative is the development of a Trauma Data network that seeks to link NHTSA's trauma center-based researchers and their important data. This network will enable the collection and storage of their crash and medical data on a universally accessible data system. A researcher will be able to enter data into the system and have it immediately accessible to fellow researchers in other centers of the network. We expect that this will lead to much earlier identification of trends in injury patterns and, therefore, quicker design and development of preventative strategies and actions. Currently, four centers are participating in the development of the network. They are the University of Medicine and Dentistry of New Jersey (UMDNJ) in Newark, New Jersey, the National Study Center for Trauma and EMS of the University of Maryland at Baltimore, in Baltimore, Maryland, the Children's National Medical Center in Washington, DC, and Jackson Memorial Hospital in Miami, Florida. They will be joined shortly by the San Diego County Emergency Medical System, Harbor View Hospital, in Seattle, Washington, and the University of Michigan Medical Center, Anne Arbor, Michigan.

Mathematical simulation efforts have progressed in several anatomical regions. In the head, an analytical algorithm has been developed and implemented into the previously developed skull/brain model that can monitor the brain's response to an impact and determine the amount and location of brain material that experience excessive and injurious deformations.

Thoracic modeling efforts have increased the realism of the model by incorporating shoulder structures to more accurately distribute loads from restraints to the torso. This model is currently being used to investigate the efficacy of force limiting torso belts used in combination with air bag restraint systems.

The anatomic neck model has also been improved by adding the shoulder and first rib from the thorax model. This has enabled the inclusion of musculature into the model to better represent a living human subject.

Lower extremity models are also being developed to study the injury mechanisms associated with both impulsive loading and occupant compartment intrusion. One model will focus on knee and hip injuries caused by contact with the instrument panel, while the other model will provide a detailed analysis of ankle injuries resulting from impulsive toe pan loading and intrusion.

Considerable effort continues to be devoted to Impact Injury Research. One study investigated human cadaver and Hybrid III dummy sled tests conducted at the University of Heidelberg. The investigation asked if it is possible to obtain both the thoracic injury mitigating benefits of an air bag only restraint and the all-impact-direction benefits of the belt from a combination restraint system. The combination system adds a force limiter to the shoulder belt. The results suggested that when one combines a driver air bag with a 3-point belt system that limits the torso belt loop to 4 kN, one obtains additional injury mitigation benefits. This approach of combining force limited torso belts with air bag systems is now being applied by several automotive manufacturers.

Twenty-three human cadaver ankle impacts were performed by the Medical College of Wisconsin. These experiments axially loaded the foot/ankle. An ankle injury risk function using axial load was developed on the basis of these experiments. This function suggests that there is a 50 percent chance of an ankle/foot injury when the tibia experiences an axial force of 6 kN or greater. The response of the Hybrid III dummy ankle/foot was examined under the same experimental conditions and showed that the axial force measured in the Hybrid III dummy can be linked with the risk of function to assess the potential for foot/ankle trauma. Efforts at developing our Advanced Lower Extremity (ALEX) continue and are expected to provide a new design with both improved realism and accuracy to evaluate lower extremity injuries in crash situation.

Another study that crosses several disciplines is our investigation of air bag induced upper extremity trauma. Based on 1988 - 1994 NASS files, the study found, for maximum injury to the arm, that a driver restrained by a belt/air bag system is roughly four times more likely to have an AIS ≥ 2 injury than a driver restrained by only a belt. It appears there are two primary mechanisms of injury arising from the arm interacting with the air bag. The first occurs when the arm is in very close proximity to the module when ignition occurs. The rapid expansion of the module and bag induces high bending movements in the arm causing fractures. The second mechanism occurs when the arm is close but not in the immediate proximity to the module. Here the bag flings the arm at high velocities into various interior components causing hand or wrist injuries. Based on consultation with the Society of American Engineers

(SAE) and agency experts, a research arm injury device (RAID) has been built. The RAID is a "flesh"-covered aluminum tube with a universal joint-type pivot. The RAID is instrumented in five axial locations to indicate bending and with additional instrumentation to capture fling velocities. To assess air bags in an experimental setup, the RAID is placed over a driver air bag and the bag is deployed with the moment and velocities recorded. Current experience indicates substantial performance differences among various module designs. Efforts are now underway that will pursue appropriate biomechanical fracture thresholds to use in conjunction with the RAID to determine if and when injury would occur.

NHTSA has been very active in the two years since the last ESV Conference in furthering the development of a new advanced frontal crash test dummy. The TAD-50M torso design, previously developed and presented, has undergone extensive evaluation, in cooperative efforts between NHTSA, its development contractor, and numerous worldwide partners. Based upon these results, a significant upgrade of the TAD-50M design has now been completed. These efforts have developed many new components, including a new face, neck, thorax, abdomen, spine, pelvis, and lower extremity designs, along with the attendant expanded and improved instrumentation complement to detect injurious impact conditions. A prototype of this new advanced frontal dummy is on display at this conference.

In hopes that this device can become the basis of harmonizing frontal impact protection standards worldwide, it has been a NHTSA priority during the development of the advanced frontal dummy to consult with potential users and expert groups on an international basis. Close liaison has been established and maintained with the SAE in the United States, European Experimental Vehicle Committee (EEVC) Working Group 12 in Europe, and Japan Automotive Research Institute (JARI) and Japan Automobile Manufacturers Association (JAMA) in Japan. The support and cooperation of these groups have been invaluable, and these cooperative efforts will continue as the dummy program moves on to its test and evaluation phases.

Emergency Medical Care -- Automatic Life Saving Project

Effective emergency medical care provision to victims of vehicle crashes depends on the timely notification of emergency care providers of the crash, the dispatch of the appropriate level of response directed toward the scene of the crash, and the transportation of injured occupants at the scene to the appropriate level of medical care. NHTSA's Automatic Life Saving project is focused on identifying ways to improve emergency medical care for crash victims. It is expected that the results of this research will add to first responder's, emergency medical technician's, nurse's, and physician's knowledge base on injury patterns and assist them in their treatment of persons injured in motor vehicle crashes.

The project will focus on identifying life saving strategies for the provision of emergency medical services based on vehicle dynamics, vehicle damage, scene medical status, and injuries in a crash event. The Automatic Life Saving project will combine the efforts of a number of projects being conducted within the Crashworthiness and Crash Avoidance research areas. Our Automatic Collision Notification (ACN) system project is aimed at developing and proving the concept of automatic crash notification and emergency medical response to the scene of a crash. As part of the project, equipment will be installed in 1,000 passenger cars to allow for automated in-vehicle collision sensing and emergency message transmission.

A key element within the Automatic Life Saving project is the development of vehicle based trauma algorithms that will notify emergency dispatch personnel of the severity of the crash and the possible injury consequences. The algorithms are being developed as part of a multidisciplinary effort by emergency medical care experts and engineers in vehicle crash dynamics and·accident reconstruction, and will relate occupant injury with specific crash event parameters. These algorithms will be developed using parameters that are associated with the dynamic conditions during a crash and also with parameters that can only be obtained at the crash scene.

Highway Traffic Injury Studies

NHTSA's studies at four trauma centers have continued to advance the science of crash injury research. The contributions include the following:

- Discovery of a new injury pattern of potentially fatal liver injuries in people protected by a two-point belt, but not wearing the lap belt, in frontal crashes.
- Development of the "Look Beyond the Obvious" poster to increase the recognition of "occult" internal injuries and improve nationwide practices of triage, diagnosis, and treatment of people injured in crashes.
- Improved understanding of injuries in crashes with people protected by air bags.

- Developed new multimedia computer software that advanced the state-of-the-art of injury research.

As more and more motorists are protected in crashes by belts and/or air bags, the need for "occult" injury detection is increasing. This research helps prevent people who have been protected by restraint systems from needless death or disability from missed internal injuries because traditional "tell tale" signs of crash avoidance are not apparent to crash responders.

These contributions were made possible by the synergy of medical and engineering researchers in government, industry, and academe working cooperatively together to make advances in injury prevention and treatment.

RULEMAKING

Crashworthiness

- On July 21, 1994, a Final Rule was published to amend FMVSS No. 207, "Seating Systems," to modify the test procedure for pedestal seats. The new procedure applies two separate loads simultaneously: one through the pedestal and one through the seat. The rule became effective October 19, 1994.
- On August 3, 1994, a Final Rule was published amending FMVSS No. 208, "Occupant Crash Protection," to require adjustability of safety belt systems on light vehicles. The requirement applied to all Type 2 safety belts installed at adjustable seats. The purpose of the requirement is to improve the fit and increase the comfort of the belt for a variety of different size occupants. Ways of complying with the requirement include adjustable upper anchorages, integrated seats, and inboard moveable anchorages. This requirement will be effective September 1, 1997.
- On February 8, 1995, a Final Rule was published amending FMVSS No. 213, "Child Restraint Systems," to improve the language of the label on rear-facing child restraints that warns against placing such seats in front of a passenger side air bag. This rule was effective May 9, 1995.
- On March 14, 1995, a notice of proposed rulemaking (NPRM) was published to amend FMVSS No. 205, "Glazing materials," to permit rigid plastic glazing on the side of windows of hatchbacks and station wagons, behind C-pillars. This limited application of rigid plastic window glazing in automobiles, for the first time, would be in an area away from vehicle components.
- On May 9, 1995, the agency published a Final Rule amending FMVSS No. 217, "Bus Emergency Exits and Window Retention and Release." These amendments permitted manufacturers to install two emergency exit windows as an alternative to an emergency exit door on school buses. The rule also permitted manufacturers of buses to meet either the non-school bus emergency exit requirements or the recently amended school bus requirements. The rule modified the requirement specifying the number of additional emergency exits that are required on school buses of varying capacity. These amendments will provide increased clarity and ensure that manufacturers meet the recently upgraded school bus requirements by providing additional exits rather than by increasing the size of the existing exits. The rule also removed the design restrictive language in the standard and allowed the use of sliding emergency exit windows on all buses. These amendments will increase manufacturers flexibility in meeting emergency exit requirements while maintaining the existing level of safety. The effective date for these amendments was May 9, 1996.
- On May 10, 1995, a final Rule was published that amended the language in FMVSS No. 213, "Child Restraint Systems," and that modified FMVSS No. 213 to permit the manufacture and sale of belt-positioning booster seats. This rule became effective on August 8, 1995.
- On May 23, 1995, a Final Rule was published amending FMVSS No. 208, "Occupant Crash Protection," to allow manufacturers the option of installing a cut-off switch to suppress the operation of the passenger side air bag in certain vehicles. The switch is permitted only for vehicles that have no rear seats or have rear seats that are too small to accommodate a rear facing infant restraint. The switch must be accompanied by an illuminated yellow warning light that warns the occupants of the air bag suppression and must be operable using the vehicle's ignition key. The option expires on September 1, 1997 for passenger cars and September 1, 1998 for light trucks.
- On June 9, 1995, a NPRM was published proposing to amend FMVSS No. 213, "Child Restraint System," to no longer permit backless child booster seats, vest type and harness type child restraint devices to be certified for use in aircraft. This proposal was a companion notice to a Federal Aviation Administration NPRM, which proposed not to allow the use of such child

restraints in aircraft.

- On June 13, 1995, the agency published a NPRM to amend FMVSS No. 208, "Occupant Crash Protection," by allowing design flexibility of the safety belt systems in the rear seats of police vehicles.
- On July 6, 1995, a Final Rule was published to amend FMVSS No. 213, "Child Restraint Systems," to add a new array of test dummies and new compliance test procedures that are dependent upon the heights and weights of children for which the child restraint is designed. This rule became effective January 1, 1996.
- On July 28, 1995, the agency published a Final Rule extending the side impact protection requirements of FMVSS No. 214, "Side Door Strength," for passenger cars to trucks, buses, and multipurpose passenger vehicles with a GVWR of 6,000 pounds or less. The effective date for this rule is September 1, 1998.
- On August 7, 1995, the agency held a public meeting to discuss possible upgrades to FMVSS No. 206, "Door Locks and Door Retention Components." At the meeting, the agency presented: accident data analyses to identify the safety problem; the results of research using four new test procedures developed to simulate real world latch failures; and three possible options for upgrading FMVSS No. 206.
- On August 18, 1995, the agency published a Final Rule to amend FMVSS No. 201, "Occupant Protection in Interior Impact," by requiring passenger cars, trucks, buses, and multipurpose passenger vehicles with a GVWR of 10,000 pounds or less to provide improved head protection when an occupant's head strikes upper interior components. This rule became effective on September 18, 1995.
- On September 28, 1995, the agency published a Final Rule extending and expanding the requirements of FMVSS No. 206, "Door Locks and Door Retention Components," for side doors to back doors of passenger cars and multipurpose passenger vehicles with a GVWR of 10,000 pounds or less. The effective date for this rule is September 1, 1997.
- On November 9, 1995, the agency published a notice requesting comments on potential solutions to the undesired effects of air bags. The notice stated that the agency had become aware the current air bag designs appear to have contributed to serious injuries and even death to vehicle occupants under certain circumstances. Over 58 comments were received from the automotive industries, air bag suppliers, the medical profession and other interested parties. Based on the initial responses, NHTSA undertook a research program to assess different air bag systems and their affect on out-of-position occupants, particularly the small child and infant.
- On November 16, 1995, the agency published an NPRM to exclude vehicles from FMVSS No. 204, "Steering Control Rearward Displacement," if the vehicles were certified to comply with the agency's occupant crash protection standard, by means of an air bag. As part of the President's Regulatory Reform, the agency identified this as a potentially redundant certification requirement. The agency believes the design consideration for an adequate air bag platform are the same as the design considerations for steering column rearward displacement.
- On March 7, 1996, the agency published an Advance Notice of Proposed Rulemaking which granted four petitions to commence rulemaking to amend upper interior head protection requirements of FMVSS No. 201, "Occupant Protection in Interior Impact," to accommodate vehicles equipped with a dynamic head protection device that is activated in a side impact (e.g. a side air bag).

Crash Avoidance

- On November 2, 1994, an amendment to FMVSS No. 108, "Lamps, Reflective Devices and Associated Equipment," was published that established an objective requirement for exposure resistance of plastic materials used for reflex reflectors and lenses covering reflectors. After 3 years of exposure to the elements in Florida and Arizona, samples of such plastics must exhibit no more than 7% haze. This criterion replaced a subjective criterion that test samples not "show" haze. This rule was effective on November 2, 1995.
- On February 2, 1995, a Final Rule was published to establish Federal Motor Vehicle Safety Standard No. 135, "Passenger Car Brake Systems." This standard will replace FMVSS No. 105, "Hydraulic Brake Systems," as it applies to passenger cars. Standard No. 135 will achieve the goal of international harmonization. This rule was effective on March 6, 1995.
- On March 10, 1995, a Final Rule was published to reinstate stopping distance requirements in FMVSS No. 121, "Air Brake Systems." The amendments would require vehicles to stop in a specified distance from 32.2 to 96.56 kmph (20

to 60 mph) when tested on a surface with a peak friction coefficient (PFC) of 0.9; loaded truck tractors would be tested with an unbraked control trailer. This requirement will be effective on March 1, 1997.

- On March 10, 1995, a Final Rule was published which extended the stopping distance requirements in FMVSS No. 105, "Hydraulic Brake Systems," to medium and heavy vehicles with a gross vehicle weight rating (GVWR) greater than 10,000 pounds. The amendments will require vehicles to stop in a specified distance from 32.2 to 96.56 kmph (20 to 60 mph) when tested on a surface with a PFC of 0.9. These rules are part of a comprehensive effort by the agency to improve the braking performance of heavy vehicles. The requirements will be effective on March 1, 1999.
- On April 20, 1995, a Final Rule was published amending FMVSS No. 108, "Lamps, Reflective Devices and Associated Equipment," to allow the photometric performance of rear center high-mounted stop lamps to be determined on the basis of a grouping of test points, rather than by absolute performance at each point. This action was consistent with the requirements for other lamps and reduced the testing burden for manufacturers. This rule was effective May 22, 1995.
- On June 12, 1995, a NPRM was published to amend FMVSS No. 108, "Lamps, Reflective Devices and Associated Equipment," to require a pattern of retro reflective material to be used on the rear of truck tractors. This conspicuity treatment would be similar to the retro reflective material required on new heavy trailers, and it would increase the night visibility of truck tractors traveling without trailers in tow.
- On June 19, 1995, a NPRM was published proposing to amend FMVSS No. 108, "Lamps, Reflective Devices and Associated Equipment," to allow high intensity discharge (HID) light sources to be used in replaceable bulb headlamp systems in addition to their presently allowed use in integral beam headlamp systems.
- On September 5, 1995, a Final Rule was published, which amended FMVSS No. 108, "Lamps, Reflective Devices and Associated Equipment," to update the test specification processing requirements of plastic material used for optical parts such as lenses and reflectors. It allows manufacturers more discretion in the size and manufacturing methods of plastic test samples used in outdoor exposure durability tests. This rule became effective on March 1, 1996.
- On November 25, 1995, an amendment to FMVSS No. 108, "Lamps, Reflective Devices and Associated Equipment," was published allowing replaceable bulb headlamps with on-board aiming devices to be manufactured with replaceable lenses. It is intended to reduce the cost of repairing headlamps with cracked or punctured lenses, and it established more rigorous anti-corrosion requirements for the reflectors of such headlamps to prevent deterioration prior to lens replacement. This rule became effective on December 26, 1995.
- On November 28, 1995, a Final Rule was published amending FMVSS No. 108, "Lamps, Reflective Devices and Associated Equipment," to allow the agency to transfer replaceable bulb headlamp dimensional and specification information housed in Standard No. 108 to the Part 564 docket. This rule was effective January 29, 1996.
- On February 1, 1996, the agency held a public meeting to discuss its findings to date on advanced glazing materials that may prevent ejection of vehicle occupants through motor vehicle windows during crashes.
- On February 15, 1996, and December 13, 1995, the agency amended the ABS Final Rules in response to petitions for reconsideration. The amended rule now require continuous power for trailer ABS systems in place of dedicated power and a separate ground, which were previously required. It delays the implementation date for the in-cab trailer malfunction indicator by four years and extends the period in which exterior ABS failure indicators are required by three years.
- On February 21, 1996, a NPRM was published proposing to amend Standard No. 108, "Lamps, Reflective Devices and Associated Equipment," to adopt new photometric requirements for motorcycle headlamps and improve the objectivity of the aiming of the upper beam. The new photometric requirements would be those of SAE Standard J584 OCT93. They would be an alternative to the current photometric requirements of SAE J584 April 1964, initially, and would become mandatory between two and four years after issuance of the final rule. For tests of photometric performance, the upper beam of motorcycle headlamps would be aimed photoelectrically rather than visually, as at present.
- On March 21, 1996, a Final Rule was published rescinding FMVSS No. 107, a standard that

regulated the reflectivity of specified metallic components located in front of the driver. The action was part of the President's Regulatory Reinvention Initiative. The effective date for this rule was May 6, 1996.

SUMMARY

Lifesaving progress has been made because of safety improvements. It is estimated that over 65,00 lives were saved by the use of safety belts from 1982 through 1994. Since 1982, more than 2,600 children's lives have been saved by using child restraints. We also estimate that the minimum drinking age law of 21 years old has saved almost 15,000 lives since 1975. Air bags are beginning to show substantial benefits as the number of air bag-equipped vehicles in the fleet continues to rise. Between 1988 and 1995, an estimated 1,500 lives were saved.

AUSTRALIA

Peter Makeham
Federal Office of Road Safety

ABSTRACT

This paper reviews Australia's recent road safety record within the context of a nationally integrated strategic policy approach developed and agreed to by Federal/Sate/Territory road safety authorities, local government, industry, and community groups.

The *National Road Safety Strategy* outlines the broad road safety policy framework that has been developed and the range of tools employed in Australia to achieve our recent overall successes. The paper foreshadows that Australia will face even greater challenges in the future, and that there is a dynamic strategic platform in place to meet these challenges.

Road safety and trauma is a major problem to Australia, as it does to many other governments and communities worldwide. After a number of years of reduction in the national road toll, the number of road fatalities increased in 1995.

The current approach to road safety in Australia is based upon partnerships and flexibility of approach within a broad framework. The role of vehicle safety standards and design within that framework is also discussed.

This paper sets out the context in which Australia expects to manage road trauma. It recognizes the clear need for a multidisciplinary approach that harnesses the energy of the "stakeholders" at all levels, and that mobilises integrated and coherent actions.

Vehicle safety, which has a strong international dimension, plays an important, integral and ongoing part in the scenario for reducing road trauma. As vehicle and road design standards continue to improve, so does the corresponding need to progressively introduce other road safety measures based on strategic planning, community participation, and adoption of best practice.

AUSTRALIA'S EXPERIENCE

The number of people killed on Australian roads fell by 49% from a peak of 3,798 in 1970 to 1,940 in 1994. The 1994 road toll was the lowest since 1954. In addition to seat belts and random breath testing, other significant features in reducing the road toll over this period included continuing improvements in vehicle design and improved roads, uniform drink drive regulations (0.05 BAC for fully licensed drivers, 0.02 BAC for young drivers and heavy vehicle drivers), and compulsory wearing of bicycle helmets.

The 49 percent decrease in fatalities from 1970-94 is even more impressive in view of growth in the population and the number of registered vehicles over the period. The population rose by 40 percent from 12.7 million in 1970 to 17.8 million in 1994 and the number of registered vehicles rose by 118 percent, from 4.9 million in 1970 to 10.7 million.

In particular from 1989 to 1994, Australia experienced a major reduction in the level of trauma on the roads. The number of people killed fell from around 2,900 to below 2,000. Similar reductions have also been achieved through the introduction of an integrated program of action by federal and state governments. It comprised increased enforcement, increased public education, regulatory changes, road "blackspot" remedial programs, and improvements in vehicle design standards.

The level of police enforcement was increased dramatically and was accompanied by intense public media campaigns in some states.

As a consequence of these initiatives, the road toll was maintained below 2,000 for three years despite increased economic activity in 1993 and 1994.

In terms of the rate of fatality per 100,000 or 10,000 registered vehicles, Australia has shown one of the greatest rates of improvement of any OECD nation. Australia now is among the safest places to drive in the world. However, the decrease in the fatality rate evident since 1989 was reversed in 1995 as the road toll rose slightly above the 2,000 level.

Australia's short term objective is to get the road toll below 1,900 deaths annually by the end of the century. This target was achieved in 1993, when the fatalities for the preceding twelve months remained at 1,900 or below for six consecutive months.

In the immediate future, the scop for achieving our objective on an on-going bases, will depend on the extent to which those Australian States and Territories with world class performance maintaining their efforts, and others improving theirs. In the long term, the development of strategic approaches to road safety and the arrival of significant new technologies have the potential to ensure that Australian roads will continue to be among the safest sin the world.

In terms of vehicle safety standards, Australia has taken decisions to adopt the leading international standards which apply to both passenger vehicles, trucks and buses. Australia seeks to harmonise its regulations with international standards and is playing a central role in this area.

ROAD SAFETY PARTNERSHIPS AND A MULTISECTORAL APPROACH

The question which faced the Australian road safety community was how best to lock in the gains of the past and plan for further reductions. It became evident that a structured approach was needed aimed at meeting national objectives. There had to be a national ethos or a national commitment.

Australia was attracted to strategic planning efforts being developed by some overseas countries. There was recognition that the authoritative, top-down approach adopted in the past had not always worked. The recent strategic approach is cooperative and intersectoral, with bottom-up "grass roots" community involvement in the development of strategies and actions.

The strategic model can take advantage of the changing political, economic and social environments to provide new linkages, challenges and opportunities to share experiences from which best practices can be drawn. The underlying philosophical change which has gained support in recent years is that road safety has to be addressed as a multi-sectoral issue.

If properly introduced, the strategic approach can be used to meet planning objectives, bring about wide support at the political level, have the capacity to keep reducing the levels of trauma, have appeal to the community as a whole. It provides a base for stakeholders to implement core programs, but give enough flexibility for individual States, Territories and local government councils to adopt policies to meet local needs.

Road safety in Australia is a shared responsibility where Federal, State and Territory Governments complement each other. The Federal Government has a national leadership role, while the States and Territories are responsible for implementing local policies affecting their communities. The Federal Government has, because of the social and economic significance of road trauma, taken an important role in setting the national road safety agenda, in bringing together national policies, and in establishing vehicle safety standards.

Federal, State and Territory Governments have worked with industry, private sector and community groups to develop a national strategic approach to broaden awareness and to increase community responsibility for the road trauma problem.

THE STRATEGIC APPROACH - A "LIVING" BLUEPRINT FOR ACTION

Australia's approach recognizes that there is an ongoing problem; that governments and the community cannot afford to be complacent. Road trauma occurs in a dynamic and changing environment, socially and economically. The ways of tackling the problem have to be adaptive and continually improved if further gains are to be made.

Australia was aware that many countries were actively planning to reduce their levels of road trauma, and recognised the need to focus on components of the problem that would achieve most gains within local physical, political, social and financial environments.

We expressed a will to adopt best world wide practice where that was possible, and attempted to make road safety a major economic, as well as a social issue. We realised that strategic planning was critical if we were to achieve consistent levels of safety across the country and then achieve even gains in future.

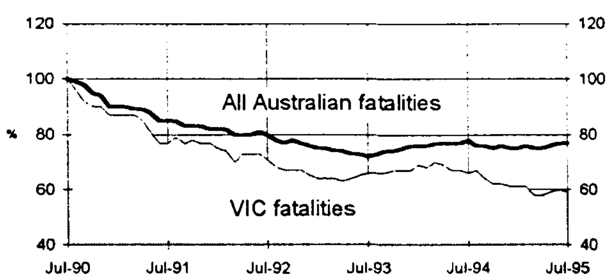
The Federal Government, got together with States and Territories, local government and other key community stakeholders to develop a National Road Safety Strategy. The Strategy seeks to ensure that Australia's road safety performance in the year 2001 would be equal to that of the leading developed countries in the world. As mentioned earlier, a target of 10 deaths per 100,000 population was set. We are close to that rate now and will adjust the target downwards as time goes by.

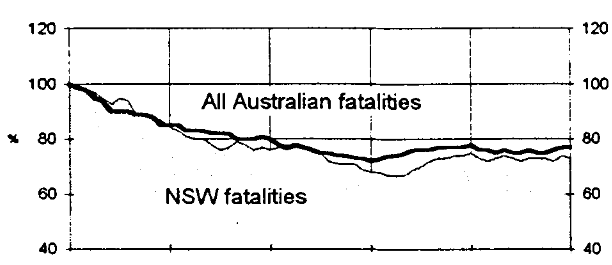
The National Strategy has broad community ownership and wide community involvement in its implementation as a key objective. It emphasises road safety as both a major public health and economic issue. It truly provides the framework for a coordinated national assault on road trauma in the coming years.

It has been endorsed by all ministers at the Australian Transport Council, and has the support of some 70 organisations both within and outside government. Its purpose is to place road safety firmly on all national agenda.

The National Strategy and Action Plan were reviewed in June 1995. This was carried out by the same diverse group of stakeholders responsible for the development of strategy. The concerns at this review conference targeted what more needs to be done if Australia is to meet its goal of 10 deaths per 100,000 population by the year 2001. The issues underlining the trends in the road toll are most important in answering this question.

Taken nationally the improvements in the road toll over that past five years have been quite dramatic. But the trend across Australia has been far from uniform both in terms of outcomes from individual States and results achieved in metropolitan and rural areas. As a result of progressive politics, and a fairly substantial investment in enforcement supported by public education, Victoria and New South Wales have shown improvements below the national trend.

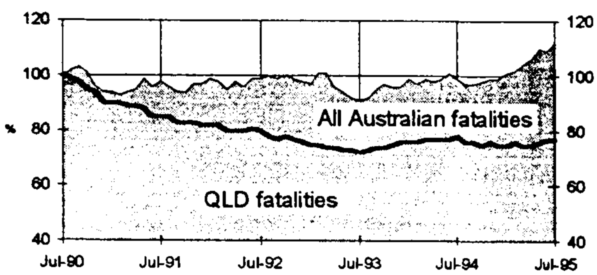


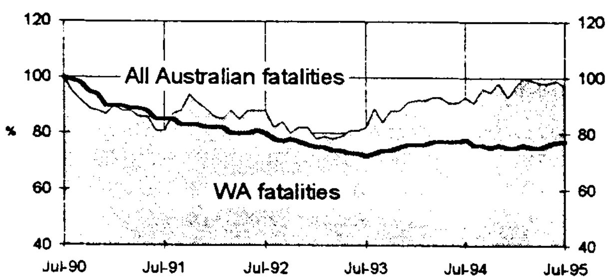


When it is recognised that some 58% of national vehicle kilometres travelled are carried out in Victoria and New South Wales, these two states are clearly the driving force underlying the national figures.

South Australia, Tasmania, the Northern Territory, and the ACT made solid progress in the years to 1994.

By comparison Queensland and Western Australia went against the national trend. In the case of Queensland the road toll has consistently remained unchanged over the past 5 years.





Western Australia had a relatively high fatality rate in 15-24 age group, a high incidence of single vehicle crashes, and nor real improvement in the rural road toll.

It is useful to compare the reductions in fatal crashes between urban and rural area. The percent reductions in fatal crashes by speed limit and state from 1989 to 1994 were:

Percent Reduction from 1989 to 1994

	100kph or more	*less than 100kph*
Australia	27.7%	32.5%
New South Wales	26.2%	31.8%
Northern Territory	26.5%	65.2%
Queensland	7.3%	(3.0%)*
South Australia	20.4%	33.6%
Tasmania	27.9%	20.0%
Victoria	51.5%	57.3%
Western Australia	2.2%	11.2%

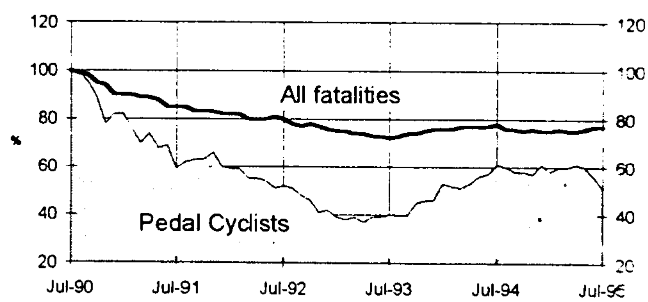
* - denotes an increase in fatal crashes [source: FORS]

These data indicate how overall results can disguise variations amongst States and Territories. Victoria has performed very strongly in both urban and rural areas. Most other States and Territories tend to cluster around the national average in terms of reductions in the number of fatal rural crashes.

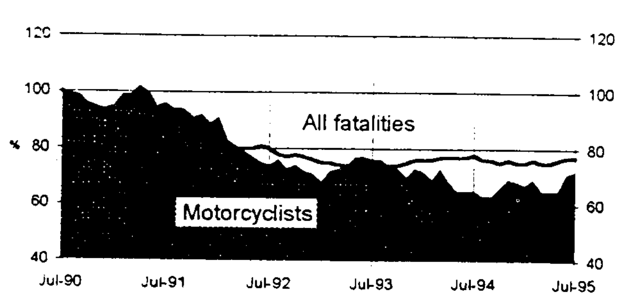
However, Western Australia and Queensland until recently have experienced very little improvement in both rural open road crashes and urban crashes. In 1989 Queensland and Western Australia ranked first and second among the states in road fatality rates, now they rank last and second last.

Over the past 12 months, both States have begun to obtain regular reductions in their monthly road tolls. From our national perspective it is important that the principal regulatory policies are much the same nationwide, and there is no strong co-relation between these trends and national economic indicators.

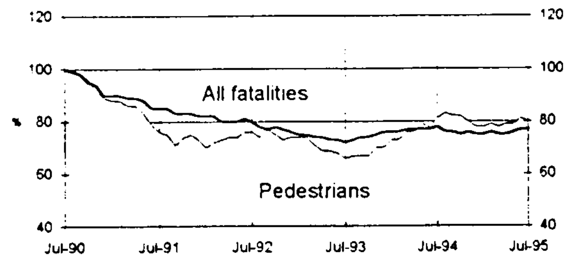
There are other interesting differences. Nationwide the introduction of compulsory helmet wearing and other measures to improve pedal cycle safety, such as bike paths, have brought about a profound reduction in fatalities.



Motorcycle safety has also done rather better than the national trend. Motor cyclists have experienced the same improvements as other road users, but at the same time there appear to be fewer people riding motorcycles.



By contrast, pedestrian safety has become a greater relative problem.



Rural road safety continues to be a problem throughout Australia, raising a number of specific issues which I will return to later.

THE 1996 ACTION PLAN - A NEW FOCUS

The foregoing gave a snapshot of some of the underlying issues which were focused upon in some detail at the June 1995 conference to review the National Road Safety Strategy and Action Plan.

The review conference proposed a revised National Action Plan which will in the short term target enforcement and management of speeding, drink driving, fatigue, and other high risk actions. The conference emphasized that greater priority should be given to rural and pedestrian road safety issues.

The need for medium term changes in driver behaviour and attitude were also identified. These involve the strategic deployment of high levels of both enforcement and public education as tools to change the safety culture. Vehicle standards, investment in roads, a focus on rural initiatives, "blackspot" remedial works, local government and community involvement were seen as the means of achieving long term changes.

The draft 1996 National Action Plan has been prepared and is scheduled to be considered by Transport Ministers in June 1996. The 10 areas suggested for new emphasis in the 1996 national Road Safety Action Plan include:

- **Rural & Remote Areas Measures:** Rural road safety continues to be a problem throughout Australia. The Action Plan focuses on development of a Rural Action Plan and includes a number of specific measures coming from a recent Rural Road Safety Seminar.
- **Enforcement & Resources:** Action Plan calls for upgrading of enforcement intensity and supporting publicity in regions with bad crash records. It also proposed development of

systems for regional monitoring and evaluation of enforcement of road safety measures.

- **Speed Management:** Action Plan aims for increased safety through improved compliance with speed limits that better reflect relative safety of different roads, and well publicised reduction of speed enforcement tolerances, rationalisation of speed limits, and reviewing demerit points for speed offences to better reflect risks involved.
- **Drink Driving & Drugs:** Measures include upgraded emphasis on drink driving enforcement, general public awareness, and targeting drivers who continue to drive at high BACs. Also calls for compulsory carriage of licenses, periodic random licence checks at RBT tests, and licence revocation and associated publicity to deter unlicensed driving.
- **Driver Fatigue Management:** Involves investigation into causes of driver fatigue and ways of correcting it.
- **Vehicle Standards & Cost Effective Safety Technology:** Action Plan will place increased emphasis on vehicle and road safety standards. It calls for the monitoring of new safety related technology and intelligent systems which are cost effective.
- **Multi Media Education:** Action Plan supports the development of multimedia packages aimed at educating and training young drivers and other vulnerable road users.
- **Improved Safety for Young and Novice Drivers:** Further research will be undertaken into driver training, driver competencies, testing and licensing requirements.
- **Vulnerable Road Users - Older Road Users & Pedestrians:** The implications for vulnerable road users will be assessed in the development of all road safety management programs.

CHANGING THE "CULTURE"

The Australian community has shown the capacity to accept measures provided they are perceived to be relevant to the problem, are practical, and the reasons are clearly and simply articulated.

Two particular areas illustrate the point that we can change the "culture" in regard to road safety.

Drink driving remains a major road safety problem in Australia. But we are making significant inroads. In the early 1980s some 44% of driver fatalities had been drinking over the legal limit (0.08 in most jurisdictions). Today that figure is below 30%, and approaching 20% in some of the more populous states while the limits are 0.05 for fully licensed drivers and zero or 0.02 for the probationary, heavy vehicle, public transport, and dangerous goods vehicle drivers. Around 20 percent of drivers have a zero or 0.02 BAC limit.

Social attitudes to drinking and driving have moved radically. Drink driving, once considered a relatively minor issue by drivers, is now seen as strictly anti-social behaviour by a vast majority of the Australian population.

The involvement of young people, particularly males in alcohol related crashes has reduced. Importantly there is the clear expectation in the community that anywhere and anytime you could be breath tested. This is made so by the fact that in some areas police have exceeded expectations and are breath testing one driver in two every year, and at high risk periods, one in one.

We have not yet beaten drink driving but we have made great progress and that is a direct outcome of those measures in the 'Package' targeting alcohol.

One measure opposed by many, but solidly supported by our research at the time, was the making of bicycle helmet wearing compulsory. The statistics available since the measure was introduced provide unqualified support for the move.

Since the introduction of mandatory helmet wearing bicycle related fatalities have fallen by 50% and serious head injuries from bicycle crashes have also fallen: reduction from 55% to 70% have been recorded in different states. Success by any measure. Initially there was some reduction in bicycle usage, but with the exception of teenage males, usage has generally returned to long term rates.

LOCAL GOVERNMENT & COMMUNITY INVOLVEMENT

Right from the very start local government was involved in the process of building safer communities. Local government, through the Australian Local Government Association and the Institute of Municipal Engineers, and other professional Associations, have become key stakeholders in the strategies and must continue to play in working closely with the community at "grass roots" level to improve road safety.

RESEARCH AND PUBLIC EDUCATION

Research is vital to the development of effective cost effective and publicly acceptable road safety initiatives.

The National Road Safety Strategy emphasises the importance of research and public education, and provides a framework within which our efforts are coordinated at a national level to support road safety priorities agreed between the Commonwealth, the States and other stakeholders. The Research Mangers' Forum forms part of that process.

Federal funding has played a significant part in this national research and public education effort. Vehicle standards research has of course been one of our core priority areas, but we have also funded major research projects on road user issues such as drink driving, novice drivers, fatigue and speed management, as well as national statistical databases.

On the vehicle side, our research effort on technical engineering issues has been complemented by extensive work on analysis of the costs and benefits of proposed standards. In Australia we are well aware of the benefits of international harmonisation, but we do not adopt international standards uncritically. I am happy to be able to say that research we have sponsored provides clear evidence of the net economic benefits of vehicle standards regulation: quite apart from the less tangible benefits in human terms.

In public education Australia has been able to tackle many of the key road safety problem areas through professionally based large scale national mass media and public relations programs that have proven particularly successful. An important role has been to facilitate cooperation and coordination with the States in the development and implementation of road safety public education activities across Australia.

The aim has been to provide focus for developing and improving long term countermeasures, ensuring cost effectiveness and promoting best practice. Two national forums covering police traffic enforcement, media promotions and public education are pivotal in this regard and are directly linked to the National Road Safety Strategy.

VEHICLE STANDARDS

For more than a quarter of a century Australia has recognised the importance of improving the integral safety of its car fleet. Australian vehicle safety standards have been among the most stringent in the world. We started early in occupant protection with compulsory fitment and use of seat belts and built on international experience, with the support of local industry. We have developed and implemented a full frontal barrier crash test for passenger cars based on FMVSS 2078 and have issued a draft rule for side impact, which accepts both US and ECE Standards. Australia has also upgraded occupant protection requirements for light commercial and four wheel drive vehicles to a level comparable to passenger cars.

The Federal Office of Road Safety has always had a central role in this area, but since 1989 it has operated under Federal legislation. The development of new standards involves close consultation between all Australian governments, manufacturers and consumer representatives, and a rigorous regulatory impact analysis. Established in 1992, the National Road Transport Commission has a key role in achieving national uniformity in vehicle regulation. Our joint objective is to have uniform regulatory arrangements that extend from the current Australian Design Rules for new vehicle construction right through operations, registration and licensing.

Like many other countries, Australia has moved away from design-specific regulations towards performance-based standards: leaving maximum room for innovative solutions to produce the best results as cheaply as possible.

Our regulatory impact procedures exert a discipline that often requires a very significant local research effort. We have a very efficient and cost-effective certification and type approval system, based on a state-of-the-art quality assurance approach.

We are committed to international harmonisation of standards and conformity testing. Australia in relative terms is a small market with four major manufacturers who are today internationally competitive. The industry is export oriented and continually seeking new markets. We also import vehicles from major centers around the world. Harmonisation of standards is, therefore, an important aspect of our philosophy. For us, that includes:

- Timely adoption of the best in international practice, once we have satisfied ourselves that there is scope for worthwhile safety gains, at a reasonable price, in the Australian context.
- A flexible approach that accommodates alternative standards where necessary (our approach to side impact protection is an instance of this).
- A strong commitment to *'functional equivalence'* as a circuit breaker to facilitate trade
- A commitment to contribute to the development of truly international vehicle regulations.
- A substantial local research effort to support these objectives: including detailed studies of real-world crashes, crash testing, modelling and economic analyses.

Within APEC, Australia is leading the harmonisation project in the Transport Working Group. This project is aimed at establishing a basis for trade facilitation through harmonisation of standards and mutual recognition of conformity testing. There is general support for UN/ECE regulations to be the basis for harmonisation of standards. Australia will be pressing for real progress in these areas over the next few years.

Australia is also working with New Zealand to establish a mutual recognition regime for automotive products under the auspices of the Trans - Tasman Mutual Recognition Arrangement. This will involve agreement of a common pool of standards (based on UN/ECE) and a common type approval system. The work will be carried out in conjunction with a review of the Australian Design Rules as part of a general review of business regulation.

In the area of bus safety, Australia has run at the forefront of early implementers with standards covering rollover strength, seat mounting strength, emergency exits and lap/sash seat belts. Since July 1994 all new coaches are required to have lap/sash seat belts and seat mountings capable of withstanding 20g.

For cars, Australia has introduced a performance standard for full frontal crashes, based on FMVSS 208 and are working toward standards for offset and side impacts. Publicity surrounding this work has played an important role in increasing public awareness of the importance of vehicle safety features, and road safety generally.

Four wheel drive vehicles and vans are an important and growing part of the Australian passenger vehicle fleet and Australia has a research program including crash testing and in-field crash analysis for those vehicles. A number of truck safety projects are also under way, ranging from spray suppression in wet weather, to the economic evaluation of anti-lock brakes.

The Federal Office of Road Safety is working closely with the international standards agencies to ensure that the new standards in the pipeline reflect best practice. Under the intergovernmental agreement FORS works with the National Road Transport Commission in the development of new standards [Australian Design Rules]. These include:

- A new draft standard for passenger car side impact.
- A new design rule addressing offset crash standards.
- Heavy vehicle braking standards for combination vehicles.
- Requirements for school bus flashing warning lamps.
- A code of practice of low effort heavy vehicle braking.
- Requirements for coach sleeper berths.
- Review of bus design rules.
- Anti-lock braking for heavy vehicles.
- Review of requirements for mechanical connections between vehicles.

WHERE DO WE GO FROM HERE?

In recent years Australia has made real improvement in its road safety performance. The key question is can we keep this performance up.

The Federal Office of Road Safety and others had quantitative analysis undertaken to assess the extent of that impact on road safety performance. Estimates suggested that up to a third of our improvement in road safety performance could have been attributed to economic conditions.

The hindsight, I believe that travel probably flattened out rather than kept on increasing at a constant rate, and that our broad ranging strategic measures were concurrently pushing down the actual road toll. But the national toll has begun to increase and this will continue to put further pressure on us as policy makers.

Our policy responses have to be such that they will offset the increase in economic activity and travel. We cannot afford to just sit back and say that the outcomes of economic growth, population changes, or shifts in social behavior or attitudes cannot be influenced. That won't be accepted here.

As policy makers we have to be able to influence both the political arm of government and the community that our goal is possible and is worthwhile. We also have to develop a range of cost effective measures which will address areas where greatest gains can be made and the highest risks can be reduced. We are confident that the feasible and adaptable strategic planning process with wide input from the community will help achieve our aims.

The continuing improvements in Australia's economic conditions will pose new challenges for road safety. We are committed to managing road safety during the economic upturn.

To continue with the present mix of road safety strategies and regulatory tools would without doubt during the economic recovery see the level of road trauma on the upswing. In fact in recent times our statistical collection and analysis has clearly revealed the possible beginnings of such a trend.

The developments of the past five years have provided Australia with the means to cope with the problems of the future. I believe their are a number of benefits that will flow from national strategic planning.

- "Stakeholder" involvement results in outcomes greater than governments and individuals can achieve separately.
- Effective partnerships and linkages together with leadership make the strategy work and coalesce the interests of divergent groups.
- It challenges us all to be creative, innovative and accept best practice.
- It energises groups in the community and creates the climate for community commitment.
- It provides a basis for groups who may not get full support in their community to point to the national strategy and action plans.

Strategic planning won't work without drive, determination and commitment. It won't work without some people of bodies being willing to look beyond the horizon for new and better ways to do things - and to actually implement new ideas. We will only get out of this planning mechanism what we put into it.

The Strategy, including its immediate antecedents referred to earlier, has seen the annual number of road fatalities in Australia fall from 2,900 in 1989 to 1,941 in 1994, even though much of the action taken, particularly in the vehicle, road, and school education areas will have their greatest effect in years to come. There was a slight increase last year, but this was not consistent across the whole country, and hopefully will prove to be a statistical aberration. From our short experience, I would recommend the process to others. Well used, it can help accelerate improvement in road safety.

The social and economic costs of road trauma are so great that we must in a structured way continue to seek cost effective measures which will help to minimise the number of people killed or seriously injured in road crashes.

The National Road Safety Strategy offers Australia the path ahead where each level of government (Federal, State, and local) and the major industry, health, education and consumer sectors play their roles toward an agreed national objective.

Vehicle safety will be a major priority focus for the rest of the 1990's. In performing its role, the vehicle industry has to address the key issues of harmonisation of vehicle standards, an issue in which Australia is taking a strongly proactive role - many issues of which are being discussed at this ESV Conference. Occupant protection is clearly a key ongoing issue, as are pedestrian safety and developing areas of I.T.S.

Australia will be supporting the America initiative to focus on our four key areas of vehicle safety. In the short term, improve occupant protection and pedestrian safety will be the key vehicle design issues for Australia.

Section 3
Executive Panel

Opportunities for World-Wide Harmonized Safety Standards for Occupant Protection in Passenger Cars, Including Advanced Frontal Protection

Moderator: John Bowdler, Australia

United States

Andrew H. Card
President and Chief Executive Officer
American Automobile Manufacturers Association

INTRODUCTION

Global harmonization of motor vehicle regulations has recently taken a major step forward. The recommendations made at the Seville Conference of the Transatlantic Business Dialogue (TABD) led to the signing of a new "Transatlantic Agenda" and Action Plan by President Clinton and officials of the European Union during the Madrid Summit on 3 December, 1995. The Transatlantic Agenda is aimed at creating the New Transatlantic Marketplace. In this context, the April 10-11 Transatlantic Automotive Industry Conference on International Regulatory Harmonization was held to advance the harmonization of technical regulations affecting the construction and use of motor vehicles.

Compliance with diverse national and regional requirements imposes substantial cost penalties and engineering, design and manufacturing constraints, that are fundamentally inconsistent with the reality of a global auto market, and thus adversely affect world trade. These inconsistencies also diminish the potential to achieve societal objectives, notably in the field of safety and environment, and also reduce vehicle affordability and customer choice. With the rapid development of new markets in developing nations, there is a great risk that the number of new and different regulatory requirements world-wide will escalate quickly, creating new "technical barriers to trade."

European and U.S. automakers believe that this strategically uncoordinated approach no longer is sustainable either in terms of resources or results. It must be emphasized that industry is still committed to abide by the high levels of safety and environmental protection offered by today's standards. Yet it seems difficult to comprehend the need for multiple differing approaches to address the same objectives.

HARMONIZATION

Our vision of regulatory harmonization would eliminate completely the barriers to trade resulting from unwarranted differences in vehicle regulation, certification and compliance procedures. At a minimum, this vision includes world-wide parallel acceptance of major rulemaking authorities' vehicle regulations and mutual recognition between those major authorities of each others' vehicle regulations. Equally critical would be a common set of testing procedures and compatible certification systems.

Our vision also features a strong mechanism for coordinating pre-regulatory research and the global development of common new regulations.

Achieving progress in harmonization is a major challenge that will require much dialogue and cooperative effort. Industry's thinking on this issue, which is still evolving, is captured in the following list of principles. These ten principles, a distillation of our present thoughts on regulatory harmonization, are intended to initiate dialogue between the world's governments and industry.

- Commit to global regulatory harmonization by becoming Contracting Parties to the 1958 Agreement[1] and participating in the development of new UN-ECE regulations with the intent of adopting and implementing them to the maximum extent feasible.[2]
- Work through and strengthen Working Party 29

to expand it into a broadly recognized body for the development of global vehicle[3] regulatory requirements.

- Establish a work program to contribute to the global harmonization of regulatory differences, to the maximum feasible extent.
- Continue the process of global harmonization of vehicle regulatory requirements and expand these discussions to all countries.
- Establish mutually recognized certification processes.
- In the process of global harmonization: establish means to incorporate functional equivalence of alternative vehicle regulatory requirements in the regulatory process; and establish means to achieve mutual recognition of corresponding regulatory requirements.
- Coordinate pre-regulatory research on the need for and development of new regulatory requirements, thereby minimizing the likelihood of future divergence.
- Avoid developing unique new national or regional technical requirements without adequate justification[4].
- Improve processes for informing the public about the development of harmonized regulatory requirements.
- Encourage a policy of accepting vehicles fully meeting ECE or U.S. or EU requirements as equivalent. (EU, Australia, Canada, Japan and South Africa have already accepted UN-ECE regulations.) The adoption of hybrid requirements for vehicles (selectively combining elements of different jurisdictions) should be avoided.

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[1] United Nations Economic Commission for Europe Agreement Concerning the Adoption of Uniform Technical Prescriptions for Wheeled Vehicles, Equipment and Parts which can be Fitted and/or be Used on Wheeled Vehicles and the Conditions for Reciprocal Recognition of Approvals granted on the Basis of these Prescriptions (as amended).

[2] The U.S. government is ready to commit to global harmonization but considers that the 1958 Agreement should be amended. Discussions are ongoing.

[3] Vehicle defined as including equipment and parts.

[4] As defined in WTO, Articles 2.1-2.5

Certified Once, Accepted Everywhere

The following discussion relates to harmonization of safety regulations. The concepts, ideas and recommendations presented here may apply equally well to environmental regulations. Global Harmonization of motor vehicle safety regulatory requirements is desirable to assure a high-level of safety performance worldwide and a broad range of consumer choice. Industry would like to realize a condition in which certification to a regulatory requirement in one jurisdiction, would be universally accepted in other jurisdictions. The following observations and recommendations are made to move toward this objective.

Increase Cooperation Concerning the Development of New, Unique Regulatory Requirements Where a new requirement is determined to be necessary, regulators are encouraged first to utilize an existing regulatory requirement on that subject from another regulatory system. If one is not available, then the new requirement should be developed in consultation with the appropriate concerned parties. The concerned parties should develop fora to establish common regulatory agenda and research activities.

Mutual Recognition of Existing Regulatory Requirements in Areas of Common Interest Mutual recognition is the process whereby two or more countries/regions recognize each other's regulatory requirements on a specific subject as satisfying the requirements of both/all parties. This is a step toward global harmonization that can be immediately initiated and can help to eliminate the trade barrier effects of differing regulations. Mutual recognition will facilitate the application of a single design solution for a given product that satisfies the regulatory requirements for markets around the world.

The "Mutual Recognition" of comparable technical regulatory requirements should be based on an assessment of "Functional Equivalence" of the regulatory requirements. There are significant precedents for determination of functional equivalence. Today, several countries, e.g., Australia, Argentina, Brazil, Israel, Japan and Korea have taken this pragmatic approach of accepting regulatory requirements of other countries/regions as alternatives to their own national regulatory requirements.

For example, Australia recently announced a new national standard, Australian Design Rule DSIOP for side impact protection. In November 1992, Australia's MOT announced that it was conducting a review of side impact protection. There were three alternatives under study: (1) specify the U.S. standard FMVSS 214, (2) specify the European standard ECE R95 or (3) allow manufacturers the choice of designing vehicles to either the U.S. or European standard. After completing a vigorous study of safety benefits for all three alternatives, it was determined that both the U.S. and ECE regulations provided positive safety benefits.

Although this analysis indicated that the "Harm benefits" of ECE R95 were slightly greater than for FMVSS 214, Australia noted that vehicles built to either standard provided substantial safety protection and therefore decided that allowing the manufacturer the option of certifying to either standard provided "the most equitable solution and one which was likely to provide the greatest positive benefits."

The following criteria are suggested recommendations for use by regulatory agencies in determining functional equivalence for safety regulatory requirements:

1. Same/equivalent regulatory language or same/equivalent intent or purpose.
2. Same/equivalent design execution to meet regulatory requirements.
3. Substantial and substantive successful prior experience with acceptance of differing regulations, concerning the same systems in a single jurisdiction.
4. Same/equivalent test performance levels.
5. No substantive safety performance difference based upon field accident injury data assessment.

An initial necessary step is a simple listing of regulatory requirements to identify where regulatory requirements exist that address the same safety need or purpose. **Attachment 1** contains brief and expanded listings of those regulatory requirements for common systems in ECE/EU/U.S. with an initial assessment of the "potential" for mutual recognition within short, medium or long terms. The grouping of regulatory requirements into short, medium and long term time frames is not indicative of priority for action. Industry believes that action to determine functional equivalence for all areas of regulatory requirements should start simultaneously. In this context, short term means within one year, medium term 1-3 years and long term 3-5 years.

The preceding criteria form the basis for the proposed functional equivalence assessment shown in **Attachment 2.** The first six columns contain the technical details taken from the U.S. and European regulatory requirements including notation of technical differences in regulatory requirements. The seventh column contains the product impact, if any, resulting from these technical differences. The last column allows for a safety benefit analysis by appropriate government regulatory experts.

The most important, and arguably the most subjective, portion of the equivalency process is the assessment of performance and benefit differences. Our selected comparisons are those that are intentionally "simple" to assess, using some or all of the first two or three criteria. It may be necessary to conduct joint research to develop a more objective comparative assessment for those regulatory requirements that are more divergent.

If assessment based on the first three criteria does not provide a determination of functional equivalence, then more involved assessment criteria will need to be developed, such as:

- Analytical Modeling.
- Jury Assessment.
- Comparative Testing.
- Real World Accident Data Analysis.

Cooperative Development of Future Regulatory Requirements

<u>Minimize Divergence</u> The functional equivalency assessment efforts should lead to mutual recognition of many existing regulatory requirements. To minimize regulatory divergence, government regulators around the world should establish an agreed schedule of regulatory meetings, to facilitate a continuous exchange of communications, and coordinated pre-regulatory research. Regulators may chose to initiate this exchange through working groups under WP29.

<u>Pre-Regulatory Research</u> The cornerstone of any future regulatory harmonization is cooperative pre-regulatory safety research including development of a common research agenda. Therefore, the bilateral agreement between NHTSA and the EU Commission should include a commitment for coordinated research. A possible organizational concept that establishes "centers of responsibility" is provided in **Attachment 3.** Also included in this attachment is a listing of potential cooperative international research for consideration. Industry, in conjunction with regulatory bodies, should at the earliest possible date establish a forum to initiate a process of international cooperative safety research.

ATTACHMENT 1.
EU/U.S. LISTING OF REGULATIONS

Examples of Performance Elements
Regulated in the U.S. and EU

SHORT TERM
Windshield defrosting and defogging systems
Windshield wiping and washing systems
Tire selection and rims
Headlamp concealment devices
Occupant protection in interior impact_(frontal)
Head restraints
Impact protection for the driver from the steering control system
Steering control rearward displacement
Glazing materials
Door locks and door retention components
Seating systems

MEDIUM TERM
Controls and displays
Lamps, reflective devices and associated equipment
Rearview mirrors
Theft protection
Vehicle identification number - basic requirements
Air brake systems
Passenger car brake systems
Seat belt assemblies
Seat belt assembly anchorages
Child restraint systems
Seating_reference point
Side impact anthropomorphic_test dummy

LONG TERM
Occupant crash protection in frontal impact
Side impact protection
Fuel system integrity
Flammability of interior materials
Bumpers
Side impact barrier
Child anthropomorphic test dummies

Mutual Recognition - Safety

Short Term Candidates (within 1 year)

Subject	ECE	EU	U.S.
Windshield defrosting and defogging systems	-	78/317/EEC	FMVSS 103 **
Windshield wiping and washing systems	-	78/318/EEC	FMVSS 104 **
Tire selection and rims	30, 54, 64	92/23/EEC	FMVSS 110, 109 **, 119, 120
Headlamp concealment devices	48	76/756/EEC	FMVSS 112 **
Occupant protection in interior impact (frontal only)	21	74/60/EEC	FMVSS 201
Head restraints	25	78/932/EEC	FMVSS 202 **
Impact protection for the driver from the steering control system	12	74/297/EEC	FMVSS 203
Steering control rearward displacement	12	74/297/EEC	FMVSS 204 **
Glazing materials	43	92/22/EEC	FMVSS 205
Door locks and door retention components	11	70/387/EEC	FMVSS 206
Seating systems	17	74/408/EEC	FMVSS 207

** Candidate for U.S. Regulatory reform

Mutual Recognition - Safety

Medium Term Candidates (1-3 years)

Subject	ECE	EU	U.S.
Controls and displays	-	78/316/EEC	FMVSS 101 **
Lamps, reflective devices and associated equipment	*	*	*
Rearview mirrors	46	71/127/EEC	FMVSS 111
Theft protection	18	74/61/EEC	FMVSS 114
Vehicle identification number - basic requirements	-	76/114/EEC	FMVSS 115
Air brake systems	13	71/320/EEC	FMVSS 121
Passenger car brake systems	(13H)	-	FMVSS 135
Seat belt assemblies	16	77/541/EEC	FMVSS 209 **
Seat belt assembly anchorages	14	76/115/EEC	FMVSS 210 **
Child restraint systems	44	-	FMVSS 213
Seating Reference Point	-	77/649/EEC	Part 571.3
Side Impact Anthropomorphic test dummy	95	(draft)	Part 572

* see separate sheet
** Candidate for U.S. Regulatory reform

Mutual Recognition - Safety

Long Term Candidates (3-5 years)

Subject	ECE	EU	U.S.
Occupant crash protection in frontal impact	94	(draft)	FMVSS 208
Side impact protection	95	(draft)	FMVSS 214
Fuel system integrity	34,94,95	70/221/EEC	FMVSS 301
Flammability of interior materials	-	95/28/EEC	FMVSS 302
Bumpers	42	-	Part 581
Side impact barrier	95	(draft)	Part 587
Child anthropomorphic test dummies	44	(draft)	Part 572

Lighting and Light Signaling Components

Subject Harmonization	ECE	EU	U.S.
Installation of lighting/signaling	48.01	76/756	FMVSS 108
Headlamps	R 1.04 + R 2.04 R 5.03 R 8.02 R 20.02 R 31.01	76/761	FMVSS 108
Reversing lamps	R 23.01	77/539	FMVSS 108
Direction indicator lamps	R 6.02	76/759	FMVSS 108
Hazard warning signal	R 48.01	76/756	FMVSS 108
Stop lamps	R 7.02	76/758	FMVSS 108
Rear registration plate lamp	R 4	76/760	FMVSS 108
Front position lamps (U.S.: parking lamps)	R 7.02	76/758	FMVSS 108
Rear position lamps	R 7.02	76/758	FMVSS 108
End-outline marker lamp (U.S.: clearance lamp)	R 7.02	76/758	FMVSS 108
Retro reflectors	R 3.01	76/757	FMVSS 108
Side-marker lamp	R 91	-	FMVSS 108
Daytime running lamps	R 87	-	FMVSS 108
Center high mounted stop lamp	R7.02	76/756	FMVSS 108

ATTACHMENT 2.
FUNCTIONAL EQUIVALENCE PROCESS

FMVSS 209
77/541/EEC, ECE R16
SAFETY BELTS

ITEM	*FMVSS*	*EU*	*ECE*	*TECHNICAL DIFFERENCES IN REGULATIONS*	*PERFORMANCE DIFFERENCES FOR PRODUCTS*	*PRODUCT IMPACT*	*SAFETY BENEFIT*
SUBJECT	Seat Belt Assemblies -209	Safety-belts and Restraint Systems for Adult Occupants of Power-driven Vehicles -- 77/541/EEC	Safety-belts and Restraint Systems for Adult Occupants of Power-driven Vehicles-- ECE R-16				
VEHICLE APPLICATION	Passenger cars, MPV's, trucks and buses.	Power-driven vehicles with four wheels; a design speed > 25 km/h and intended for use as individual equipment by adult persons in forward facing position.	Power-driven vehicles with three or more wheels and intended for use as individual equipment, by persons of adult build occupying seats facing forward.	77/541/EEC is applicable to M1 vehicles -- a passenger vehicle with a capacity of 9 passengers or less including the driver.			
SEAT BELT SYSTEM HARDWARE APPLICATION	Type 2 front and rear outboard seat positions. Types 1 or 2 front and rear center seat positions. FMVSS 208 upper torso requires emergency locking retractor; lower torso (lap belt) requires ELR, ALR or manual adjustment device.	Type A (lap/shoulder belt) for front and rear outboard seat positions. Type A or B (lap belt) in front and rear center seat positions.	Type A (lap/shoulder belt) front and rear outboard seat positions. Type A or Type B (lap belt) in front center and rear center seat positions.	Basically the same for three and two point belt systems. Except: 1) EEC/ECE retractors require two emergency locking sensors; FMVSS 209 requires one. 2) FMVSS 209 requires a child locking seat locking device [except driver's seat] that is integral with belt & retractor assembly.		Seat belt systems hardware are basically the same, except for compliance to some unique performance requirements and procedures noted below.	
CAUTION: THIS IS A SUMMARY ONLY. REFER TO THE COMPLETE TEXT FOR DESIGN PURPOSES.							

ITEM	*FMVSS*	*EU*	*ECE*	*TECHNICAL DIFFERENCES IN REGULATIONS*	*PERFORMANCE DIFFERENCES FOR PRODUCTS*	*PRODUCT IMPACT*	*SAFETY BENEFIT*
TEST PROCEDURES & REQUIREMENTS	WEBBING SENSITIVITY: If the retractor is sensitive to webbing withdrawal it must not lock before the webbing extends 2 inches (50.8mm) when the retractor is subjected to an acceleration $\leq$ 0.3g -- test with webbing at 75% extension -- apply acceleration of 0.3g within 0.05 seconds or at a rate $\geq$ 6g's/sec.	WEBBING SENSITIVITY: retractor must not lock at strap acceleration of less than 0.8g in the direction of unreeling. If locking does not occur before 50 mm of webbing is unwounded, this is considered satisfied. Retractor -- must lock within 50 mm of strap movement at webbing accel. relative to the retractor of not less than 2.0g -- test with 300 +3mm of webbing remaining in the retractor -- apply accel. at a rate > 25g's/sec < 150g's/sec.	WEBBING SENSITIVITY: retractor must not lock at strap accelerations of less than 0.8g in the direction of unreeling. If locking does not occur before 50mm of webbing is unwound, this is considered satisfied. Retractor -- must lock within 50mm of strap movement at webbing accel relative to the retractor of not less than 2.0g -- test with 300 n$\pm$ 3mm of webbing remaining in the retractor -- apply accel at a rate $\geq$ 25g's /sec.	FMVSS 209 does not require locking by this requirement.	Both FMVSS 209 and 77/541/EEC, ECE 16 have a No-lock requirements, but only 77/541/EEC, ECE 16 has a lock requirement. This does not have any effect on retractor lock-up because both regulations have a vehicle sensing lock-up feature as a primary method. EEC/ECE requires two methods of sensing emergency (or inertia) lock-up, whereas FMVSS requires only one. Apparent benefit is that occupant can verify that the retractor will lock-up by quickly pulling on belt. This feature is considered as a back-up to vehicle sensing lock-up, even though there is no evidence that such a feature is required.	Compliance with EEC/ECE requirements may be considered a nuisance to U.S. consumers because of the higher frequency of belt lock-ups.	
	EMERGENCY LOCKING: Retractor must lock within 25.4 mm of belt payout at 0.7g acceleration at a rate of$\geq$ 14 g's/sec.	EMERGENCY LOCKING: Retractor must lock within 50 mm of belt payout at 0.45g at a rate of $\geq$ 25 g's/sec < 150g's/sec.	EMERGENCY LOCKING: Retractor must lock within 50.0 mm of belt payout at 0.45g at a rate of $\geq$ 25 g's /sec.	EEC/ECE retractor is more sensitive to inertia locking conditions; 0.45g Vs 0.7g. Thus, a 0.45g sensor will lock-up infinitesimally sooner.	Retractor by locking with less inertia means that it will also lock-up quicker and easier during turning, braking, acceleration of the vehicle.	Compliance with EEC/ECE requirement may be considered a nuisance to the U.S. consumer because of more frequent locking during driving maneuvers.	
	TILT NO-LOCK: Retractor must not lock when tilted 15 degrees or less in any direction from installation position.	TILT NO-LOCK: Retractor must not lock when tilted 12 degrees or less in any direction from installation position.	TILT NO-LOCK: retractor must not lock when tilted 12 degrees in any direction from installation position.	EEC/ECE retractor may lock up to 3 degrees less than FMVSS retractor. This requirement allows reasonable window to meet the lower EEC/ECE tilt lock requirement; see below.	Compliance with EEC/ECE requirement allows the retractor to lock-up easier/quicker because of lower tilt angle.		

ITEM	*FMVSS*	*EU*	*ECE*	*TECHNICAL DIFFERENCES IN REGULATIONS*	*PERFORMANCE DIFFERENCES FOR PRODUCTS*	*PRODUCT IMPACT*	*SAFETY BENEFIT*
TEST PROCEDURES & REQUIREMENTS BELT & RETRACTOR ASSEMBLIES	TILT-LOCK: Retractormust lock when tilted more than 45° from installation point.	TILT-LOCK: Retractormust lock when tilted more than 27° from installation point.	TILT LOCK: Retractor must lock when tilted more than 27° from installation point.	Even though tilt lock angle is significantly different, the actual tilt lock angle between FMVSS and EEC/ECE is not, because of the closer inertia lock requirements, for example FMVSS retartors tilt lock from 24 - 32 degrees.			
	RETRACTING FORCES: If retractor is part of a lap belt shall be $\geq$ 0.6 Lbf (0.27 Kgf or 2.7N).	RETRACTING FORCES: If retractor is part of a lap belt shall be ≥ 7N (1.57Lbf).	RETRACTING FORCES: If retractor is part of a lap belt $\geq$ 7N (1.57 Lbf).	EEC/ECE has higher minimum retraction force requirements: must exceed 7 N (1.57 lbs) Vs 2.7 N (0.6 Lbs).	Higher retraction force assures belt stowage, however may result is reduction in perceived comfort.		
	If retractor is part of a combination shoulder/lap belt, shall be between 0.2 and 1.5 Lbf (0.9 N and 6.6 N) Starting with all webbing extracted, measure lowest force of extraction within $\pm$ 2 inches of 75% extraction stowed length when webbing is being retracted at a constant speed.	If the retractor is part of the shoulder belt, shall be between 2 and 7N (0.45 and 1.57 Lbf). Measures as close as possible to the point of contact with the manikin, while the belt is being retracted at an approx. speed of 0.6 m/min.	If retractor is part of a shoulder belt between 2 and 7N (0.45 and 1.57 Lbf). Measure as close as possible to the point of contact with the manikin, while the belt is being retracted at an approx. speed of 0.6 M per minute.	EEC/ECE has higher minimum retraction force requirement; 2 N (0.45 lbs) Vs 0.9 N (0.2 lbs).	No significant difference, because typical retraction efforts for U.S. vehicles exceed 0.45 lbs.		
	RETRACTOR DURABILITY: Same as ECE, except conduct 5,000 cycle with 90 N applied load at max. extraction. In addition to corrosion and dust alos exposed to 80° C for 48 hours. Then conduct 45,000 cycles; 50,000 cycles total. Then meet all functional checks.	RETRACTOR DURABILITY: Same as FMVSS, except conduct 40,000 cycle first, then expose to corrosion and dust (no high temp.). Then conduct 5,000 cycles, 45,000 cycles in total. Then meet all functional checks.	RETRACTOR DURABILITY: Same as FMVSS, except conduct 40,000 first, then expose to corrosion and dust (no high temp.). Then conduct 5,000 cycles; 45,000 cycles total. Then meet all functional checks	Testing procedures similar except EEC/ECE requires 5,000 cycles less test cycles, and has no exposure to high temperature.	FMVSS has high temperature exposure which can effect plastic components and 5,000 cycles more total cycles, plus conduct major ity of cycles after exposure rather than before as in EEC/ECE. Both require same functional checks for lock-up and retraction force after exposure and cycles, thus no safety benefit effect. But durability is more severe test for FMVSS.		

FMVSS 209
77/541/EEC, ECE R16
SAFETY BELTS

ITEM	*FMVSS*	*EU*	*ECE*	*TECHNICAL DIFFERENCES IN REGULATIONS*	*PERFORMANCE DIFFERENCES FOR PRODUCTS*	*PRODUCT IMPACT*	*SAFETY BENEFIT*
TEST PROCEDURES & REQUIREMENTS WEBBING	Width: With $\geq$ 46 mm under load.	Width: With > 46 mm under load.	Width: With $\geq$ 46 mm under load.	Similar requirements.			
	ELONGATION: Tensile force of 2500 Lbf webbing not to exceed -- Type 1, 20% Type 2, 30% pelvic and 40% upper torso.	ELONGATION: No specific requirements for *77/541/EEC*	ELONGATION: No specific requirements for ECE 16.	No EEC/ECE requirement.			
	STRENGTH AFTER ROOM CONDITIONING	STRENGTH AFTER ROOM CONDITIONING	STRENGTH AFTER ROOM CONDITIONING	FMVSS 209 more stringent than EEC/ECE 16.			
	LIGHT CONDITIONING	LIGHT CONDITIONING	LIGHT CONDITIONING	FMVSS 209 slightly more stringent			
	EXPOSURE TO $H2_0$	EXPOSURE TO $H2_0$ Immerse in $H2_0$ at temp. of 20 + 5 degrees C for 3 hrs.Test breaking srengthwithin 10 minutes of removal from $H2_0$.	EXPOSURE TO $H2_0$ Immerse in $H2_0$ at temp. of 20 + 5 degrees C for 3 hrs.Test breaking srengthwithin 10 minutes of removal from $H2_0$.	FMVSS 209 has no equivalent test.			
	CONDITIONING BY ABRASION	CONDITIONING BY ABRASION	CONDITIONING BY ABRASION	Both FMVSS 209 and 77/541/EEC, ECE 16 have very specific test requirements for abrasion, however, 209 is somewhat more stringent.			
	RESISTANCE TO MICRO-ORGANISMS Test only if material is not inherently resistant to micro-organisms	RESISTANCE TO MICRO-ORGANISMS No test.	RESISTANCE TO MICRO-ORGANISMS No test.	No ECE requirement.			
CAUTION: THIS IS A SUMMARY ONLY. REFER TO THE COMPLETE TEXT FOR DESIGN PURPOSES							

FMVSS 209
77/541/EEC, ECE R16
SAFETY BELTS

ITEM	*FMVSS*	*EU*	*ECE*	*TECHNICAL DIFFERENCES IN REGULATIONS*	*PERFORMANCE DIFFERENCES FOR PRODUCTS*	*PRODUCT IMPACT*	*SAFETY BENEFIT*
TEST PROCEDURES & REQUIREMENTS WEBBING	COLORFASTNESS Test procedure specific	COLORFASTNESS	COLORFASTNESS	EEC/ECE 16 no test required			
RIGID BELT COMPONENTS BELT ATTACHED HARDWARE	STRENGTH REQUIREMENTS:	STRENGTH REQUIREMENTS:	STRENGTH REQUIREMENTS:	Test procedures similar somewhat different pound forces applicable.			
	BELT ADJU.S.TING DEVICE:	BELT ADJU.S.TING DEVICE:	BELT ADJU.S.TING DEVICE:	Similar requirements with EEC/ECE 16 more stringent.			
	BUCKLE:	BUCKLE:	BUCKLE:	Similar requirements - accessible, easy to use. EEC/ECE more stringent in buckle opening test (13.5 Lbf vs 30 Lbf following tensile load to buckle. EEC/ECE requires a low temperature test -- 209 has no such requirement..			
	GENERAL REQUIREMENTS FOR RIGID PARTS:	GENERAL REQUIREMENTS FOR RIGID PARTS:	GENERAL REQUIREMENTS FOR RIGID PARTS:	No differences in requirements.			
		DYNAMIC TEST (30 MPH, SLED)	DYNAMIC TEST (30 MPH, SLED)	No FMVSS 209 requirement.	FMVSS 208 requires dynamic performance criteria. at least equivalent to 30 mph sled test.		

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FMVSS 210
76/115/EEC, ECE R14-03
SAFETY BELT ANCHORAGES

ITEM	*FMVSS*	*EU*	*ECE*	*TECHNICAL DIFFERENCES IN REGULATIONS*	*PERFORMANCE DIFFERENCES FOR PRODUCTS*	*PRODUCT IMPACT*	*SAFETY BENEFIT*
SUBJECT	Seat belt assembly anchorages - FMVSS 210	Seat-belt anchorages 76/115/EEC.	Seat-belt anchorages ECE 14.				
VEHICLE APPLICATION	Passenger cars, MPV's, trucks and buses.	Motor vehicles of categories M and N.	Motor vehicles of categories M and N.	Applicable vehicle differences -- vehicle weights.			
ANCHORAGES FOR TYPE OF SEAT BELT ASSEMBLIES REQUIRED	Type 2 for each outboard forward seating position, or Type 1 for other designated seating positions.	Front and all out-board seats shall have two lower and one upper belt anchorages; Type A front center and all other seat positions shall have two lower belt anchorages; Type B.	Front and all out-board seats shall have two lower and one upper belt anchorages; Type A front center and all other seat positions shall have two lower belt anchorages; Type B.	Basically the same for two and three point safety-belts	No difference; common seat belt restraint system anchorages to accommodate Type 1 (A) or Type 2 (B) seat belts.		
STRENGTH REQUIREMENTS	Type 2 (lap and shoulder) anchorages shall withstand separate belt loop loads of 13,344 N (3,000 Lbf) each applied with a body block to the upper torso and pelvic belt anchorages.	Type A (lap and shoulder) anchorages shall withstand separate belt loop loads of 13,500 N [+ or - 200 N] (3,035 Lbf) each applied with a body block to the upper torso and pelvic belt anchorages. For vehicle categories M3 and N3 the belt loop load 13500 N shall instead be 4500 N. For vehicle categories other than M1, N1, M3 and N3, belt loop load shall be 6750 N.	Type A (lap and shoulder) anchorages shall withstand separate belt loop loads of 13,500 N [+ or - 200 N] (3,035 Lbf) each applied with a body block to the upper torso and pelvic belt anchorages. For vehicle categories M3 and N3 the belt loop load 13500 N shall instead be 4500 N. For vehicle categories other than M1, N1, M3 and N3, belt loop load shall be 6750 N.	Basically the same for light vehicle categories M1 and N1. Considerable differences for other vehicle categories.			

NOTE: The term "anchorages includes all belt attaching hardware, attachment bolts, and seat/track structure, if belt system is attached to the seat, for both FMVSS and ECE.

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FMVSS 210
76/115/EEC, ECE R14-03
SAFETY BELT ANCHORAGES

ITEM	*FMVSS*	*EU*	*ECE*	*TECHNICAL DIFFERENCES IN REGULATIONS*	*PERFORMANCE DIFFERENCES FOR PRODUCTS*	*PRODUCT IMPACT*	*SAFETY BENEFIT*
STRENGTH REQUIREMENTS CONTINUED	Type 1 (Lap only) anchorages shall withstand belt loop load of 22,240 N (5,00 Lbf) applied with body block to pelvic belt anchorages.	Type B (Lap only) anchorages shall withstand belt loop load of 22,250 N *+ or - 200N* (5,002 Lbf) applied with body block to pelvic belt anchorages. For vehicle categories M3 and N3 the belt loop load 22250 N shall instead be 7400 N. For vehicle categories other than M1, N1, M3 and N3, belt loop load shall be 11100 N.	Type B (Lap only) anchorages shall withstand belt loop load of 22,250 N *+ or - 200N* (5,002 Lbf) applied with body block to pelvic belt anchorages. For vehicle categories M3 and N3 the belt loop load 22250 N shall instead be 7400 N. For vehicle categories other than M1, N1, M3 and N3, belt loop load shall be 11100 N.	Basically the same for light vehicle categories M1 and N1. Considerable differences for other vehicle categories.			
	If a seat belt anchorage of the Type 1 or 2 is attached to the seat /track assembly, then a load equal to 20 times the weight of the seat/track assembly is also applied simultaneously on the seat/track assembly while the body block load is applied.	If anchorages are on the seat structure, force equal to 20 times the weight of the complete seat is to be applied horizontally and longitudinally through the center of gravity of the seat.	If anchorages are on the seat structure, force equal to 20 times the weight of the complete seat is to be applied horizontally and longitudinally through the center of gravity of the seat.	Basically the same for light vehicle categories M1 and N1. Considerable differences for other vehicle categories.			
		For vehicle categories M3 and N3 the supplementary force equal to 20 times the mass of the complete seat shall instead be 6,6 times the mass of the complete seat. For vehicle categories M2 and N2, the supplementary force shall instead be equal to 10 times the mass of the complete seat.	For vehicle categories M3 and N3 the supplementary force equal to 20 times the mass of the complete seat shall instead be 6,6 times the mass of the complete seat. For vehicle categories M2 and N2, the supplementary force shall instead be equal to 10 times the mass of the complete seat.				

ITEM	FMVSS	EU	ECE	TECHNICAL DIFFERENCES IN REGULATIONS	PERFORMANCE DIFFERENCES FOR PRODUCTS	PRODUCT IMPACT	SAFETY BENEFIT
STRENGTH REQUIREMENTS CONTINUED	The vehicle structure and the seat belt assembly anchorages must sustain simultaneous loads of all lateral seat positions.	All anchorages of the same group must be tested simultaneously.	All anchorages of the same group must be tested simultaneously.	Basically the same.			
	Seats must be placed in the rearmost position.	Seats must be in driving position or worst case conditions with respect to the strength of the system. Seat back angle as close as possible to 25 degrees. For vehicle categories other than M1 and N1, the seat back angle shall instead be as close as possible to 15°. These angles apply unless otherwise specified by the vehicle manufacturer.	Seats must be in driving position or worst case conditions with respect to the strength of the system. Seat back angle as close as possible to 25 degrees. For vehicle categories other than M1 and N1, the seat back angle shall instead be as close as possible to 15°. These angles apply unless otherwise specified by the vehicle manufacturer.	Rearmost seat position vs EEC/ECE 14 with seat in position of driving.	No apparent difference.		
	Load is applied to an upper torso and/or pelvic body block at an angle of 10 + or - 5° above the horizontal in a plane parallel to the longitudinal centerline of the vehicle.	Specified tractive forces must be applied in a forward direction by means of specific traction devices reproducing the upper torso and pelvic portions of the human body, at an angle of 10 degrees (plus/minus 5) above the horizontal, in a plane parallel to the median longitudinal plane of the vehicle.	Specified tractive forces must be applied in a forward direction by means of specific traction devices reproducing the upper torso and pelvic portions of the human body, at an angle of 10 degrees (plus/minus 5) above the horizontal, in a plane parallel to the median longitudinal plane of the vehicle.	Direction of the force is the same			
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FMVSS 210
76/115/EEC, ECE R14-03
SAFETY BELT ANCHORAGES

ITEM	*FMVSS*	*EU*	*ECE*	*TECHNICAL DIFFERENCES IN REGULATIONS*	*PERFORMANCE DIFFERENCES FOR PRODUCTS*	*PRODUCT IMPACT*	*SAFETY BENEFIT*
STRENGTH REQUIREMENTS CONTINUED	Load rate is such that the load requirement must be reached in not more than 10 seconds, and is then maintained for 10 seconds.	Full application of load requirement as rapidly as possible, but not less than 0.2 seconds.	Full application of load requirement as rapidly as possible, but not less than 0.2 seconds.	FMVSS 210 is more stringent because more time is allowed for metal structure to rupture.	EEC/ECE load rate is more realistic to the accident scene conditions, however, FMVSS 210 is the more conservative test condition.		
	Test Results-- permanent deformation or rupture of a seat belt anchorage or its surrounding area is <u>not</u> considered to be a failure, if the required force is sustained for at least 10 seconds. No specific distortion limits.	Test Results--permanent deformation, including rupture or breakage of any anchorage or surrounding area shall not constitute failure if the required force is sustained for a period of at least 0.2 seconds, and deformation of minimum lateral spacing for lower anchorages and lower horizontal plane of "permitted area" for upper belt anchorages shall not be exceeded.	Test Results--permanent deformation, including rupture or breakage of any anchorage or surrounding area shall not constitute failure if the required force is sustained for a period of at least 0.2 seconds, and deformation of minimum lateral spacing for lower anchorages and lower horizontal plane of "permitted area" for upper belt anchorages shall not be exceeded.	Sustained force of 10 seconds vs 0.2 seconds. Also deformation of lower anchorage must still meet "permitted area" in lateral direction, and deformation of upper anchorage must not extend below lower limit of "permitted area" even after conducting pull test.	FMVSS requirement is more conservative because the load requirement could be reached but to maintain that load for 10 seconds could allow metal structure -- which had developed a partial tear, to separate in the longer period Distortion in the directions limits are not typically a compliance issue. EEC/ECE lower anchorages are limited to how much lateral distortion, and upper anchorages there is a limit to how low the guide distortion is allowed.		

ITEM	*FMVSS*	*EU*	*ECE*	*TECHNICAL DIFFERENCES IN REGULATIONS*	*PERFORMANCE DIFFERENCES FOR PRODUCTS*	*PRODUCT IMPACT*	*SAFETY BENEFIT*
ANCHORAGE LOCATION	A) Type 1 and pelvic portion of Type 2: The nearest point (take-off point) of the belt hardware attachment shall be in a zone between 30 and 75° from horizontal in side view. (Origin of zone is dependent if fixed or moveable)	*M1 vehicles only:* *A) Lower belt anchorages shall be located between* *-front seat* *30 and 80 (non buckle)* *45 and 80 (buckle)* *or for constant angle* *60 + or- 10* *-rear seat* *30 and 80* from the horizontal in the side view. with the origin of the zone extending from the "seating reference point."	*M1 vehicles only:* *A) Lower belt anchorages shall be located between* *-front seat* *30 and 80 (non buckle)* *45 and 80 (buckle)* *or for constant angle* *60 + or- 10* *-rear seat* *30 and 80* from the horizontal in the side view. with the origin of the zone extending from the "seating reference point."	Basically the same	Expect comparable performance of occupant kinematics		
	Two adjacent belts to one anchorage.	Two adjacent belts may be attached to one anchorage.	Two adjacent belts may be attached to one anchorage.	Same	Typically anchorage lateral locations are designed to the width of a 50%ile -- which is similar to EEC/ECE requirement.		
	For seat anchorage location shall be 165 mm apart (6.5 inches) apart.	Lateral location shall be 350 mm (13.8 inches) apart and no less than 120 mm (4.7 inches) to the median longitudinal plane of the seat.	Lateral location shall be 350 mm (13.8 inches) apart and no less than 120 mm (4.7 inches) to the median longitudinal plane of the seat.	EEC/ECE lateral anchorage location is significantly more design restrictive.	None. Anchorages must meet test strength requirements with adjacent belts loaded simultaneously.		
	B) Type 2 Upper torso portion with seat adjusted full rearward and downward and seat back in most upright position, anchorage shall be located within specified zone.	B) "Effective upper belt anchorage" (point where belt leaves guide) shall be located in specified zone identified as "permitted area."	B) "Effective upper belt anchorage" (point where belt leaves guide) shall be located in specified zone identified as "permitted area."	EEC/ECE anchorage zone allows belt guide anchorage to be significantly farther forward than FMVSS 210.	Load performance of anchorages are more dependent on structure than location.		

ITEM	*FMVSS*	*EU*	*ECE*	*TECHNICAL DIFFERENCES IN REGULATIONS*	*PERFORMANCE DIFFERENCES FOR PRODUCTS*	*PRODUCT IMPACT*	*SAFETY BENEFIT*
ANCHORAGE LOCATION EFFECTIVE BELT ANCHORAGE POINT A) FOR LOWER BELT ANCHORAGES	Nearest contact point of belt with attaching hardware attached to seat or vehicle structure.	Point where belt (strap) is attached to rigid part (Attaching hardware)	Point where belt (strap) is attached to rigid part (Attaching hardware)	Basically the same.			
B) FOR UPPER TORSO OR STRAP GUIDE D-RING	Anchorage point (where load is transferred to vehicle structure) is considered as the bolt attaching point.	Point where belt (strap) leaves the guide.	Point where belt (strap) leaves the guide.				
OWNER'S MANUAL INSTRUCTIONS	Owner's manual shall include explanation that child restraints systems are designed to be secured by lap belt portion of a Type 2 belt, and have a statement that children are safer in a rear seat position rather than in a front seat position.	Vehicle operating instructions require no statement for child restraints systems, however, the location of all anchorages and Type of belts intended for use are to be stated.	Vehicle operating instructions require no statement for child restraints systems, however, the location of all anchorages and Type of belts intended for use are to be stated.	EEC/ECE requires no specific child restraint instructions.			
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ATTACHMENT 3
INTERNATIONAL COORDINATED PRE-REGULATORY

INTERNATIONAL SAFETY RESEARCH COORDINATION

Committee	Responsibilities	Meeting Frequency	Membership	Chair	Support Staff (Secretariat)	Term
Research Steering Group made up of government research representatives	Identify Issues Establish Priorities Create/Monitor working groups Set Milestones, Set Schedules Disseminate results Establish Budget	At least annually	Senior government officials	Rotating	Provided by Chair's organization	Open
Working Group(s) comprising recognized public and private sector "experts" for each subject area	Define Tasks and methodology Assign tasks Agree work statements Monitor process Report results	As needed, At least semi-annually	Government specialists Private sector specialists	Agreed Center of Responsibility for the particular subject area	Provide by Chair's organization	Duration of assignment

Potential International Research Projects to Support Regulatory Harmonization

Functional Equivalence -- Develop a project to identify technical and performance differences between selected existing FMVSS and ECE/EU Regulatory requirements on the same aspects of motor vehicle systems, and determine the significance of the performance differences with respect to motor vehicle safety performance. (As an example: FMVSS 209 and ECE R16 both prescribe requirements for safety belts that results in unique safety belts for U.S. and Europe for the same type vehicle. Research project should define the safety benefit, if any, resulting from the performance differences between these two regulatory requirements.)

Lighting -- Develop a project to determine traffic targets and maneuvers that need to be seen and recognized that could form the basis for a performance based common regulation on vehicle lighting.

Side Impact -- Develop a project for the next generation of side impact testing, including dummy development and injury tolerance criteria.

Frontal Impact -- Develop a project for globally acceptable frontal impact configuration.

Child Dummy -- Develop a project for a globally acceptable child dummy for child restraint testing.

Seating System, Rear Impact Performance -- Develop a project to determine the cause of injuries resulting from rear impacts that could form the basis for a performance based common regulation on seat strength and head restraint design.

Accident Data Analysis -- Develop a project to define a common procedure for gathering accident data and uniform analysis.

Glazing -- Coordinate global research on glazing performance requirements.

Development of Analytical Models -- Math model development and validation.

CONCLUSION

Compliance with multiple regulatory frameworks is fundamentally inconsistent with free trade in a global automobile market. The concept of global harmonization of motor vehicle regulations has been endorsed around the world. The time is right for new thinking and bold action. Industry is ready to work in partnership with government regulators to advance harmonization of regulations from a concept to reality.

ACKNOWLEDGMENTS

This paper is based on the working papers prepared for the April 10-11, 1996, Transatlantic Automotive Industry Conference on International Regulatory Harmonization. In particular, the authors would like to recognize the contributions of the experts from the European Automobile Manufacturers Association (ACEA) and the American Automobile Manufacturers Association (AAMA) whose position papers on process and safety provides the content of this paper.

Commission of the European Community

Herbert Henssler
Head of Sector

World-wide harmonisation of technical regulations in the automobile sector is an objective to which the European Commission is strongly committed. This has been highlighted at the recent Transatlantic Automobile Industry Conference in Washington. To the greatest extent possible the development of new regulations as well as the evolution of existing regulations affecting the design and construction of motor vehicles should, in the future, take place on a world-wide basis and no longer nationally or regionally. Not only would our industries benefit from this principle but so would customers who would be able to acquire motor vehicles offering a high standard of safety and environmental protection at reasonable cost.

As to the best forum for the establishing global regulations we think that the UN/ECE WP29 offers the best perspective and the revised 1958 Agreement represents the appropriate legal framework for such regulations. No other international Agreement or organisations appears to be at hand which could become operational for such regulatory initiatives in the near future.

Therefore, the EC intends to adhere to the revised agreement as soon as possible in order to establish a legal link between the ECE regulations and its own type-approval procedure for motor vehicles.

We are of course prepared to envisage additional amendments to the Agreement in order to transform WP29 to an even broader platform for establishing international regulations. These amendments should be limited to what is necessary to attract the wider membership which we all desire, without jeopardising the achievements of the Agreement in its present form as far as the international quality of the regulations and the reciprocal recognition of approvals granted in accordance with these regulations are concerned.

The latter aspect is of paramount importance to the EC. It is not worthwhile to create harmonised regulations if there are no rules relating to the implementation of such regulations, in other words, to establish the rights and obligations of the contracting parties to a newly revised agreement in relation to the harmonised regulations they chose to apply.

In complying with its objective of an internal market without barriers to trade the EC has acquired considerable experience in harmonising technical regulations in the automobile sector and is prepared to share it with its partners in the world. The 15 EC Member States have learned to live with harmonised uniform regulations which substitute for their previous national regulations and they do not consider this an unacceptable interference with their sovereignty, nor a diminution of safety or environmental protection.

However, global harmonisation will not be achieved at once: a stepwise approach appears necessary. Agreements on mutual recognition of regulations which are functionally equivalent could be a useful interior step. More promising in the longer term appears to be the creation of a common scientific basis for future harmonised regulations. The International Harmonised Research Agenda is a very positive step in this direction and the ESV Conference an excellent contribution to this objective.

United States

Philip A. Hutchinson, Jr.
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Association of International Automobile Manufacturers, Inc.

ABSTRACT

In order to achieve harmonization of automobile standards it is necessary to involve the World Trade Organization (WTO). The WTO has the stature and authority to coordinate the interests of concerned governments to achieve the harmonization goal. Similarly, a private organization should be identified to coordinate the interests of national automobile trade associations and standards writing authorities so that all interested parties have an opportunity to participate in harmonization decisions. The International Organization of Motor Manufacturers (OICA) is proposed as a candidate for this endeavor provided it recognizes motor vehicle manufacturers associations interested in participating. Ongoing regional efforts to achieve harmonization should be expanded to include a global automotive perspective.

INTRODUCTION

Over the past twenty years, the automobile industries of the world have evolved into a vast, global enterprise. Today, few industries are as internationalized. This meeting, the 15th International Technical Conference on the Enhanced Safety of Vehicles, includes representatives from virtually every auto-producing country in the world, and is verification of our industry's global status.

Components and materials from around the globe go into every motor vehicle. General Motors, Ford and Chrysler manufacture vehicles in Europe for export to Japan. The largest exporters of automobiles from the United States to Japan are Honda, Toyota and Nissan. In other parts of the world - China, South America, India, Viet Nam - the auto industry is poised for a huge expansion.

At the same time this globalization of our industry has occurred, we have witnessed the development of the socially responsible automobile. Market forces did not respond to this need, so governments stepped in and required automakers to manufacture cars and trucks that met higher safety and environmental standards. The problems that we must deal with here today were created because these standards were established on a national basis, by national governments, without regard for the growing globalism of the auto industry.

Given the global nature of our industry, the maturation of the regulatory system in the industrialized nations, and the competitive demands of the marketplace, we find ourselves today in a situation where there are many conflicts between the official requirements imposed on manufacturers doing business in different countries, the societal benefits conferred upon consumers and the corresponding costs imposed.

It is obviously in the consumer's interest to achieve essential social benefits at the least cost. To attain this, harmonization of motor vehicle safety and environmental standards is a necessity.

In an automobile industry that has become increasingly global, that manufactures and sells its product in more than a hundred nations, from Abu Dabai to Zimbabwe, achieving this goal now takes a position of almost unchallenged priority.

For more than thirty years, several nations with cooperation from some automakers, struggled to create a system of harmonized standards.[1] The methods they chose to achieve this laudable goal was to engage in point-by-point discussions of differing national standards.

I suggest this approach will never provide the global harmony that we seek. New and conflicting standards and testing materials are being promulgated faster than we can reach agreement on existing differences. It has taken us 18 years to achieve a harmonized braking standard just between the United States and the European community!

Needed: A Single Global Forum

To achieve a global harmonization of motor vehicle standards, we must first establish a single forum that will create a process for harmonizing safety and environmental standards, a method for the adjudication of disagreements and for national enforcement of agreed upon, harmonized standards. There is a fundamental need to link all the activities now under way to a final coordinating authority that has the active participation of all the concerned governments, manufacturers and standards-setting organizations.

Even now, efforts are being made on a regional level to bring specific conflicting national standards into greater harmony.[2] This may simplify compliance matters for some manufacturers, but the net result is still dealing with global regulatory differences on a piecemeal basis. Such regional arrangements also lend themselves to charges of establishing technical barriers to trade because discussions do not include all interested parties[3] and may be designed to accommodate participants, but not others, on a most favored nation or equal national treatment basis.

While the United Nations Economic Commission for Europe Working Party 29 has diligently labored to promote harmonization, this body would need to broaden its global make-up and outlook to fill the role needed to set us on the path to universal harmonization. Provided it can expand its focus and assure broader representation of all parties, Working Party 29 could remain the operating forum for internation negotiations on specific issues.

WTO: A Truly Global Authority

The growing globalism of world commerce has recently brought into being a new organization that is the proper forum for achieving a truly universal agreement on automotive standards harmonization. It is the World Trade Organization,[4] designed to promote open markets and commerce, to which all of the major automotive-producing nations have subscribed, and which has the stature and the authority to formulate the procedures that will bring us positively toward the goal of harmonization.

Within WTO are two operating committees that are appropriate forums for harmonization of automobile standards for both safety and environmental purposes. They are the Committee on Technical Barriers to Trade and the Committee on Trade and the Environment.[5] A proposal from the automobile producing nations that these committees accept the responsibility for coordinating development of harmonized automotive standards would bring us nearer to realization of the goal we have pursued for so many decades.

A Parallel Industry Forum

There is a special need for a single forum on harmonization on the industry side too. Such a forum must be international in scope and all-inclusive in its representation. There are a number of national trade associations grappling with the problems and opportunities of harmonization.[6] Perhaps the International Association of Motor Manufacturers, known as OICA, could serve as such a focal point for industry participation, providing it becomes truly international in its outlook and opens its ranks to all established automotive trade associations whose members are affected by the differing national regulatory and legislative programs.[7]

AIAM proposes that the various government and industry representatives participating in this ESV Conference agree to call for an international summit conference later this year for the purpose of requesting the World Trade Organization to coordinate and expedite the harmonization of automotive safety and environmental standards and a global acceptance of a policy of mutual recognition of certification procedures. Such a meeting could be held under the auspices of several interested trade associations. It should involve representatives of the International Organization of Motor Manufacturers and regulatory bodies, as well as standards setting organizations.

Harmonization Essential To Controlling Costs

All manufacturers today are struggling to control costs, to maintain prices that will overcome consumer resistance. We all are uncomfortably aware that automobile prices over the past decade have increased substantially more than consumer buying power. In America, the average automobile now costs more than $20,000. This means that a worker labors longer to purchase an average car than he or she did five years ago.

A factor in this price escalation has been the cost of manufacturing vehicles to differing standards for different countries. It is also an impediment to the expansion of world trade in automobiles, which is costly to manufacturers, as well.

In the long run, it is the consumer who is paying for this Balkanization of safety and environmental standards. To keep our products within the range of affordability and to maintain a healthy growth for our industry, we must apply the same creative energy and intensity to bringing about global harmonization as we have to creating today's efficiency and productivity in the manufacturing process.

The first step is to elevate the issue to the highest international levels, to go beyond regionalism, and to recognize that harmonization requires the active participation of all the automobile-producing nations, and a parallel cooperation among *all* automakers.

Two more steps are necessary:

1. Obtaining the endorsement of the WTO.
2. Facilitating the participation of interested international automotive societies under one umbrella organization to support the concept.

If we are serious about harmonization, we should resolve at this ESV Conference to elevate the issue to the required level of international attention -- ie., to place harmonization within the forum of the World Trade Organization -- and to approach its resolution on an all inclusive, industry-wide basis.

FOOTNOTES

1. UN-ECE Working Party 29, organized June 20, 1959, for the purpose of "The adoption of uniform technical prescriptions for wheeled vehicles..." Twenty-eight countries are a party to the agreement; however, the United States and Japan are not parties.

2. A Transatlantic Automotive Industry Conference on International Regulatory Harmonization was held in Washington, D.C., April 11, 1996, under the auspices of the American Automobile Manufacturers Association (AAMA) and the Association of European Automobile Manufacturers (ACEA).

3. There are 12 automobile manufacturers building cars in the United States. AAMA represents three of the twelve manufacturers.

4. The WTO officially entered into force on January 1, 1995, pursuant to the Final Act Embodying the Results of the Uruguay Round of Multilateral Trade Negotiations, December 15, 1993, and the Marrakesh Agreement Establishing the World Trade Organization adopted by ministers representing 124 Governments and the European communities on April 15, 1994.

5. The WTO's Committee on Technical Barriers to Trade was established as part of the Uruguay Round Agreement on Technical Barriers to Trade. At the Marrakesh meeting of April 12-15, 1994, the Ministers decided to direct the first meeting of the General Council of the WTO to establish a Committee on Trade and Environment open to all members of the WTO. The ministerial decision establishing the committee directed it to address several matters, including: "the relationship between the provisions of the multilateral trading system and requirements for environmental purposes relating to products, including standards and technical regulations...".

6. Several of the major automotive trade associations concerned with harmonization of standards include:

United States

American Automobile Manufacturers Association (AAMA)
Association of Int'l Automobile Manufacturers (AIAM)

Europe

Association of European Automobile Manufacturers (ACEA)
Verbond der Automobilindstre (VDA)
Society of Motor Manufacturers and Traders (SMMT)
Comite' des Constructeurs Francais d'Automobiles (CCFA)

Asia

Korean Automobile Manufactures Association (KAMA)
Japan Automobile Manufacturers Association (JAMA)

International

Int'l Organization of Motor Vehicle Manufacturers (OICA)

7. Only three of the twelve automobile manufacturers building cars in the United States are represented in OICA through AAMA's membership.

United States

George L. Parker
Philip A. Hutchinson, Jr.
Association of International Automobile Manufacturers, Inc.

ABSTRACT

Motor vehicle safety regulation in industrialized countries has followed the increased road accidents and fatalities resulting from increasing road traffic. These regulations have been developed without a focus on harmonization between countries except in regions with close economic ties. However, economic ties are increasingly important in the expanding global economy. As countries emerge economically and their populations demand more mobility, and as automobile manufacturing and sales

become more global, harmonizing safety standards to remove non-tariff trade barriers and to lower costs to consumers becomes more important. There is little current activity to harmonize existing and future safety standards in spite of the almost universal acknowledgment of its importance. However, there is activity to form alliances to promote harmonization and to establish a forum and a process to achieve harmonization. The United Nations Economic Commission for Europe Working Party 29 appears to be the best forum for harmonization if it can become truly international in its organizational and representational principles. For safety research leading to new safety regulations, either the Enhanced Safety of Vehicles governing body or the Groupes des Rapporteurs supporting WP 29 could be used. There are many ways in which harmonization of safety standards could be promoted and achieved. The first step is a commitment. Not to make the commitment means higher costs for consumers with no safety benefit and burdens for automakers.

INTRODUCTION

The increase in road traffic and the ever-growing number of vehicles on the road combined with the high number of traffic accidents and the resulting injuries and fatalities have resulted in public demand for both motor vehicle safety programs and highway safety programs in many countries. The motor vehicle safety programs have taken the form of regulatory requirements for safety equipment and minimum levels of safety performance in specified test conditions along with a demand for improved safety of vehicles from manufacturers beyond the regulatory requirements. Highway safety programs have included better roads, reduced roadside hazards, better roadside collision barriers and breakaway sign poles, stricter driving under the influence of alcohol laws, and mandatory safety belt and child restraint use laws.

This increasing demand for improved motoring safety in many countries and the resulting safety programs began in the 1960s. In the U.S., the Motor Vehicle Safety Act of 1966 resulted in the first vehicle safety regulations in 1967. Since that time, over 50 safety regulations have been promulgated in the U.S. A similar trend has taken place in other countries. For example, since the early 1970s, vehicle safety regulatory activities in Germany have increased by a factor of four and have doubled in Japan. Since the mid-1980s, vehicle safety regulatory actions in Sweden have increased by a factor of three and in Australia by a factor of more than six.

At the present time, populations of all countries of the world are demanding more personal mobility. At the same time, political barriers are almost nonexistent, more countries are emerging economically, and workforces are becoming more sophisticated. This is particularly true in the Asian and Pacific regions. These conditions have led the motor vehicle industry to become increasingly global in both manufacturing and sales. Regardless of where motor vehicles are manufactured, they are exported to most parts of the world. The international manufacturers that are members of the Association of International Automobile Manufacturers in the U.S. sell vehicles in more than 130 countries from Abu Dabai and Andorra to Zambia and Zimbabwe.

National boundaries are disappearing for both manufacturing and sales. In some cases, manufacturers establish manufacturing capabilities in countries that are major markets, but not their home countries, and then export back to the home market or to other parts of the world. Also, manufacturers are rapidly moving to "world" cars that are developed to meet the needs of consumers in a number of markets. Perhaps the best example of the global nature of the motor vehicle industry is Canada. About 85 percent of the vehicles made in Canada are exported, and about 70 percent of the vehicles sold in Canada are imported.

Because of the increasingly global nature of the automobile industry and recognizing that the human beings safety standards are designed to protect are the same from country to country, safety standards can be much more harmonized than they are now. Differences among safety standards worldwide should only reflect differences in driving environments and automotive fleet composition in different countries. For manufacturers, the cost of research, designing vehicles, testing vehicles, and manufacturing to different safety standards is a cost that is not justified when the needs of consumers are considered since these costs are passed on to consumers. Consumers are willing to pay for safety protection, perhaps now more than in any time in the past, but they are paying extra costs because of the inefficiency of having to comply with different safety standards.

DIFFERENT INTERNATIONAL REGULATIONS

As previously stated, there may be legitimate reasons for some differences in safety standards because of driving environments and vehicle fleet differences. For the most part, crash avoidance standards can be identical or almost identical since they are not dependent upon driving and fleet differences. One exception would relate to higher speed driving in some countries. This might require higher test speeds for tire and brake performance standards, but the basic requirements for tire and brake performance, and the basic test procedures can be the same. Standards for lighting and signaling, controls and displays, windshield wiping/washing and most other crash avoidance safety standards can be essentially the same.

Crashworthiness standards might be somewhat different to reflect fleet differences between countries. For example, vehicles in the U.S. tend to be heavier and larger on average than in other countries. Thus, a crash test using a moving deformable barrier would use a heavier and larger barrier in the U.S. However, the basic approach to the test, the test dummy, and the injury criteria should be the same.

There are numerous instances in which the vehicle safety regulations of different countries or regions are not compatible and require manufacturers to design and build several variations of vehicles. A good example is the lighting standards of various regions. There are many instances of incompatible equipment requirements such as the U.S. requirement for center high-mounted stop lamps, incompatible photometric requirements, and different test procedures for demonstrating compliance with performance requirements. For many years, the U.S. only permitted sealed beam headlamps, and, when replaceable bulb headlamps were first allowed, the dimensional and performance requirements were incompatible with those of Europe.

Frontal Impact Standards

In the area of crashworthiness safety, there is one instance in which compatibility of standards has been approached, but there are important differences and emerging divergence. That instance is frontal impact crashworthiness protection. In the U.S., Federal Motor Vehicle Safety Standard (FMVSS) 208, Occupant crash protection, requires that vehicles demonstrate the protecting effects of structure and restraint systems in a 48 km/h impact into a rigid barrier. The test procedure calls for use of unrestrained Hybrid III dummies in the driver and the right front passenger positions. The injury criteria are HIC $\leq$ 1000, chest acceleration $\leq$ 60 g, femur force $\leq$ lOkN, and chest deflection $\leq$ 75 mm. After September 1, 1998, all passenger cars and light trucks, vans, and utility vehicles must meet this requirement using driver and right front passenger air bags. In Canada, the same test procedure is being proposed but with a belted dummy and HIC is replaced by a head acceleration limit of 80 g and the chest deflection limit is proposed to be 50 mm. In Australia, the same test procedure is used with the same injury criteria as the U.S., but the dummies are restrained by safety belts. Japan also will use the same test procedures and injury criteria as FMVSS 208 but with belted dummies. Thus, the U.S. is unique in requiring an unrestrained dummy test.

In Europe, there is currently no dynamic crash test requirement for frontal impacts. However, there is a proposal for a 56 km/h offset impact into a fixed deformable barrier with 40 percent overlap between the deformable barrier face and the vehicle front surface. Modified Hybrid III dummies would be used and the injury criteria would include requirements for neck shear and bending, tibia shear and bending, tibia axial load, and knee displacement in addition to the usual head, chest, and femur requirements. The chest requirement would be the viscous criterion (V*C) instead of chest acceleration. There is no plan in Europe for a full barrier frontal impact as in the U.S., Canada, and Japan. The Insurance Institute for Highway Safety (IIHS) in the U.S. and the New Car Assessment Program (NCAP) in Australia are using a similar offset test for consumer information purposes.

Thus, although there is some commonality of frontal impact test procedures in four countries, there are important differences that should be reconciled. Europe is diverging from the pattern established by the U. S., Canada, Australia, and Japan. An even more important potential conflict is the U.S. development of an offset test procedure that could become the basis for a future regulation. The research being pursued in the U.S. to develop an offset test is diverging from the offset test proposed in Europe and being used by IIHS and NCAP in Australia.

As stated above, in the U.S. and in many other countries, the primary frontal crash test is a 48 km/h full barrier impact. Real-world crashes are better represented by an offset test in which only a portion of the front of a vehicle is engaged. This is a good test of the structural integrity of the passenger compartment whereas the full frontal impact into a rigid barrier is a good test of a vehicle's restraint system. In the U.S., the National Highway Traffic Safety Administration (NHTSA) is developing an oblique offset test using a moving deformable barrier. The barrier would strike the stationary test vehicle at an oblique angle to the longitudinal axis of the test vehicle and engage only a portion of the front of the test vehicle. The dummy injury criteria would relate to head, chest, and lower extremity injuries. This test conflicts with the offset frontal test being used in Australia, being proposed in Europe, and being used in the U.S. by IIHS for consumer information purposes. NHTSA says it prefers its test because it is more likely to replicate life threatening injury situations than the offset test the rest of the world is using.

A major concern with each of these offset tests is that they could result in increasing the stiffness of the front structure of vehicles since only a portion of the front structure would be involved in the impact of the test, and that structure would have to provide protection against passenger compartment intrusion. A stiffer front structure would be more aggressive in side impact, rear, and large car/small car impacts in the real world. The result could be that frontal impact occupant protection benefits would be less than increases in injuries and fatalities in other crash modes. The development of an offset procedure and the promulgation of any regulations must take this into account.

Even though the offset test using the fixed deformable barrier is already being used, the test is not without criticism. There is concern that the test does not produce reproducible vehicle deformation patterns and that the deformation pattern produced does not replicate real-world offset crash deformation. The result is that air bag deployment and safety belt pretensioning onsets tuned for the test may not be optimum for real-world crashes. There also is concern that the injury criteria, especially those for the lower extremities, are not fully validated and may be inappropriate. Also, the Hybrid III lower extremities are not sufficiently biofidelic to be used to measure lower extremity injury potential. It appears that further research is needed to improve this offset procedure and the supporting dummy and injury criteria.

These offset tests being developed in the U.S. and proposed in Europe and used elsewhere are sufficiently different that they would require manufacturers to design for each one. This would, once again, raise the cost of compliance to manufacturers and raise the price of the vehicle for consumers. Fortuitously, neither of these tests has been adopted as regulations yet, so there is time to harmonize. However, the European Union (EU) has taken the first steps to adopt the fixed deformable barrier offset tests.

Side Impact Standards

Another example of divergence is the dynamic side impact tests developed in the U.S. and Europe. The U.S. test uses a moving deformable barrier with an energy-absorbing aluminum honeycomb front impacting surface. The moving deformable barrier impacts the side of the test vehicle with the wheels in a crabbed configuration to represent both vehicles moving. It uses an instrumented dummy called SID, for side impact dummy, to assess an injury criterion called the Thoracic Trauma Index, which is based on peak rib accelerations. A maximum spine acceleration level also is specified. The European test uses a smaller barrier with a stiffer energy-absorbing impacting front surface. It impacts the test vehicle at a 90 degree angle. The dummy used is called Eurosid, for European side impact dummy, and the injury criteria are for head accelerations, rib deflection, thoracic viscous criterion (V*C), and abdominal peak force. The only difference which is defensible is the different deformable barrier to represent the different fleet composition. There are no good reasons for the other differences. It seems that the scientific communities that developed these two tests could have collaborated to develop more harmonized tests. There are other differences between safety standards and other opportunities for cooperation besides those already mentioned.

Developing the human surrogates for crash testing and the injury criteria by which crash test results are assessed are both very expensive. Developing dummies requires extensive design work, computer modeling, instrumentation design, test evaluations, redesign, reevaluation, and continued refinement until a dummy is ready for use in crash test regulations. This process can take 10 years and more. Injury criteria are developed by extensive injury tolerance studies covering a range of simulated crash configurations and human parameters. Testing in support of developing injury criteria is expensive and time consuming, and analysis of test results leading to injury criteria is a combination of art and science and requires extensive experience. If only to preserve scarce safety research resources and to pool research capabilities, the world's biomechanics community should work cooperatively to develop dummies and injury criteria for crash tests. The result would be harmonized injury criteria and dummies. At the present time, there is an effort underway to develop an advanced dummy for frontal crash testing. This effort enjoys widespread international cooperation, which may lead to a single next-generation dummy.

The goal in developing new safety tests does not have to be absolute uniformity from country to country. There should be enough commonality of test procedures and overlap of performance requirements that a manufacturer can meet all standards with a single design. The spirit of compromise should prevail when new tests are developed by different countries because the goals of protecting the public are the same. To develop different tests, different dummies, and different performance requirements makes it difficult for any country's automotive industry to compete in a world marketplace.

HARMONIZING SAFETY STANDARDS

With regard to harmonizing existing standards, the prospects are not very promising if history has a lesson to teach. The effort to harmonize the European and U.S. braking standards began in 1978 and is only now complete. At least the two braking standards are now harmonized, and, even if it takes a long time to harmonize other existing standards that are the most costly for manufacturers, the effort should be undertaken. An alternative is that countries begin an effort to establish legal reciprocity of standards. This would be based on the premise that, in developed nations, vehicles designed to meet the home country's safety standards have equivalent levels of safety compared to vehicles in other countries. Thus, different countries could agree to accept certification to a vehicle's home country safety standards. Some exceptions would have to be made for such items as air bags and center high-mounted stop lights which are required equipment in the U.S. but not in other countries. This same concept of reciprocity would apply to the certification scheme in different countries. Self-certification would be considered equivalent to type approval. It is especially important that economically emerging countries embrace the concept of reciprocity of standards, even before the major industrial countries reach this point. In other words, rather than adopting U.S., European, or Japanese safety standards, the economically emerging countries should recognize these safety standards as equivalent. It would be unfair to auto manufacturers planning to compete in economically emerging countries to carry the extra burden of having to meet different safety standards than those in effect in their home countries. Also, consumers in the economically emerging countries will benefit from greater choice and lower prices if auto manufacturers do not have the burden of different safety standards.

The first principle of harmonization is a commitment by all governmental entities developing safety standards that harmonization is to be achieved. This requires that high level officials in the governments issue directions to their appropriate elements that harmonization is an integral part of their activities. At present, it is not clear that all governments share the vision of worldwide harmonization to promote free trade. If harmonization was an integral part of governmental regulatory activities, the regulatory organization staff would be aggressive promoters of harmonization, would be the instruments of harmonization, and would find the proper forum and means to accomplish harmonization.

A forum and a process for harmonizing safety standards are necessary if the goal is to be achieved. This applies whether harmonizing means changing different standards to make them the equivalent or establishing reciprocity of standards. At the present time there are several regional harmonizing activities under way. The first and oldest is the United Nations Economic Commission for Europe (UN-ECE) Working Party 29 (WP 29) established by the 1958 agreement on Uniform Conditions of Approval and Reciprocal Recognition of Approval for Motor Vehicle Equipment and Parts. The activity of this group over the years has been the basis for uniformity of safety standards in Europe and reciprocity of type approval among the European countries.

Japan has recently notified the UN-ECE of its intention to sign the 1958 agreement, and Australia, China, South Korea, and South Africa have indicated their interest in signing. Developing countries that are establishing their own regulatory schemes are tending to adopt ECE regulations. Thus, it seems that the UN-ECE WP 29 is the appropriate international forum for the future advancement of regulatory harmonization.

The United States is not presently a signatory to the 1958 agreement, but submitted proposed revisions in 1994 that would facilitate the U.S. signing on. The proposal includes consensus voting and also provides for a "general registry of national technical regulations" and a "global registry of international technical regulations." A process is described for moving regulations from the general registry to the global registry. The goal is to promote harmonization by adopting as international technical regulations, those national regulations that are contained in the general registry. Contracting parties to this revised 1958 agreement would be encouraged to adopt regulations from these two registries. Because of some concerns with the U.S. proposal, a small working group consisting of representatives of the European Union, Japan, the U.S., Australia, and South Africa has been formed to develop a new working document for WP 29 on a proposed global agreement to change the 1958 agreement. This new working document may be presented at the June 1996 WP 29 meeting.

Two problems that have concerned the U.S. and other countries not presently signatories to the 1958 agreement are the regional focus of the WP 29 activities and the overrepresentation of European nations in the voting. To be truly international and representative of the various economic regions of the world, WP 29 activities and voting should be based on regional representation. Thus, perhaps, the EU should have a vote, North America, Asia, and other regions. Besides the European Union, there are other regional economic unions that could form the basis for regional representation on WP 29. Examples are the North American Free Trade Agreement, the Andean Pact, and the Asia-Pacific Economic Cooperation. Caucuses within these groups would establish regional voting preferences. The proposal for revising the 1958 agreement addresses this issue and proposes that decisions be made by consensus voting and that "regional economic integration organizations" can represent their members and have a number of votes equal to their number of members. This latter provision would be counterproductive to worldwide harmonization as organizations with the most members could dominate the voting.

There is one other governmental harmonization activity underway under the auspices of the Asia-Pacific Economic Cooperation. The Australian Federal Office of Road Safety is leading an effort to catalog the standards that apply in different countries as the first step in establishing a harmonization dialogue. There are three other related activities that have been established. First is an agreement between the American Automobile Manufacturers Association (AAMA) and the Japan Automobile Manufacturers Association (JAMA) to work together to achieve harmonization. The two groups have agreed to establish working groups of experts on harmonizing safety standards. Second is the Trans-Atlantic Business Dialogue (TABD), an informal organization of European and U.S. businesses that has harmonization of safety standards as one of its areas of activity. This harmonization would be based on functional equivalence and mutual recognition of standards and certification/approval and developing common standards. A meeting to develop a plan to promote these activities to the EU and the U.S. Governments was held on April 10 and 11, 1996. The harmonization process recommendations resulting from this meeting were to develop prior to a November 1996 TABD meeting 1) a process for agreeing upon functional equivalence of regulatory standards; 2) a process for mutual recognition of regulatory standards and certification procedures; 3) a plan for coordinating research, both by industry and government; and 4) the role and structure of the UN-ECE WP 29 as the forum for global regulatory harmonization. Recommendations on longer-term issues were to consider cooperation in the development of new testing procedures and regulations and coordination of views on emerging market regulations. Regarding safety, the recommendations were to 1) initiate a process to develop cooperative programs in the areas of common regulatory matters and regulatory research programs prior to the 15th ESV Conference; 2) mutual recognition of certain items currently regulated in Europe and the U.S.; 3) mutual recognition of functional equivalence for those requirements that mandate unique equipment design or performance but do not provide meaningful differences in motor vehicle safety, and 4) consideration of harmonizing other items. Finally, the AAMA and the Association des Constructeurs D'Automobile Européens (ACEA) are working to promote the same harmonization goals focusing on WP 29 as the harmonizing body. This effort also is linked to the Trans-Atlantic Business Dialogue.

These activities and organizations are all helpful in the effort to achieve harmonized safety standards. What is needed is the connection of governments to make the necessary regulatory changes. From the discussion above, it appears that WP 29 is the best focus if it can become truly international in its organizational and representational principles. If this is not possible, a new governmental harmonizing body should be created in which the free trade principles that make harmonization desirable are accepted by all parties.

HARMONIZING SAFETY RESEARCH

For future standards, the safety research community should establish a process and an authority for cooperation and collaboration on research. Cooperation in this case means sharing information and collaboration means sharing research tasks. A coordinating body should be put in place to begin the coordination and collaboration process. Hopefully, this body would be relatively free of political pressures and could develop the basis for harmonized future standards purely on scientific grounds.

In November 1995, the Administrator of NHTSA asked the international Governmental Focal Points Group that supports and promotes the biennial Enhanced Safety of Vehicles (ESV) Conference to prepare safety research priorities to be considered at the 15th ESV Conference. Also at the 15th ESV Conference discussions would be held to identify a lead country for each priority, to agree on a process, to agree on next steps, and to make an announcement on actions taken. This was a follow-up to a proposal put forth by NHTSA to this same group for a safety research harmonization activity. The Governmental Focal Points Group represents the various governmental safety research organizations of the major industrialized nations. The proposal was for sub-groups representing various research specialties to be formed under the top level group for coordinating safety research. Details were to be worked out beyond this overall concept.

If this activity is to be successful, research supporting potential new safety regulations should be coordinated through the groups formed. Ideally, research programs would be segmented into research projects that different countries would agree to undertake. Thus, for example, one country might agree to conduct any needed biomechanical research, another country might agree to be responsible for any dummy development, while another country might undertake to develop the test procedure. This activity would have to be highly coordinated to be successful, thus guaranteeing cooperation. For example, dummy development depends on both understanding injury modes and tolerances and knowing the crash mode, and test development relies on having a dummy to measure injury potential.

An alternative to this ESV Governmental Focal Points Group research management approach would be to expand the responsibilities of the Groupes des Rapporteurs that provide technical support for WP 29 activities to include coordination of research activities. This would ensure linkage of the research to the harmonizing authority.

SUMMARY

Whatever the organization and process for harmonizing current safety standards and future safety research that are established, there is a fundamental need to link all of the activities now underway to the final harmonizing authority. A number of loosely connected alliances are now being formed, and these alliances demonstrate the strong need for a harmonizing activity. What is now needed are an international forum and process to link strongly and combine these activities. This should be the current priority of this effort as harmonization cannot occur without strong linking and cooperation. WP 29 appears to be the first target for developing the harmonization linkages. If it is not possible for this group to expand its focus, then a new governmental group should be formed to proceed with the harmonization activities. Ideally, the research leading to new regulations should be linked within the same group that works toward harmonizing existing regulations. The proposal that NHTSA has put forward for cooperating and coordination through the Governmental Focal Points Group maybe should be redirected toward research cooperation managed through WP 29 or a new group.

There are many ways in which harmonization of safety standards could be promoted and achieved. The first step is a commitment. Not to make the commitment means higher costs for consumers with no safety benefit and burdens for automakers.

Japan

Takashi Shimodaira
Director, Engineering Planning Division
Ministry of Transport

Below is a brief introduction of the activities that the Japanese Ministry of Transport has carried out up to the present or plans to carry out in the future.

ACTIVITIES UP TO THE PRESENT

Establishment of Internationally Harmonized Standards and Promotion of Adopting these Standards.

Technical standards for motor vehicles differ among each country, and there are no international standards in the world. For this reason, Japan has been positively participating in the Working Party on the Construction of Vehicles organized under the United Nations Economic Commission for Europe (UN/ECE/WP29), in order to promote the establishment of internationally harmonized standards.

In addition, whenever a harmonized standard is established by ECE/WP29, Japan intends to adopt it as a domestic standard even before the European and North American countries. Japan has already adopted the following harmonized standards as its domestic standards:

(a) Braking performance of passenger cars (1993).
(b) Installation of lights (1996).
(c) Light distribution performance of headlamps (1996).

Introduction of Overseas Technical Standards

When planning to establish a new domestic standard or revise an existing one, Japan has considered that the new or revised domestic standard are harmonized with the overseas standards. At the same time, Japan is continuing its harmonization effort at bilateral expert meetings with the U.S. and EU.

In addition, with respect to the standards for window glass, seatbelts, seat installation strength and some other items, Japan considers that test results based on overseas standards are equivalent to test results based on domestic standards, although these standards are slightly different. Accordingly, since 1975, Japan has gradually accepted overseas standards as equivalent standards in the case of imported vehicles. To my knowledge, the European and North American countries have not done such equivalency treatment yet.

Japan's Unilateral Deregulation Measures

Japan's Government is carrying out its own deregulation measures on the basis of its unilaterally decided "Deregulation Action Program". The measures that were carried out include the improvement of diesel smoke test procedures and the relaxation of installation requirements for Heat Damage Warning Devices.

ACTIVITIES IN THE FUTURE

Basic Concepts

In order to promote the harmonization of standards, first of all, it is important that each country start discussions on the harmonization of those specific standards that will not greatly affect their traffic conditions to catch the understanding and support in their countries. Thereafter, once a harmonized standard has been established, each country should promptly adopt it as their domestic standard. And, as for the establishment of new standards which do not exist in any country, the establishment of harmonized standards is easier and should be undertaken on a priority basis.

Before starting the establishment or revision work for technical standards, each country normally conducts necessary investigations and research. The establishment of internationally harmonized standards can be promoted by international cooperation at this research stage. The past ECE/WP29 discussions on a harmonized standards for lateral collision have taught us that research cooperation is also a difficult task. Nevertheless, I think that research cooperation is beneficial to the promotion of harmonization and to the efficiency of standards establishment. Japan therefore intends to be an active participant in international harmonization project of research.

Specific Measures in the Future

In the future, Japan intends to continue its active participation in ECE/WP29 and become a member nation of the revised UN/ECE 1958 Agreement, in order to promote the establishment of international harmonized standards and promptly adopt any international harmonized standard as a domestic standard. Therefore, Japan is now preparing to make the necessary legal arrangements to accede the revised UN/ECE 1958 Agreement.

In addition, the activities to harmonize motor vehicle standards and certification systems are underway as part of the road transport harmonization project of the APEC Transport Working Group. Japan is a major member of this harmonization movement and, as example, the Japan Automobile Standards Internationalization Center (JASIC) is engaged in research into the existing automobile standards of the APEC region. With a view to promoting global harmonization in the future, APEC has the fundamental policy of increasing dialogue with the UN/ECE/WP29.

In the light of these circumstances, the Japanese government intends to continue its commitment to international activities for the harmonization of motor vehicle standards, respecting especially UN/ECE/WP29 as the central forum for the promotion of international harmonization.

United States

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International Harmonization - Functional Equivalence - MRA's (Mutual Recognition Agreements) - Global Regulations - GATT Uruguay Round - TBT (Technical Barriers to Trade) - WP29 - Globally Compatible Regulations - Transatlantic Business Dialogue - EU-US Summit - NAFTA Automotive Standards Council - APEC (Asia Pacific Economic Cooperation), etc. I've noticed that these words and acronyms are being used with increasing frequency around the world.

During the past year, we at NHTSA have been reinventing the agency, reviewing the work that is underway, streamlining processes to accomplish that work more efficiently, and, building partnerships with many interested organizations in an effort to innovate in the accomplishment of NHTSA's mission of preventing crashes, injuries and fatalities on our highways. During this work one of the recurring issues being brought to our attention was the issue of international harmonization of motor vehicle safety regulations. During the past six months, this particular issue has been the beneficiary of attention at very high levels in both industry and government.

International harmonization of motor vehicle safety regulations has been underway on a regional basis in Europe for more than thirty years, in North America for almost 30 years, and is being pursued in Asia. What is different today is that the regulated industry in all three regions has become a global one and the issue of harmonization of regulations has moved from being regional harmonization to global or world-wide harmonization. If we go back to the beginning of my remarks, you will see that all of the words or acronyms I mentioned are associated with either regional harmonization - WP29, APEC, NAFTA - or, global harmonization - GATT Uruguay Round, globally compatible regulations, Transatlantic Business Dialogue etc.

If we all agree that it is in our collective interest to work towards global harmonization of motor vehicle regulations or globally compatible regulations, then, we need to consider what must be done, how will it be done, who will do it, when it might be done, and where it will be done. I've left out the why it must be done because I am presuming that the world wide interest in safe vehicles, affordable vehicles and environmentally responsible vehicles, as well as free trade, jobs and economic well-being, answers that.

Whenever we speak of international - global - harmonization, the existing different regulations immediately come to mind. Not far behind is the observation that there is not very much that has been globally harmonized, although there has been some success in harmonization between regions. Finally comes the thought that we should cooperate to the maximum extent possible in the development of new regulations or the significant amendment of existing regulations. The rest of my remarks will be devoted to recent events on the subject of international harmonization, the agency's current thoughts and plans, and, finally and most importantly, a harmonized research agenda.

The subject of international harmonization has become an increasingly frequent subject of discussion during the last six months. The events that have prompted these discussions include: the 107th Session of the Working Party on the Construction of Vehicles (WP29) in Geneva, Switzerland (November, 1995); the first Transatlantic Business Dialogue Conference in Seville, Spain (November, 1995); the EU-US Summit in Madrid, Spain (December, 1995); and, the Transatlantic Automotive Industry Conference on International Regulatory Harmonization in Washington, D.C. (April, 1996). Each of these events led to a certain outcome or conclusions that I would like to summarize briefly here:

- At the 107th session of WP29, the United States stated its criteria for an agreement on the development of globally harmonized regulations, These criteria addressed both the process of harmonization as well as the rights of individual nations with respect to voting, adoption of global technical regulations, and accession to the agreement. The United States stated that, should such an agreement meeting the criteria be adopted, the United States would sign it.
- Also, at an informal meeting of delegations participating in this 15th ESV Conference and attending the 107th session of WP29, the United States proposed a process for arriving at a harmonized motor vehicle safety research agenda. Several aims were outlined, among them the development of regulations from the

same science and the reduction of duplicative research.

- The Transatlantic Business Dialogue in Seville reported 11 recommendations on the subject of Standards, Certification and Regulatory Policy, several of which were applicable to the automotive sector.
- Several of the above recommendations were incorporated generically in the New Transatlantic Agenda and Action Plan issuing from the EU-US Summit in Madrid. In particular, and I quote, "We will devote special attention to cooperatively developing and implementing regulations on vehicle safety requirements and on measures to reduce air and noise emissions. We will build on existing efforts aimed at facilitating international regulatory harmonization, taking account of our respective policies on safety and environmental protection, while recognizing the need to achieve, wherever possible, global regulatory uniformity."
- Finally, several recommendations issued from the Transatlantic Automotive Business Conference in Washington this April. They called for coordination of vehicle safety research; that the United States become a signatory to the "1958 Agreement"; and, the U.S. engage in bilateral discussions regarding the harmonization of existing and future regulations. In the recommendation concerning harmonization of existing regulations, proposals advancing the concept of functional equivalence were made.

Given the above, the agency has given some thought as to what we should do and I would like to describe briefly our current thoughts and plans.

Before I get into specifics, there is one observation that I want to make concerning the Transatlantic Automotive Business Conference. The statements made at the plenary session of the Conference by Mr. Eizenstat, Undersecretary of Commerce for International Trade, Commissioner Bangemann of the European Union, Mr. Alan Donnelly, Member of the EU Parliament, Mr. Helmut Petri of Mercedes-Benz, Mr. Ron Boltz of Chrysler Corporation, Dr. Mary Good, Acting Secretary of Commerce and Mr. James Kolstad of AAA were unanimous on the position that harmonization of safety regulations not lead to the lowest common denominator regulation nor lead to degradation of safety. That is the first time that I have heard such widespread endorsement of the principle that guides our work in the area of international harmonization. As Administrator and a physician, let me assure all principals involved that I will not accept any degradation of safety. We are protecting people, not dummies.

There is another principle that we support and insist on and that is, transparency in our work on regulations. Transparency was also one of the recommendations of the Transatlantic Business Dialogue. In affirmation of that principle, we are organizing a public meeting this summer to invite other stakeholders to express their views on our harmonization efforts.

With regard to the harmonization of existing regulations, there are both regional and global efforts underway. Within North America, the Automotive Standards Council of the North American Free Trade Agreement (NAFTA) has identified some thirty or forty incompatibilities among the respective regulations of Canada, Mexico and the United States. Agreement on the establishment of Working Groups to address these issues and make recommendations to the Council has been reached. A federal register notice is being prepared for publication in the near future. Within the Asia Pacific Economic Cooperation, a Road Harmonization Project is underway under the leadership of Australia. In Europe, WP29 continues its work in the development of regulations for voluntary adoption by the twenty-eight signatories to the "1958 Agreement."

To harmonize existing regulations, there is the "functional equivalence" approach for achieving harmonization of certain regulations - more about this approach below - and the long term institutional approach of a second-stage amendment of the "1958 Agreement." The latter would institutionalize a process of developing globally harmonized regulations out of the existing regional regulations.

With regard to the second-stage amendment of the "1958 Agreement," we have been working very hard to draft a more simple approach to the amendment of the first-stage revision which has been effective as of November, 1995, for the present signatories. We expect to have that draft ready for circulation among the small drafting group (Australia, South Africa, Japan, and the European Union) before the end of this month. If all goes well, we should be in a position to have a meaningful and productive discussion at the 109th session of WP29 at the end of June. This latest draft will address the concerns of the EU on the matters of "cherry picking" or hybrid regulations, convergence towards global harmonized regulation and preservation of the current process for development of European regulations (EU Directives and ECE Regulations); and the concerns of the United States on recognition of its regulatory investment in the development of global regulations - the full consideration of the FMVSS in the drafting of global regulations - and a vote that is commensurate with its market.

One of the proposed approaches to global harmonization of regulations is to determine which existing standards are "functionally equivalent." That is, if two different countries have regulations addressing the same aspect of a problem and accomplishing similar results, compliance with either regulation should be acceptable.

While determining functional equivalence sounds simple in concept, it may not necessarily be easy to do in practice. We must define what we mean when we say that two regulations "accomplish essentially the same purpose" and agree on what methods should be used to determine when that definition is satisfied. If two different regulations addressing the same problem are stated in nearly identical terms, it should be relatively easy to obtain agreement on whether they are functionally equivalent.

Typically, regulatory requirements are not stated in identical terms. Some regulations are based on performance, while others are based on design. Even if the two regulations addressing the same general problem are both based on performance, they may reflect entirely different approaches to solving the underlying safety problem. Finally, the regulations may differ substantially in their test procedures, and may cover different specific aspects of a general safety problem.

Before any regulatory body can reasonably conclude that a regulation of another country is functionally equivalent to one of its own regulations and permit compliance with the foreign regulation as an alternative to its existing regulation, it must assess and consider the safety consequences of granting that permission. Once "functional equivalence" is defined, many scientific techniques, such as crash data analysis, analytic modeling and comparative testing, can be used to help assess whether different requirements are functionally equivalent.

Thus, the United States proposed that one more, short term, research priority be added to the harmonized research agenda, namely, an effort aimed at developing an acceptable model for determining functional equivalency of existing regulatory requirements. We proposed that this be done in concert with the other proposed research priorities. I am happy to report that our proposal has been accepted and we hope that, in the not too distant future, we will have a uniform model to provide the discipline behind all our efforts to develop sound and cost effective global harmonization of regulatory requirements.

This brings us to the harmonization of future regulations and the establishment of a harmonized research agenda. You will recall that at the beginning of this talk, I mentioned the discussions concerning the establishment of a harmonized research agenda that were held in Geneva on the occasion of the 107th session of WP29. The sequence of events that has brought us to an agreed harmonized research agenda included an initial - February 1995 - letter proposing the possibility of using this 15th ESV Conference to reach agreement on a global harmonized research agenda; further discussion with the focal points present in Geneva in November, 1995; and followup letter of December 26, 1995; aggregation of replies and proposed research topics distributed to focal points April 8, 1996; scheduled meeting for discussion of proposed priorities May 12, 1996, in Melbourne Australia. We now have agreement on a harmonized research agenda that should serve us all in our development of safety regulations based on the same science and therefore harmonized at the outset.

Agreement on a harmonized research agenda should enable us to develop our future regulations in a harmonized fashion, reduce duplicative research and thus obtain more information for the same expenditure, address the most pressing safety problems on a world wide basis, and, of course, minimize the differences in regulatory requirements thus providing economies of scale in the manufacturing arena, and reducing costs for the consumer.

Our selection of Biomechanics as a priority responds to the need for injury measurement surrogates for the head, neck, face, thorax, and lower limbs and the development of test procedures for all crash modes. The fact that these parts of the human anatomy are not very different from continent to continent is a powerful argument for cooperative effort in the development of such surrogates. In other words, we should be able to agree on the surrogates to be used in regulation aimed at providing occupant protection and injury reduction.

The selection of Advanced Offset Frontal Crash Protection as a priority responds to a world wide demand for increased frontal protection in crashes. Europe has been working for some time to develop and establish a frontal crash protection regulation and has chosen the route of an offset crash test as the means of achieving improved frontal protection. The United Sates has been cooperating in that development because it is concerned about the high number of fatalities that occur in frontal crashes that are not being mitigated by the existing frontal protection regulation. Thus, the development of harmonized test procedures based on real world crashes to assess safety performance and compatibility for offset frontal crashes should serve as a common basis for further development of frontal crash protection regulations.

Pedestrian Safety was proposed because the pedestrian fatality and injury levels are very much still a serious safety problem worldwide. Given the increasing demand for mobility, especially in the emerging economies, this problem is likely to increase in magnitude. Thus, the development of a harmonized test procedure based on real world crashes to assess the safety performance of passenger vehicles in their interaction with pedestrians should form the basis for a harmonized approach to regulations applicable worldwide.

Finally, various versions of Intelligent Transportation Systems are being developed around the world. The opportunity afforded by this conference to reach agreement on a research priority that would be aimed at developing test procedures to assess Driver Vehicle interaction of such crash avoidance and driver enhancement in-vehicle systems had to be seized. Although the systems may be different in different parts of the world, the measurement of their crash avoidance and driver enhancement performance should not be.

Those were the proposals we aggregated from the initial round of submissions. Since then, other proposals were made and discussed at yesterday's meeting of the focal points. I have the distinct pleasure to announce agreement by 11 countries and the EC/EEVC on the following set of priorities and the lead organization for each:

- Advanced Offset Frontal Crash Protection and Vehicle Compatibility with the lead being the EC/EEVC.
- Pedestrian Safety - Japan.
- Intelligent Transportation Systems - Canada.
- Biomechanics - the United States of America.
- The technical and scientific aspects of developing a model for determining functional equivalence of existing regulatory requirements - the United States of America in cooperation with Australia.

This is a signal accomplishment and we should all be proud to have set forth this harmonized research agenda in record time.

In addition, we reached agreement on process, next steps, and a report to be submitted to WP29 at its next session in Geneva in June.

Australia

Dr. Allan Hawke
Secretary
Department of Transport and Regional Development

As clean up batter in this afternoon's debate, it is pleasing to see position we have reached. From previous sessions and what I'm about to say, it is quite clear there is a great deal of common ground and indeed agreement on "The Way Ahead".

Australia, as a trading nation, has a strong view on international harmonisation. Vehicles and components are an important part of our trade - amounting to some $1.75B per year.

Australia today builds world competitive cars which are sold in many countries - as are vehicle components which are manufactured here. A wide range of models sourced from Japan, the United States and Europe - and now Korea and Malaysia - are also sold in Australia.

Our vehicle market is a classic demonstration of why we consider it so important to pursue harmonisation of standards. Australia exported automotive products in 1995 to Europe, Asia and America, with cars and commercial vehicles accounting for the major part of that.

The costs of designing and testing to cover several sets of standards is ultimately borne by the consumer - yet the consumer is basically the same in all markets.

There can be no valid basis for differing standards for safety - a crash is a crash no matter where it happens. A smashed leg is a smashed leg in Australia, America, Asia or Europe.

I understand that different standards can add 5-10% to the cost of entering a new market - to what benefit we might well ask. In today's highly competitive international vehicle industry, manufacturers must necessarily focus on cost control. Harmonised standards will lead to lower costs. It is somewhat surprising therefore that industry until recently has not been very vocal in demanding harmonisation. It is encouraging to the thrust of this session of the conference to see that position changing.

The situation we have inherited may be partly a result of traditional use of standards as an industry protection device not an uncommon situation in the past, but rapidly becoming untenable in today's truly global industry.

Survival now depends on building the product customers want, at the right price, and to international standards of quality.

Australia already has over 60% of vehicle standards harmonised with the United Nations Economic Ccommission for Europe (UN/ECE).

We are committed to a review of new vehicle standards to achieve even greater harmonisation. This review is being undertaken in conjunction with New Zealand under the Trans Tasman Mutual Recognition Arrangement.

Australia is also a member of the working group looking at amending the 1958 agreement - to make it a more international agreement that provides the right working environment for the international trade in vehicles.

We are leading work in APEC on standards harmonisation, providing the basis for bilateral and multilateral arrangements to facilitate trade in vehicles. This work is seeking to have UN/ECE recognised as the proper international regulations, and has strong support in the transport working group.

However, those involved in UN/ECE must also do their part. The UN/ECE must ensure that it is truly international in approach that it is responsive - and that it is seen to be working at the 'cutting edge'. Australia will be doing its part to achieve this outcome.

What Else Can Be Done To Progress The Issue?

The time scale of harmonisation efforts of the past is not particularly encouraging - 18 years for passenger car braking!

The US initiative on coordinating the research agenda has to be commended. It must be done to achieve harmonised new standards.

The Question Is - What Can Be Done About Existing Standards?

Australia believes that functional equivalence and mutual recognition offer a circuit breaker. Functional equivalence will allow the acceptance of standards that have similar objectives, but differ in detail.

Side impact is a case in point. Our analysis shows that either the US, or the European standard, will produce substantive benefits for the Australian motorist.

So the draft Australian design rule on dynamic side impact testing allows for acceptance of either the US or the UN/ECE standard.

The natural complement is mutual recognition. Where Australia is to accept an alternative standard, then mutual recognition should be part of the package.

This is where industry needs to play its part. If there are real benefits in functional equivalence and mutual recognition, then industry needs to press a coherent and integrated case.

It has been notable that industry has not been united on this aspect. There are signs that this is changing.

Regrettably, there have also been signs that regulators are often not sensitive to harmonisation trade issues.

It is quite understandable for regulators to seek to 'protect' their creation and maintain the status quo. Understandable, but no longer tenable.

Regulators have to recognise and respond to trade facilitation agendas. If they don't, they could find that the agenda moves outside their control.

If existing mechanisms cannot provide the outcomes needed, then new ones will emerge.

We believe it is important to build on the established arrangements in UN/ECE and work is in hand to do just this. That work must continue and be accelerated. There is an impatience in the air. There is a mood for change.

The challenge for all of us is to harness that mood and make it the agent of change.

From our perspective, there is a need to bridge the Transatlantic Gulf - European and American regulators need to reappraise their position and come closer together.

It is encouraging to see that industry is now endeavouring to get some movement on this issue. Let us all hope that progress under the Trans-Atlantic Automotive Industry Conference will be mirrored in the regulatory sphere.

Industry also needs to work proactively to support change and there will need to be some compromise on both sides.

All those involved need to remember that the motorist - the customer - needs our help - not our pedantry.

Section 4
Technical Sessions

Technical Session 1

Improved Frontal Protection (Offset) and Advanced Occupant Protection Systems
Chairperson: Claes Tingvall, Sweden

CRANIAL-VERTEBRAL FRACTURES AND DISLOCATIONS ASSOCIATED WITH STEERING WHEEL AIRBAG DEPLOYMENT

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Paper Number 96-S1-O-1

ABSTRACT

From field investigations of 477 airbag deployments a number of cervical and craniocervical fractures and dislocations have been identified. Case histories of these injury types will be presented. Injury mechanism appears to be due to distraction of the upper cervical at the junction of the base of the skull from upward forces produced by the airbag in association with possible airbag-chest interaction that augments the distraction forces.

INTRODUCTION

Although the literature is not extensive on injuries related to airbags, there have been a number which may be worthwhile for the reader to puruse. A limited selection of these articles is found in the reference section (1-57). Much has been written on cervical spine fractures and fracture dislocations in the automotive environment, including experimental biomechanical research and field accident data analysis, but few relating the cervical spine injury to airbag deployment.

As of March 1, 1996, 477 steering wheel airbag deployment crashes have been investigated. From this data base we have identified five cases of fractures or fracture-dislocations of the cervical spine related to steering wheel airbag deployments. In addition, from NHTSA's Special Crash Investigation program we have obtained information on five additional cases.

MATERIALS AND METHODS

As part of an on-going research program, all of the police agencies in the immediate and surrounding area of Ann Arbor, Michigan are contacted, on a daily basis, to obtain recent police accident reports. Also contacts through the University Hospital's Emergency Room alert us to an airbag injured patient. In addition, through State Police contact, cases of airbag deployments outside of the local area are identified. Additionally, we have "WANTED" posters distributed thoughout the United States at body repair shops of various car dealerships. The description of injuries from the medical records and/or interviews with the driver are obtained following which the vehicle is inspected in detail, the data recorded on the University of Michigan In-Depth Vehicle Occupant Report (UMIVOR) form for computer input, and exterior and interior photographs are taken.

RESULTS

The following are case capsule descriptions of airbag related injuries to the cervical spine or base of the skull..

Case 1 This crash involved a 1991 Cadillac Deville and a 1987 Ford Econoline van (Fig. 1). The Cadillac went onto the right shoulder and then suddenly veered to the left, crossed into the oncoming lane and struck the van head-on. (Fig. 1) The 79 year-old lap-shoulder belted female driver (161 cm, 45 kg), sustained a complete atlanto-occipital separation (the first cervical vertebra (C-1) from the base of the skull) due to contact with the steering wheel airbag. (UM-2934)

Case 2 In this crash a 1994 Oldsmobile Ninety Eight swerved to avoid some debris in the roadway, crossed the centerline and struck a van head-on (Fig. 2). The 49 year-old unbelted female driver (165 cm, 81 kg), sustained a complete separation of C1-C2 due to contact with the airbag. Airbag contact produced abrasions about the face and anterior neck She also sustained a complete separation of T3 with a spinal cord laceration and lacerations of the heart and aorta, due to impact with the airbag. (UM-3286)

Case 3 In this case a 1993 Plymouth Sundance Duster driven by a 48 year-old unbelted female (165 cm, 91 kg), was struck head-on by a Chrysler LeBaron which had crossed the centerline (Fig. 3). The driver sustained a partial transection of the spinal cord with complete dislocation of the atlanto-occipital joint due to impact with the airbag. Also airbag related multiple bilateral rib fractures and lung contusions were noted at autopsy. (UM-3221)

Case 4 In this crash a 1991 Ford Taurus, driven by an unrestrained 22 year-old female (155 cm, 59 kg), avoided a rear-end collision with a stopped car by turning right, jumping a curb and striking a light post at very low speed (Fig. 4). The airbag deployed producing abrasions beneath the chin and of the anterior neck and face, and an extensive basilar skull fracture with associated injuries to the pons and brainstem. (FMA-077)

Case 5 In this case a 1990 Lincoln Town Car, driven by an unrestrained 73 year-old female (157 cm, 69 kg), went into the median and straight ahead into an overpass sign support (Fig. 5). She and her three passengers were killed, all sustaining multiple fatal injuries. The driver had a complete separation of the cranio-cervical joint with transection of the adjacent brain stem and a separation of the T1-T2. Airbag induced abrasions and contusions were noted about the face and under the chin as well as on the chest wall. (FMA-046)

The following cases were investigated by the NHTSA's Special Crash Investigation program and show similar kinds of injuries. The NHTSA cases were investigated by personnel from Calspan Corporation and Indiana University. Case number 10 was also investigated by the NHTSA's NASS CDS program.

Case 6 This 1990 Ford Taurus struck the rear end of a 1974 Cadillac at a very low speed. Minimal damage to the vehicle is noted in the left front area. The unrestrained 71 year-old (157 cm, 59 kg), driver was dead at the scene, having sustained numerous contusions of her upper anterior chest wall and beneath her chin and a dislocation at the atlanto-occipital joint with a laceration of the brainstem and associated subdural and subarachnoid hemorrhages. Additionally, lacerations of the atrium, aorta, pericardium, liver and mesentery were identified at autopsy, injuries related to the airbag. (CA 93-09)

Case 7 In this crash a 1992 Chevrolet Corsica driven by a 76 year-old unrestrained female (157 cm, 52 kg) , struck a pole producing 12" of bumper crush. The ΔV was calculated at 12-14 mph. Abrasions of the anterior neck and chin and of the upper chest were noted along with a ring type basilar skull fracture and a dislocation of the atlanto-occipital joint. In addition, there were multiple bilateral rib fractures but no noticeable internal thoracic damage. (CA 94-05)

Case 8 A 1990 Cadillac Eldorado struck the rear of a 1990 Oldsmobile. Damage was concentrated in the left front of the Cadillac. The 51 year-old unbelted female driver (163 cm, 67 kg), sustained airbag related injuries including abrasions and contusions to the face, chin, anterior neck, upper chest area, with a fracture/dislocation of C1-C2, bilateral upper rib fractures, a fracture of the sternum and lacerations of the brain stem, aorta and liver. The front right lap-shoulder belted passenger sustained a minor cervical strain. (IN 95-05)

Case 9 A 1990 Lincoln Continental, driven by a 38 year-old unrestrained female (157 cm, 50 kg), impacted the front of a 1993 Chevrolet van. Autopsy revealed an extensive hinge type basilar skull and left parietal-occipital skull fractures, brain stem transection, fractured left ribs and a heart valve laceration (IN 95-06)

Case 10 This 1992 Mercury Grand Marquis, driven by an unrestrained 75 year-old female (163 cm, 41 kg), went off the road and struck a tree in the right front corner. The ΔV was 12 mph. Airbag injuries including comminuted fractures of the distal right radius and ulna and a laceration of the mid right forearm. Ten days post crash she was found to have a fractured cervical vertebra which was surgically repaired--an injury which may be related to the airbag. (CA 93-01)

The NHTSA's Special Crash Investigation program has identified two additional (six total) minor to moderate severity crashes where a driver sustained similar fatal injuries from the deploying driver's side airbag system. All of the drivers in the NHTSA investigations involved unrestrained females.

DISCUSSION

It is well known that the airbag will reduce the frequency of fatality and serious injury to the head, face and torso. However as with every "safety feature" within the vehicle, the feature itself can be source of injuries, and the airbag is no exception. Airbag related injuries to the eyes, face, thorax and upper extremity have been documented in the medical literature. (see references)

In the cases of the high cervical injury and/or separation of the base of the skull from the first cervical vertebrae, or upper cervical vertebrae separation, it is apparent that there is a vertical force upward beneath the chin that is causing this distraction injury. In all cases there were abrasions around the under surface of the chin or anterior throat. Many of these drivers were sitting quite close to the steering wheel module at the time of deployment and a few had an imprint of the module door on their anterior chest wall. (see especially case #6) It appears then that there may be two force vectors acting on these occupants sitting near the airbag. One is a front to rear force on the upper torso with near simultaneous vertical force underneath the chin to cause the distraction injury.

For those basilar skull fracture cases it is most probable that the posterior cervical musculature rigidized the posterior head with the "upper cut" of the airbag under the chin causing a basilar skull fracture. This mechanism of injury has been recently described (49).

The cases presented here are indicative that there is a significant injury potential to airbag deployment but that it occurs relatively infrequently. High cervical vertebral injuries, related to air bag deployment, as described here, have been rarely reported previously (50, 51, 54) In the majority of the airbag deployments most drivers sustain, at most, only abrasions about the face, anterior chin, nose or forehead. We have other cases of short stature drivers sitting close to the steering wheel who did not sustain neck or skull injuries from the deployment of the airbag. The absence of these injuries would indicate that occupant stature or near seating position, in and of itself, does not directly correlate to resultant airbag related cervical spine/basilar skull injuries. As can be seen in Table 1, seven of these 10 drivers were unrestrained. All were females and were 165 cm or less in stature. Clearly, additional crash investigation and bio-mechanical research is needed to fully understand how these types of injuries can be minimized or eliminated.

There are facial or under chin abrasion marks indicating an upward or rearward thrust of the head on the neck. All, but one of the drivers had a cervical spine injury (Case No. 4) of a distraction type at the base of the skull and C-1 or C-1/C-2. This would indicate that there has to be more than an under chin loading, possibly because of the torso involved with the airbag, a rearward force component on the chest and therefore two force vectors are acting to cause this distraction.

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Table 1
Some Characteristics of the Driver Fatalities

Case #	Year/Make/Model	Age/Sex Height/Weight	Belt Use	Cervical Injury
1	'91 Cadillac DeVille	79 F, 161 cm, 45 kg	yes	atlanto-occipital dislocation
2	'94 Oldsmobile 98	49 F, 165 cm, 81 kg	no	C1-C2 dislocation
3	'93 Plymouth Sundance Duster	48 F, 165 cm, 91 kg	no	atlanto-occipital dislocation
4	'91 Ford Taurus	22 F, 155 cm, 59 kg	no	basilar skull fracture
5	'90 Lincoln Town Car	73 F, 157 cm, 69 kg	no	atlanto-occipital dislocation
6	'90 Ford Taurus	71 F, 157 cm, 59 kg	no	atlanto-occipital dislocation
7	'92 Chevrolet Corsica	76 F, 157 cm, 52 kg	no	atlanto-occipital dislocation & basilar skull fracture
8	'90 Cadillac Eldorado	51 F, 163 cm, 67 kg	no	C1-C2 dislocation
9	'90 Lincoln Continental	38 F, 157 cm, 50 kg	no	basilar skull fracture
10	'92 Mercury Grand Marquis	75 F, 163 cm, 41 kg	no	fracture cervical spine


Figure 1

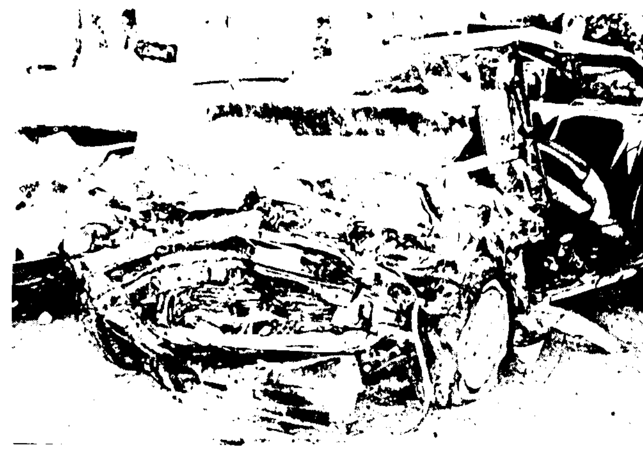
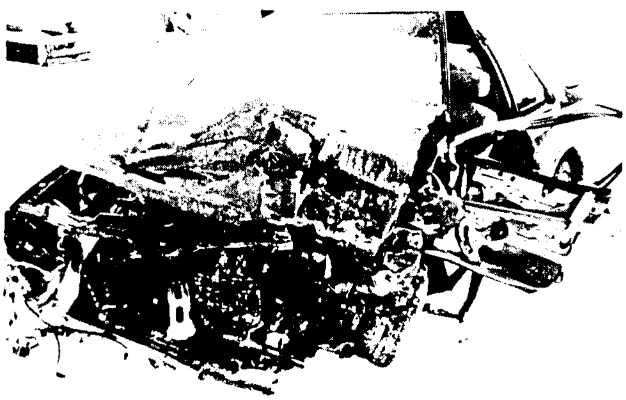


Figure 2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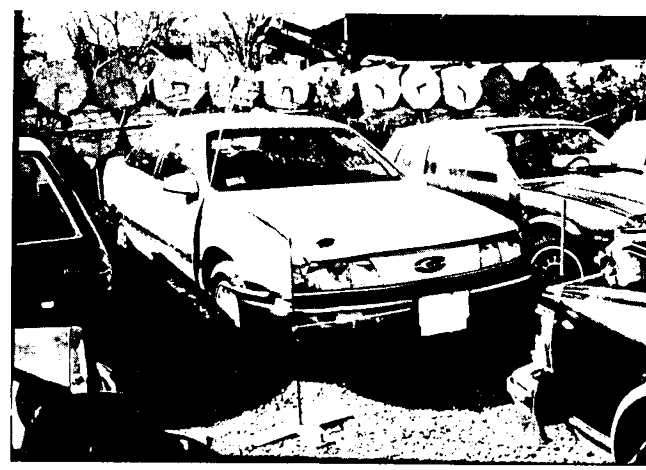
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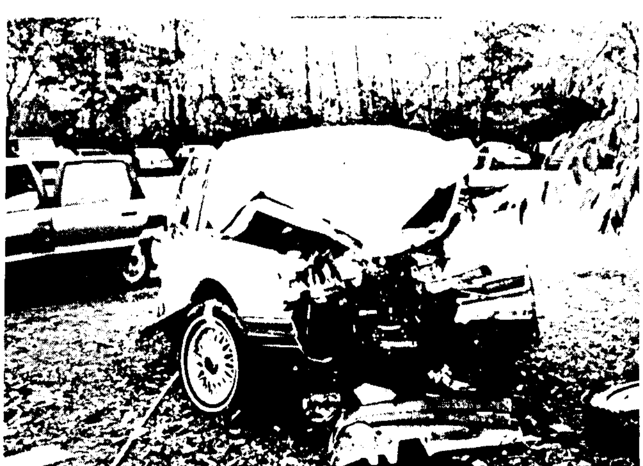
Figure 5

FRONT SEAT PASSENGERS AND AIRBAG DEPLOYMENTS

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ABSTRACT

Passenger side airbag effectiveness is relatively unknown, for infrequently is there a passenger in the front seat when the airbag deploys. Injuries from passenger side airbag deployments are usually minor and approximately 42% of the passengers have no airbag related injuries. The cases selected for presentation are fatalities related to the airbag and are from the field accident investigations conducted of airbag deployments by the National Highway Traffic Safety Administration's Special Crash Investigation program and the University of Michigan Transportation Research Institute personnel.

INTRODUCTION

Because not all cars are equipped with passenger side airbags and the relative infrequency of a front passenger in the car at the time of the crash, little data has yet been presented on the injuries, or lack thereof, of front passengers and airbag deployments. This paper includes some information on 69 front passengers in frontal crashes where the airbag had deployed.

MATERIALS AND METHODS

As part of the on-going investigation program at UMTRI, airbag crash investigations provide the opportunity to study the effectiveness of, and injuries caused by front seat passenger airbags.

At the University of Michigan Transportation Research Institute (UMTRI) we have been investigating crashes since 1961 and have been involved in airbag crash investigations as part of this on-going research program. As of March 1, 1996 477 steering wheel airbag deployment crashes have been investigated by UMTRI personnel.. There were 69 passenger airbag deployments in frontal crashes <u>with</u> a front passenger in place .

RESULTS

Table 1 is a list of the 69 cases of passenger side airbag deployments in the UMTRI series. In this group of 69 cases no one make or model automobile stands out as being more often involved. A high percentage (80%) of the passengers were lap-shoulder belted. Of these, 30 (42%) did not sustain an injury from the airbag. The majority (50%) has minor air bag related injuries of abrasions, contusions and small lacerations about the forehead, face or forearm/wrist area. In Table 1 the overall injury severity (MAIS) of the passengers is shown in the far right column along with the airbag injury severity level (AIS). Note that both are the same in most cases. Not infrequently the MAIS is at a higher level than is the airbag AIS. Of interest are the front passengers who were fatally injured in front end impacts where the airbag deployed. Only two (2) of the 69 passengers sustained fatal airbag related injuries. These two UMTRI cases and five additional NHTSA Special Crash Investigation cases are presented below. The NHTSA cases were investigated by personnel from Calspan Corporation, Dynamic Science, and Indiana University. Case two was investigated by both NHTSA and UMTRI.

<u>Case 1</u> In this crash a 1995 Ford Taurus went off the road and struck a large tree. The male driver sustained a chest injury of AIS 3. The front right 57 year-old female (155 cm, 82 kg) was wearing the 3-point restraint with the shoulder belt behind her back. Interaction with the airbag produced fractures at C2-C3 and multiple facial abrasions and lacerations, a fracture of the hyoid bone (beneath the chin), contusions and abrasions of the chest with bilateral hemopneumothorax and a fracture of T7 with spinal cord transection (FMA-076)

<u>Case 2</u> This 1993 Dodge Caravan rear-ended a 1988 Chevrolet Beretta that was stopped in the roadway yielding to school bus/pedestrian traffic. The driver and three rear seat occupants sustained minor or no injuries. On impact the unrestrained nine year-old male (137 cm, 41 kg) front right passenger moved forward toward the deploying airbag. Airbag related injuries include a fracture/dislocation of C1 with cord transection, facial abrasion, a basilar skull fracture on the right side, and a fractured sternum with a left lung contusion. (CCA-151)

<u>Case 3</u> This 1993 Lexus LX 400 struck the rear of a 1991 Lincoln Town Car. The ΔV was 10-13 mph. Both airbags deployed. The driver sustained minor injuries. The unrestrained front right 7 year-old female passenger (163 cm, 35 cm) had a separation of the atlanto-occipital joint with a fracture between C-2 and C-3 and an associated contusion of

the adjacent spinal cord. There was an abrasion under the chin with a hemorrhage about the neck muscles and esophagus along with a fracture of the left mandible. (CA 95-15)

Case 4 This 1995 Plymouth Voyager was attempting to turn left when it was struck in the front right by a 1984 Chevrolet Suburban . The driver and rear passengers sustained but minor injuries. The 9 year-old male front right passenger (140 cm, 30 kg) had the shoulder portion of the 3-point restraint system behind his back and thus was protected only by the lap portion of the belt. On impact he flexed forward and sustained a separation of the atlanto-occipital joint. Abrasions of the face and chest indicated airbag contact. (IN 95-08)

Case 5 In a parking lot a 1994 Chevrolet Camaro Z28 convertible was attempting a left turn when the undercarriage of the vehicle contacted a curbed parking lot island. Both airbags deployed. The driver sustained but minor injuries. The unrestrained 5 year-old male passenger (105 cm, 25 kg) sustained numerous injuries to his head, face, neck and upper torso as a result of the contact with the airbag, windshield and windshield header. There was separation of the C2-C3 vertebra, disruption of the spinal cord, a laceration of the posterior ligaments between C1 and C2, and dislocation of C2. Additional airbag related injuries included hemorrhages of the brain and a laceration of the inferior vena cava. He died several hours later. (CA 95-20)

Case 6 A 1994 Ford Mustang was struck in the left front corner area by the left front of a 1990 Nissan 4 x 4 pickup truck. The ΔV of the Mustang was estimated at 10-12 mph. Both the driver and passenger side airbags deployed. The driver had minor injuries. The right front occupant, a 4 year-old male (112 cm, 25 kg), was unrestrained. Precrash breaking initiated his forward trajectory and positioned him against or close to the passenger side airbag module when the airbag deployed. Airbag related multiple soft tissue injuries of the face, neck, chest and abdomen were noted along with a subluxation of C2-C3. (CA 94-43)

Case 7 A 1994 Dodge Caravan was struck by a 1990 Chevrolet Malibu that had attempted a left turn directly in front of the Caravan. The driver of the Caravan was braking and steering to the right in an attempt to avoid the collision. Both the driver and passenger airbags deployed. The driver sustained minor injuries. The unrestrained 4 year-old right front female passenger (104 cm, 16 kg) sustained airbag related injuries including an abrasion across the neck, superficial abrasions over the forehead, nose, cheeks and chin and a basilar skull fracture. The ΔV was calculated at 8 mph. (DS 94-20)

DISCUSSION

The cases presented are by no means meant to discredit the passenger side airbag. Rather, this report presents these infrequent "outlier" cases. These fatalities are not related to any specific make or model car. Of the seven fatal cases presented all but one were children and five of the seven were unrestrained. Only two were belted but the shoulder portion of the 3-point restraint was placed behind the back. In the UMTRI files are many other passenger side airbag deployment crashes where the passenger, including children, were not severely injured by the airbag (see Table 1).

In the UMTRI series approximately 42% of the front passengers exposed to the airbag did not sustain any injury from the airbag. About 50% had but minor (AIS-1) injuries to the facial or forearm areas from interacting with the airbag. Rarely are serious injuries noted.

The case capsule descriptions of those passengers with fatal injuries from the airbag appear to have sustained the cervical injuries in a manner similar to the drivers with the same injuries. These distraction cervical spine injuries are believed to be due to upward force vector applied to the head and a rearward force vector to the torso (1)

The NHTSA's Special Crash Investigation program has identified five additional (11 total) minor to moderate severity crashes where a child sustained similar fatal injuries from the deploying passenger side airbag system. All of the children were either unrestrained or improperly restrained. All of the crashes involved pre-impact braking which allowed the unsecured children to move forward and be close to the airbag when it deployed.

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Table 1

Front Right Passengers and Airbag Deployments

Case No.	Year/Make/Model	F.R. Belt Use	Age/Sex Height/Weight	Injury from Airbag	MAIS; AIS-Air Bag
FMA-020	'89 Linc. Cont.	yes	28 M, 183 cm, 95 kg	None	1;0
FMA-028	'89 Linc. Cont.	yes	67 F, 173 cm, 72 kg	Contusion right wrist	1;1
FMA-036	'90 Linc. Cont.	yes	3 F, 99 cm, 15 kg	Loss of consciousness	2;2
FMA-043	'90 Linc. Cont.	yes	73 F, 165 cm, 62 kg	Fracture nose, facial contusions/abrasions	2;1
FMA-044	'90 Linc. Town Car	yes	74 F, 168 cm, 62 kg	None	1;0
FMA-045	'90 Linc. Town Car	yes	42 F, 157 cm, 59 kg	Contusion anterior right upper arm	1;1
FMA-046	'90 Linc. Town Car	yes	72 F, 157 cm, 52 kg	Abrasions, face-fracture mandible	5;1
FMA-056	'93 Ford Taurus	yes	44 F, 165 cm, 57 kg	None	1;0
FMA-057	'92 Linc. Cont.	no	51 M, 175 cm, 84 kg	Laceration and strain 3rd left finger, contusion right anterior forearm	1;1
FMA-062	'92 Linc. Town Car	yes	64 F, 152 cm, 52 kg	Abrasion face & nose	1;1
FMA-064	'94 Ford Probe	yes	28 M, 185 cm, 80 kg	None	2;0
FMA-065	'94 Ford Mustang	yes	36 F, 168 cm, 61 kg	None	4;0
FMA-066	'93 Mercury Sable	yes	65 F, 175 cm, 54 kg	Contusion & abrasion left anterior forearm	2;1
FMA-068	'94 Ford Probe	yes	55 M, 183 cm, 100 kg	Abrasion right face, laceration below left eye, laceration right eyebrow	1;1
FMA-069	'92 Ford Taurus	yes	46 F, 170 cm, 68 kg	Abrasion forehead, nose, chin	1;1
FMA-070	'93 Mercury Sable SW	yes	15 F, 120 cm, 54 kg	Bloody nose, contusion nose, abrasion upper lip	1;1
FMA-071	'92 Ford Taurus	Shoulder belt behind	4 1/2 F, 109 cm, 18 kg	Abrasions right forehead & right side of nose	1;1
FMA-073	'94 Ford Mustang	yes	29 M, 185 cm, 89 kg	Abrasion right & left anterior wrist	1;1
FMA-074	'93 Ford Taurus	yes	21 F, 173 cm, 82 kg	None	1;0

Case No.	Year/Make/Model	F.R. Belt Use	Age/Sex Height/Weight	Injury from Air bag	MAIS; AIS-Air Bag
FMA-075	'94 Ford Thunderbird	yes	16 M, 173 cm, 68 kg	Bloody nose	1;1
FMA-076	'95 Ford Taurus	Shoulder belt behind	57 F, 155 cm, 82 kg	Fracture left & right humerus (bracing against air bag) fracture C2-C3, abrasions & lacerations-face, fracture T7 with cord transection	5;5
FMA-082	'95 Ford Explorer	yes	29 M, 188 cm, 111 kg	Erythema nose & lips	1;1
FMA-083	'95 Linc. Mark VIII	yes	14 M, 168 cm, 45 kg	None	1;0
FMA-085	'96 Ford Probe	yes	15 F, 166 cm, 53 kg	None	0;0
FMA-086	'94 Ford Aspire	unknown	37 F, 163 cm, 68 kg	Unconscious (8 days), fracture left cheek bone, fracture nose	5;5
FMA-087	'95 Ford Mustang	yes	20 F, unknown height & weight	None	0;0
FMA-088	'95 Ford Taurus	yes	32 F, 163 cm, 68 kg	Abrasions right nose, cheek, upper lip, chin & under chin	1;1
FMA-089	'95 Ford Escort SW	yes	48 F, 157 cm, 54 kg	Abrasions nose, left cheek, under chin, erythema lower lip	1;1
FMA-090	'95 Linc. Town Car	yes	71 F, 180 cm, 82 kg	Contusion left anterior forearm	1;1
FMA-092	'94 Ford Mustang	yes	26 F, 168 cm, 59 kg	Abrasion lip, chin & upper lip	1;1
FMA-093	'96 Ford Taurus	no	4 F, 102 cm, 18 kg	**This is a front center pass.** None	1;0
FMA-094	'95 Ford Windstar	no	20 M, 185 cm, 104 kg	None	0;0
FMA-095	'94 Ford Thunderbird	no	37 M, 180 cm, 70 kg	None	1;0
FMA-096	'95 Mercury Tracer	yes	19 F, 160 cm, 56 kg	None	1;0
FMA-097	'96 Ford Windstar	no	14 F, 175 cm, 59 kg	Abrasions and contusion to chin and right side of face	1;1
CCA-093	'93 Dodge Intrepid	yes	24 M, 178 cm, 82 kg	None	0;0
CCA-106	'90 Dodge Intrepid	yes	50 F, 170 cm, 59 kg	None	0;0
CCA-120	'93 Chrysler Concorde	yes	14 M, 137 cm, 34 kg	None	1;0
CCA-121	'93 Chrysler Concorde	yes	7 M, 109 cm, 20 kg	Erythema facial	1;1
CCA-123	'94 Dodge Caravan	yes	65 F, 163 cm, 57 kg	Contusion right & left hand, fracture right 5th digit	1;1

Case No.	Year/Make/Model	F.R. Belt Use	Age/Sex Height/Weight	Injury from Air bag	MAIS; AIS-Air Bag
CCA-138	'95 Dodge Neon	yes	7 M, 122 cm, 28 kg	Abrasions mouth & chin	1;1
CCA-144	'94 Plymouth Grand Voyager	yes	11 F, 122 cm, 31 kg	Abrasion right eyebrow, comp fracture right distal radius, fracture left distal radius & ulna	2;1
CCA-145	'94 Chrysler Condorde	no	59 F, 175 cm, 104 kg	None	2;0
CCA-151	'95 Dodge Caravan	no	9 M, 137 cm, 41 kg	Sternum fracture , fracture dislocation C1, spinal cord transection, basilar skull fracture , subarachnoid hemorrhage, left lung contusion, multiple face & neck contusion & abrasion	6;6
NAB-036	'93 Infiniti J30	yes	47 F, 163 cm, 49 kg	None	1;0
NAB-058	'95 Nissan Maxima	yes	16 F, 170 cm, 54 kg	Laceration & contusion right eye, eyelid, & cheek	1;1
NAB-059	'94 Nissan Altima	yes	7 M, 152 cm, 23 kg	None	0;0
UM-2962	'92 Linc. Cont.	yes	61 F, 160 cm, 59 kg	None	1;0
UM-3127	'93 Mercury Sable	yes	4 mo F, 64 cm, 6 kg	Abrasion facial	1;0
UM-3140	'93 Infiniti J30	yes	47 F, 163 cm, 49 kg	None	1;0
UM-3254	'94 Pontiac Bonneville	yes	67 M, 170 cm, 79 kg	None	1;0
UM-3294	'92 Mercedes SEL	no	50 M, 170 cm, 80 kg	Contusion forehead	1;1
UM-3318	'94 Pontiac Grand Prix	yes	18 M, 178 cm, 77 kg	None	4;0
UM-3331	'94 Toyota Corolla	yes	26 M, 177 cm, 70 kg	None	0;0
UM-3351	'95 Staturn SL1	yes	21 M, 193 cm, 91 kg	None	1;0
UM-3408	'94 Dodge Intrepid	yes	49 F, 157 cm, 57 kg	None	4;0
UM-3418	'93 Dodge Intrepid	yes	9 M, 130 cm, 39 kg	Contusion right & left forearm & face	2;1
UM-3420	'95 Plymouth Neon	yes	19 M, 183 cm, 68 kg	Abrasion right hand	1;1
UM-3423	'96 Buick Rivera	yes	46 F, 170 cm, 59 kg	Abrasions facial , black eye	1;0
UM-3428	'94 Buick LeSabre	yes	37 F, unknown height, 57 kg	None	1;1
UM-3432	'95 Geo Prizm	yes	29 F, unknown height & weight	Contusion left eye, fracture nose	1;1

Case No.	Year/Make/Model	F.R. Belt Use	Age/Sex Height/Weight	Injury from Air bag	MAIS; AIS-Air Bag
UM-3268	'94 Linc. Mark VIII	yes	15 F, unknown height & weight	Abrasion chin	1;1
UM-3074	'93 Infiniti J30	yes	16 M, 170 cm, 54 kg	None	0;0
UM-3433	'96 Nissan Maxima	no	16 F, 155 cm, 50 kg	Laceration left eye	1;1
UM-3339	'93 Linc. Cont.	yes	22 F, 168 cm, 57 kg	None	1;0
UM-3259	'95 Plymouth Neon	yes	18 M, 175 cm, 73 kg	Abrasion left cheek	2;1
UM-3112	'93 Chrysler Concorde	yes	58 F, 168 cm, 68 kg	Abrasion nose	2;1
UM-3144	'93 Dodge Intrepid	yes	14 M, 178 cm, 64 kg	None	0;0
UM-3293	'94 Mercury Sable	no	49 F, 165 cm, 59 kg	Abrasion & laceration right eyelid	3;1

A REVIEW OF DRIVER AIRBAG DEPLOYMENTS IN EUROPE AND JAPAN TO DATE

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Abstract

This study examines data from 186 crash-damaged vehicles in Europe and Japan fitted with driver airbags. There were 130 cases of airbag deployment in the vehicles 97 of which were in single-impact frontal crashes. The majority of drivers in these impacts sustained AIS 1 injuries with the head/face being the most commonly injured body region. Some AIS 2+ injuries occurred to the head/face but these almost always occurred when the optimum occupant protection circumstances were compromised in some way. The most common site of AIS 2+ injuries was the lower limb followed by the upper limb. 12 out of 14 AIS 2+ injuries to the upper limb occurred in vehicles whose airbag deployed. Slight differences were observed in injury outcomes when a differentiation was made between airbag sizes although this may have been a function of sample size. Overall encouraging results are apparent. However, this is a preliminary survey. Follow-up studies with more data are clearly the next stage if a comprehensive overview of the real-world performance of airbags is to be attained.

Introduction

Steering-wheel airbags for frontal crashes have now been widely available for some time in European motor vehicles. However, although much development and testing has been performed in laboratory studies under controlled crash conditions, a true assessment of injury-reduction capability of airbags can only be made from studies of real-world crashes in which the airbag has deployed.

United States studies of the field performance of airbags have now been available for some time as airbags were introduced much earlier in US vehicles. Backaitis and Roberts (1987) conducted one such early study investigating 112 crashes involving government-sponsored fleet vehicles in which the airbag had deployed. It was found that the airbags deployed without failure in all 112 vehicles and of the drivers in the study, 103 (92%) sustained either no or minor injury (at the MAIS 1 level) while of the remaining drivers, 6 received MAIS 2 injuries and 3 MAIS 3 injuries. No injuries were found to be attributable to contact with the airbag, all moderate and serious injuries being generally attributable to contact with the intruding vehicle structure.

Huelke and Moore (1994) conducted an anecdotal study of airbag deployments in frontal collisions and generally found that airbags were performing well even in very severe frontal crashes. Most drivers in the study sustained minor injuries and unsurprisingly, unbelted drivers sustained more minor injuries than belted drivers and also sustained more AIS 2 level injuries.

Further anecdotal evidence was presented in a study by Zuppichini et al (1994) in which three fatalities had occurred in airbag-equipped vehicles in frontal crashes despite the absence of intrusion. All three drivers were unbelted and it was suggested that fatal lesions sustained by the drivers were produced by contact with the airbags themselves.

Huelke et al (1994) examined upper extremity injuries in 50 cases of airbag-deployed vehicles. They found that contusions, abrasions and sprains were commonly reported whilst instances of hand and digit fractures occurred somewhat less frequently. Isolated fractures to the forearm were also reported, all injuries being attributable in some way to deployment of the airbag.

Crandall et al (1994) using data from the US National Accident Sampling System examined head and facial injuries sustained by drivers in one of three conditions these being (i) an airbag only; (ii) a seat-belt only and (iii) a seat belt with an airbag. They found by associating this data with data from laboratory studies that drivers involved in the airbag-only condition incurred the risk of a head contact on the windshield and with it an increased risk of brain and facial injury when compared to seat-belt only restraint emphasising the necessity for seat-belts to be used in conjunction with airbags.

The Insurance Institute for Highway Safety Status Report (1995) detailed 829 US vehicles in which the airbag had deployed and found that about 43% of deployments resulted in at least one airbag-related injury. 96% of such injuries were minor (AIS 1), 3% moderate (AIS 2) while fewer than 1% were serious (AIS 3 or greater). Serious airbag-induced injuries included heart lacerations, lung contusions and fractures to the ribs. The study also included an analysis of fatal injuries attributable directly to the airbag and there were four such cases. In each case the drivers were unbelted and sustained injuries to the head, the chest or both.

Dalmotas, Hurley and German (1995) examined data from a study of injured car occupants in Canada. In all, 242 occupants were involved in accidents in which the airbag deployed 90% of whom wore safety belts. Most of the injuries sustained by the occupants (94%) were minor (AIS 1) while 5% were rated as AIS 2. These injuries typically included brief losses of consciousness and fractures to the upper or lower extremity which were injuries not necessarily attributable to the airbag. The authors concluded that intervention of an airbag in moderate and severe crashes greatly reduced the likelihood of severe to fatal head injury in an unbelted occupant whilst there was a perceived increased risk of sustaining upper extremity and facial injuries in collisions whose severities were marginally above the deployment threshold.

Libertiny (1995) using data from the National Accident Sampling System (NASS) observed that while airbags were doing what they were designed to do (i.e. decreasing the severity of injuries in major accidents) there remained the possibility of minor airbag-induced injuries increasing in frequency. However, he concluded that a decrease in overall severity was an acceptable design trade-off.

This study is designed to be a preliminary study of injuries in crashes in which the airbag has deployed in Europe and Japan. Only limited data are available in view of the fact that airbags are a relatively new concept in the European and Japanese vehicle markets and there have been only a limited numbers of crashes involving airbag-fitted and deployed vehicles. While Otte (1995) presented a study of 42 accidents with airbag-deployments in Germany, future studies are required with more data from several countries.

Methodology

This study used data from individual retrospective studies of crash injuries that are on-going in the UK (the Co-operative Crash Injury Study), France (at PSA Peugeot-Citroen/Renault), Germany (the Medical University of Hannover) and Japan (based on the annual reports on motor vehicle accident investigation and analysis by the Ministry of Transportation, Japan). Vehicles in each study were generally inspected a few days after the collision in garages and recovery-yards, but in Germany, the Medical University of Hannover team attended the scene of accidents and documented the injuries themselves. In other cases, medical information was gathered from hospitals. Overall the study used data from crashed vehicles which contained a driver airbag which did not necessarily deploy. Only drivers were considered in the study. In all some 186 vehicles were included in the study.

All injuries where known are rated according to the Maximum Abbreviated Injury Scale (MAIS) 1985 or 1990 revision.

The combined group of crash injury cases cannot be said to be representative of any special geographical area as the relation between each sample and its national population has not been identified. The Hannover sample is based on a statistically random procedure and is representative of the German northern region. However none of the samples preferentially select 'good' or 'poor' examples of airbag performance. The sample can be used as in this analysis to identify features of airbag crashes.

Results

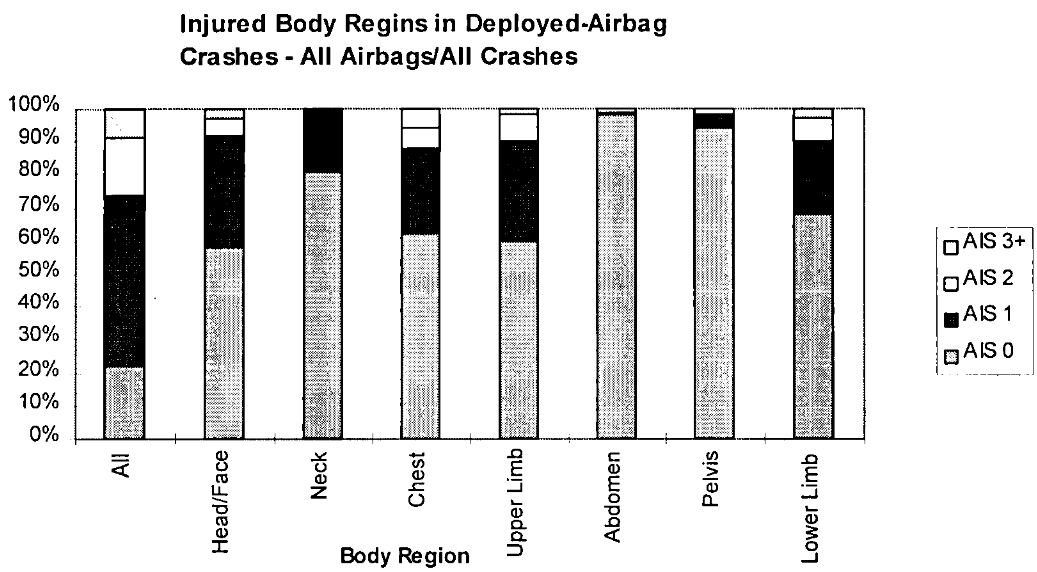
1. All Impact Types

Of 186 vehicles fitted with airbags, there were 130 deployments and 56 non-deployments. The collision directions for deployments for 152 single impact crashes in which the airbag deployed were as shown in table 1. The collision circumstances for 34 multiple impact crashes are shown in a later section.

Table 1: Collision Directions for Airbag Deployments

Single Impact Crashes (N = 152)		
Collision Direction	Airbag Deployed (N=110)	Airbag Not Deployed (N = 42)
Frontal	97	22
Right-side	8	4
Left-side	2	3
Rear	0	5
Rollover/Not Classified	3	8

Figure 1



The collision severities for vehicles in single-impact frontal crashes in which the airbag did not deploy were examined. For cases in which the Delta-V was calculated, the range was between 10 and 25 km/hr. Therefore it must be emphasised that there were no observed cases of non-deployment in frontal collisions when the collision threshold was dramatically exceeded.

Of the drivers in frontal crashes without airbag deployment, 10 sustained no injury, while a further 10 sustained MAIS 1 injuries. 1 occupant sustained a MAIS 2 injury to the thorax this occupant being a 49 year-old restrained female in a vehicle whose collision Delta-V was calculated to be 17km/hr. One occupant sustained a MAIS 3 injury, also to the thorax, this occupant being a 54 year-old restrained male in a vehicle whose collision ETS only was available, this being 32 km/hr.. Both these drivers were restrained by pretensioner belt systems which both deployed.

2. All Collision Directions & Airbag Deployments

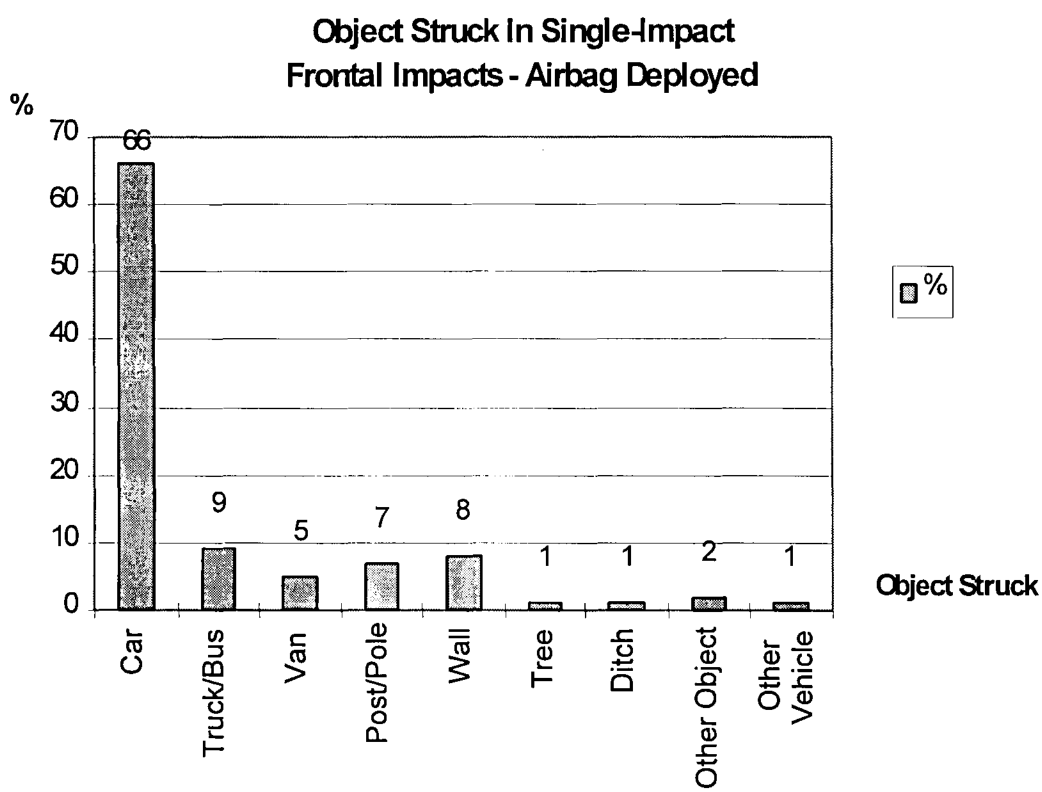
Figure 1 shows the distribution of injured driver body regions in all cases of airbag deployments in all impact directions. As can be seen from this figure, all body regions were predominantly uninjured. However, head, chest, upper limb and lower limb body regions received injuries at all levels. 8% of drivers sustained AIS 2 upper limb injuries, 6% sustained AIS 2 chest injuries while 5% sustained AIS 2 head injuries and 7% sustained AIS 2 lower limb injuries. In total, 10 drivers sustained AIS 2 upper limb injuries in airbag-deployed vehicles. 9 of these drivers were male and 8 drivers were restrained. The

ages of the drivers ranged between 25 and 71 years old (mean = 42 years).
Airbags deployed in 13 (9%) of the single impact crashes where the impact direction was not frontal.

3. Single Impact Frontal Collisions

This section focuses on the outcome of airbag deployments in single impact frontal crashes. Such collisions are worthy of consideration since theoretically this is the optimum collision condition for occupant protection. In all, 97 vehicles met this collision condition. Figure 2 shows the distribution of objects struck for vehicles. Only those vehicles in which the object struck was established are shown.

Figure 2



The majority (66%) of vehicles in single-impact frontal crashes in which the airbag deployed were involved in car-to-car accidents while a further 15% of vehicles were in collisions with other road vehicles.

4. Single Impact Frontal Collisions and Injury Severities - Deployed Airbags

Of the drivers in the 97 vehicles, there were 92 whose Abbreviated Injury Scores (AIS) were available. 5 drivers sustained injuries whose injury severities were not calculated. 67 drivers wore seat belts and 22 did not wear seat belts. Seat-belt usage could not be established for 8 drivers. With regard to airbag size, 61 drivers were involved in collisions with deployed airbags below 40 litres and 28 drivers with deployed airbags above 40 litres. Airbag size could not be established for 8 drivers.
The Delta-V distributions for drivers according to injury severity are shown in figure 3. As would be predicted, drivers sustaining no or minor injuries were involved in collisions of lesser severity. The median Delta-V for AIS 2+ injuries was 46km/hr with the lowest Delta-V being 29km/hr. The median Delta-V for AIS 0 and 1 was 36 km/hr with the lowest value being 15 km/hr

Figure 3

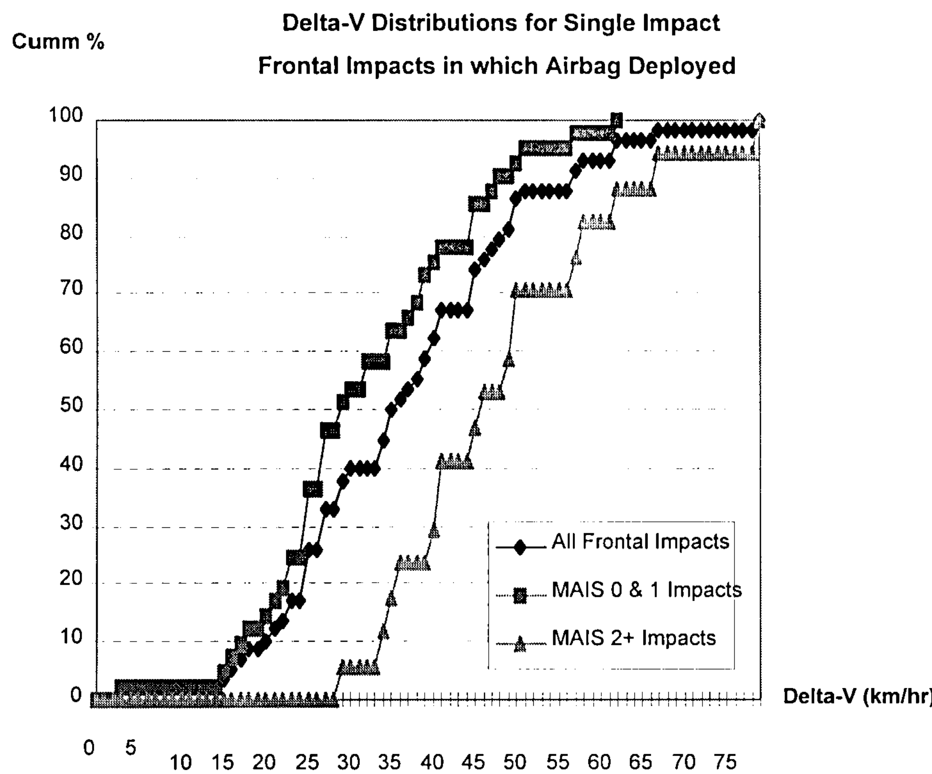


Figure 4

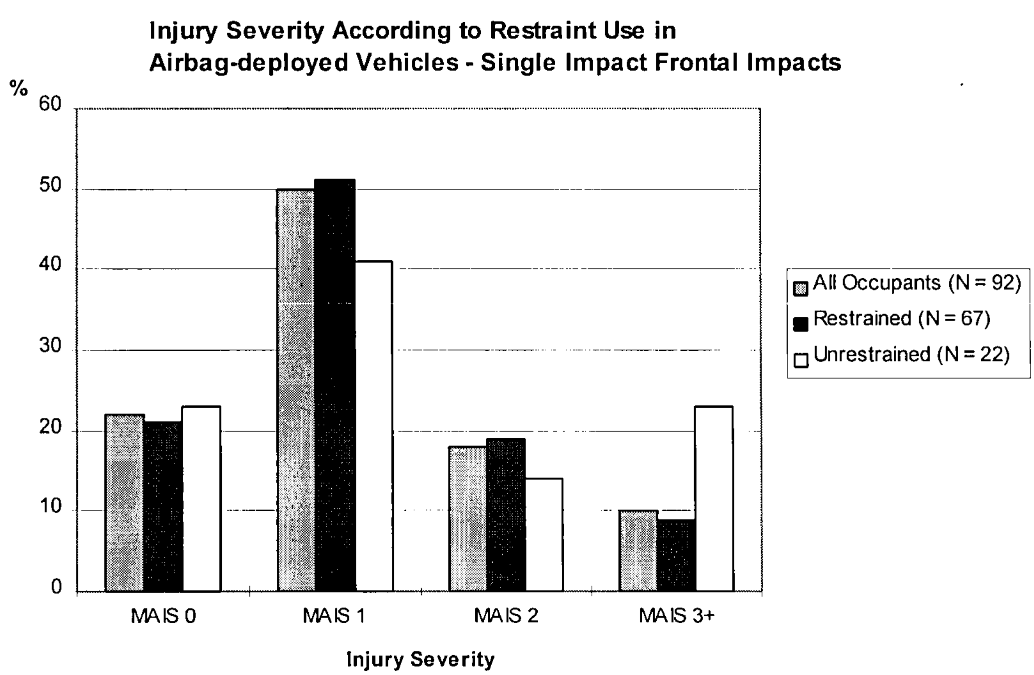
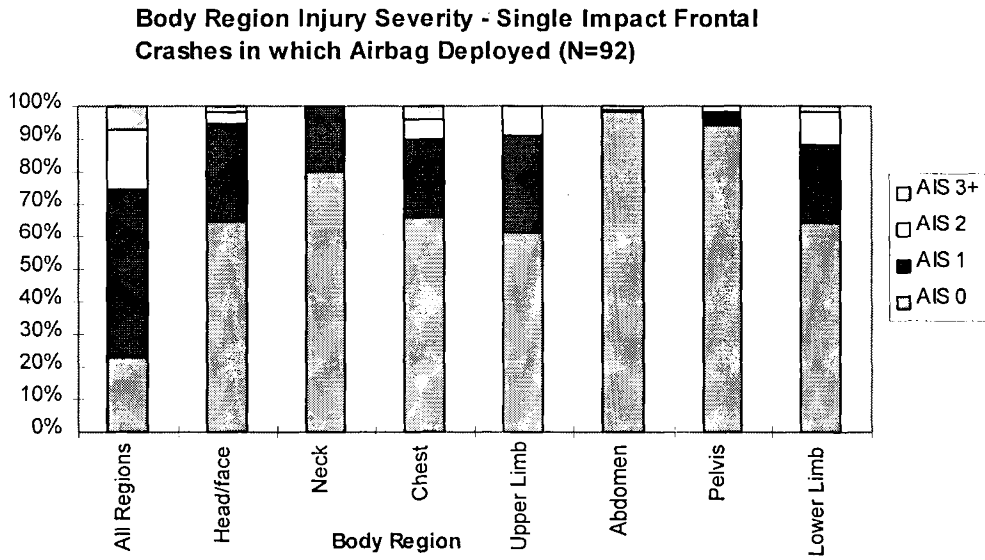


Figure 4 shows the distributions of drivers according to their restraint condition and the injury they sustained. The majority of drivers sustained either AIS 0 or AIS 1 injuries. However 19/67 (28%) restrained drivers and 8/22 (36%) of unrestrained drivers sustained injuries rated as AIS 2 or greater. Actual body regions injured for both restrained and unrestrained drivers are shown in figure 5.

Figure 5



As can be seen from figure 5, 48/92 (52%) of the drivers sustained AIS 1 injuries. The head/face was the most commonly injured body region while the lower limb was the most common site of AIS 2+ injuries. 28/92 (30%) of drivers sustained AIS 1 head/face injuries while 5/92 (5%) sustained AIS 2+ head/face injuries. 8/92 (9%) of drivers sustained AIS 2+ upper limb injuries and 19/92 (20%) of drivers sustained AIS 1 neck injuries which were predominantly soft tissue injuries/neck sprains, so-called 'whiplash' injuries.

5. Airbag Size and Injury Outcomes

The size of the airbag was known for 118 out of 130 deployed cases (regardless of impact type). A distinction was made between drivers in deployed cases where the airbag was below 40 litres capacity and drivers in deployed cases where the airbag was above 40 litres capacity. Injury outcomes were examined and are shown in tables 2 and 3. Table 2 shows the injuries sustained by the 75 drivers in <40 litre airbag vehicles. As can be seen from this table, the most commonly injured body regions were the thorax and the upper limb with 33/75 (44%) of drivers each sustaining an injury to these regions. 32/75 (43%) of drivers also sustained an injury to the head/face the majority of these injuries being at the AIS 1 level of severity although 5/75 (7%) sustained an AIS 2 or greater injury to this region. 7/75 (9%) of drivers sustained an AIS 2 injury to the upper limb while 9/75 (12%) of drivers sustained an AIS 2 or greater injury to the thorax.

Table 2: Injured Body Regions to Drivers in Deployed Airbag Crashes <40 Litres (N=75)

	AIS 0	AIS 1	AIS 2	AIS 3+	AIS Unknown	Total
Head/Face	41	27	3	2	2	75
Neck	56	17	0	0	2	75
Thorax	40	24	5	4	2	75
Abdomen	72	1	0	0	2	75
Pelvis	68	4	1	0	2	75
Upper Limb	40	25	7	1	2	75
Lower Limb	45	20	5	3	2	75

Table 3: Injured Body Regions to Drivers in Deployed Airbag Crashes >40 Litres (N=43)

	AIS 0	AIS 1	AIS 2	AIS 3+	AIS Unknown	Total
Head/Face	25	11	3	1	3	43
Neck	34	6	0	0	3	43
Thorax	26	8	2	4	3	43
Abdomen	38	1	0	1	3	43
Pelvis	38	1	1	0	3	43
Upper Limb	23	13	3	1	3	43
Lower Limb	28	8	4	0	3	43

Table 3 shows the injuries sustained by the 43 drivers in >40 litre airbag vehicles. The most commonly occurring injury in these vehicles was an injury to the upper limb with 17/43 (40%) of drivers sustaining an injury to this region although the majority of these were at the AIS 1 level. 14/43 (33%) of drivers sustained an injury to the thorax with 6/43 (14%) drivers sustaining an AIS 2 or greater injury to this body region. 15/43 (35%) of drivers sustained an injury to the head/face with 4/43 (9%) sustaining an AIS 2 or greater injury to this region.

Of 64 restrained drivers who were involved in airbag deployed vehicles in frontal crashes, 6 out of 24 drivers sustained an AIS 1 neck injury in <40 litre airbag vehicles and 5 out of 40 sustained an AIS 1 neck injury in >40 litre vehicles. Statistical analysis of this data (using a Fischer test; $p=0.12$) reveals no statistical difference in terms of neck injury risk according to airbag size.

6. Multiple Impacts

In total, there were 20 cases of airbag-deployment and 14 cases of non-deployment in multiple impacts and rollovers. A break-down according to the most severe collision type is shown in table 4.

Table 4: Most Severe Collision Types for Airbag-Equipped Vehicles in Multiple Impacts

Deployment (N=20)		**Non-Deployment (N=14)**	
Most Severe Collision	N	Most Severe Collision	N
Frontal	13	Frontal	2
Rightside	3	Rightside	0
Leftside	1	Leftside	2
Rear	1	Rear	0
Rollover	2	Rollover	8
Not Known	0	Not Known	2

For deployments in multiple impacts, the majority (13/20) occurred when a frontal impact was the most severe collision of the crash sequence. Of the drivers in these frontal impacts whose injury severities were known, 2 sustained MAIS 2 injuries, one to the chest and one to the upper limb respectively. The remaining drivers in these impacts sustained either no or minor (AIS 1) injuries only.

For non-deployments, the airbag was not activated in 2 crashes in which a frontal impact was the most severe collision of the crash sequence. For the first of these, the delta-V was calculated to be 25km/hr and for the second neither delta-V nor EES were known. In both cases, MAIS 1 injuries only were sustained by the drivers. 4 out of 13 drivers in these non-deployment cases sustained AIS 2 head injuries. 3 of these injuries occurred in a rollover and 1 in a left-side impact.

7. Case Reviews

Some interesting and exemplar cases of airbag deployments were apparent. These are as follows;

Case (1): Airbags and Arm Injuries. This vehicle was involved in a car-to-car collision with an EES of 41km/hr. The combined weight of the vehicle and occupant was 972kg. There was no driver-side intrusion in the vehicle. The impact was adjudged to be 11FDEW3 with 73% overlap. The 35-litre airbag deployed as expected. The restrained driver, a 35 year-old 98kg 170cm male driver sustained a fractured distal radius together with a sternum fracture and laceration to the right knee. It was judged that the arm injury was a result of airbag interaction as no other contact sites were observed in the vehicle.

Case (1) Airbag-deployed; 35 Year-old Restrained Male

Case (2): Under-run and Deployment This vehicle was involved in an under-ride type collision with a truck. The CDC was 10LDAW6 but the airbag (65 litres) did deploy. Neither Delta-V nor EES were calculated. There was extensive intrusion and the driver, a 36 year-old restrained male sustained extensive fatal head injuries attributable to contact with the truck itself.

Case (2) Airbag -deployed; 36 year-old Male (fatally injured)

Case 3: Under-run, Deployment and Arm Injury This vehicle was in a under-run type collision with a light truck with an adjudged CDC of 12FZAA9. The driver, a 32 year-old 73kg 170cm male sustained several cuts, bruises and abrasions but also an undisplaced fracture of the left distal radius. The airbag deployed as there was significant involvement of the side rails and engine. No relevant interior contacts were found but it remains a possibility that the arm injuries were a result of direct contact on the truck.

Case (3); 32 year-old Restrained Male

Case (4): Arm Injury Caused by Airbag at Minimal Collision Severity This vehicle drove into a stationary vehicle with a Delta-V calculated to be 12km/hr. Damage to the front of the vehicle was minimal but the airbag deployed. The female driver of the vehicle sustained a burn injury of the lower arm and a haematoma to the hand through inflation of the airbag.

Case (4) Female Occupant Age Unknown - Upper Limb Injury

Discussion

The past three/four years have seen the rapid introduction of airbags into the majority of new cars in both Europe and Japan. As this has been driven by vehicle manufactures and market forces rather than by legislation, there are a variety of systems and a number of differing sizes of airbags available. It is necessary that any vehicle design modification of such magnitude as this is monitored in order to identify potential dis-benefits as well as benefits. This has been the overall aim of this paper.

The main operation of the airbag is to reduce the level and severity of head and face injuries (and in the case of Federal systems, torso injuries) to the driver through reducing harsh contacts on the steering-wheel. In this data-set, the head/face was the body region most commonly injured, however, most of these injuries were of severity AIS 1 which are very difficult to avoid anyway. Encouragingly, there was, in fact, only one example of an AIS 2 injury occurring to the head/face of a restrained occupant of a vehicle involved in a frontal crash with a collision severity below 50km/hr. Of the remaining AIS 2+ injuries to the head/face, the circumstances of the collision exceeded the optimum crash conditions whereby injury prevention could be expected.

The level of AIS 2+ injuries to the upper limb throughout the study reinforces the concern from American studies that airbags can have a causal effect on this type of injury. Although this study is not sufficiently detailed to allocate causes to the injuries, it is notable that 12 out of 14 AIS 2+ injuries to the upper limb occurred in vehicles in which the airbag deployed; 9 in vehicles whose airbag size was below 40 litres. This data also reinforces the American view that the lower limb is an important site of residual injury as there is no evidence in the study to suggest an airbag effect.

A slight effect of airbag size was observed with more frequent head/face, neck, chest and limb injuries occurring in vehicles with airbags below 40 litres. However, it is not possible to establish whether this is a real effect without more data which will undoubtedly follow. Furthermore, the data do not give any indication of the relative risk of being injured at all since cases of no-injury in airbag-deployed vehicles were less likely to be sampled in the study. There were no instances in this study of fatal injuries being directly attributable to airbag deployment. However, there was some evidence to suggest that unrestrained drivers in frontal impacts are more prone to greater severity injuries.

There were relatively high rates of neck injury in this study which is a finding worthy of further exploration. As the data stand, the neck injury rate could be a function of sample size. However, it was predicted before the general introduction of airbags that their presence could help to reduce the occurrence of this type of injury. The data shown here, while not conclusive suggest that neck injuries will still occur in frontal crashes with deploying airbags. This was a reported effect by Otte (1995) who proposed that it was due to the result of a combination of upper-body fixation from the restraint system coupled with loading to the head during head excursion from the deploying airbag. It will be interesting to examine this issue further when more data have been gathered.

Finally, with regard to deployments themselves, it is apparent that 25% occurred in collisions whose severity was below 25km/hr. As AIS 2+ injuries of any sort are comparatively rare at this collision severity, it is possible that there are few benefits from deployment in such circumstances. Unnecessary deployment of course increases insurance costs. Again more data is required to establish an optimum deployment threshold for real-world accidents.

Overall this has been a preliminary examination of the outcomes from deployed airbags in Europe and Japan. The initial results are encouraging particularly with respect to head injury although some problems, particularly with regard to upper limb injuries are apparent. The next stage is to conduct a more in-depth study and such a study is planned as soon as sufficient data are available.

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EXPERIENCE WITH AIRBAG-EQUIPPED CARS IN REAL-LIFE ACCIDENTS IN GERMANY

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Paper Number 96-S1-O-04

ABSTRACT

This paper describes experience obtained from the analysis of 249 real-life accidents in Germany in which front airbag-equipped cars were involved. The performance of the airbag as a safety system and the injury patterns of the drivers and front seat passengers are analysed.

In comparison with belted drivers and front seat passengers (without an airbag) it was possible to observe a significant reduction in severe and fatal injuries to the belted and airbag-protected drivers; but for the belted and airbag-protected front seat passengers airbags resulted in no major reduction in injuries. Driver airbags led to a substantial reduction in head injuries, and so in frontal car collisions with airbag deployment the injury severity MAIS 2+ in the case of belted drivers was dominated by injuries to the extremities, especially to the feet. The injury severity MAIS 2+ for belted front seat passengers, on the other hand, was dominated by injuries to the thorax. In this accident material, fatal injuries even with airbag deployment were only observed in very severe frontal impacts with a high degree of intrusion of the passenger compartment ("disaster cases").

The system of "airbag plus safety belt" has to be optimised in the future and "intelligent airbag systems", especially with regard to the passenger airbag, are required. "Intelligent airbag systems" must avoid an airbag deployment on the passenger side if there is no passenger present and also have to solve the conflict with rearward facing child seats. These systems should avoid a too aggressive airbag deployment. Premature inflation of the airbag (EES about 10-20 kph) can lead to additional injuries such as burns and abrasions, and from this point of view, but also for reasons of costs, the firing threshold of airbags should be about 25 to 30 kph. The optimisation of the airbag has to focus on the protection criteria of occupants wearing safety belts.

INTRODUCTION

The idea of an "inflatable container" for the purpose of avoiding injuries to car occupants was patented by Walter Linderer at the German Patent Office in Munich on 6th October 1951 [1]. This basic idea was also described in a US patent in 1952 in the form of a general airbag system. The first production car in Germany equipped with a driver's airbag left the assembly line in December 1980. The passive safety system called SRS (supplemental restraint system) was intended, in conjunction with the three-point seat belt, to provide the driver with optimum protection in head-on collisions. For almost one decade only cars in the executive class were equipped with airbags, and it was not until the beginning of the 90s that cars in the medium class and small class were fitted with airbags to any great extent. Today almost 100% of new cars can be bought with a driver-side airbag; in the future the proportion of cars with a passenger-side airbag as well will continue to increase.

This presentation reports on the first experience with front airbag-equipped cars in Germany and the effects of the airbag in real-life accidents.

DESCRIPTION OF THE ACCIDENT MATERIAL

On account of the very small proportion of airbag-equipped cars among all the registered vehicles it was relatively difficult in Germany in the 80s and early 90s to build up reliable accident material relating to airbag cars. Thus in the past only relatively few individual accidents involving airbag-equipped cars were available to the German motor insurers. So in 1994 a new approach was adopted: in a joint operation of the Verband der Schadenversicherer (VdS) (Association of Motor Insurers) and the General German Automobile Club (ADAC) in July 1994 the readers of 'motorwelt' (the official organ of the ADAC; monthly circulation 13 million) were called upon to report on accidents with airbag-equipped cars. This operation made it possible to collect a very well documented and reliable body of material covering 249 accidents and evaluate them in depth by additional analysis of the appropriate German motor insurers' claims records.

This accident material is the basis of this presentation and is referred to below as "airbag material".

Time and Scene of the Accident

The great majority of the accidents (93.4%) occurred in the years 1993 to 1995 (Table 1); only 6.6% of the airbag-equipped cars were involved in accidents in 1992 or earlier.

Table 1.
Time of Accident

Year	Number	%
1995	77	34.1
1994	102	45.1
1993	32	14.2
≤1992	15	6.6
Total	226	100.0
n.a.	23	
Total	249	

n.a. = not ascertained

Table 2.
Scene of Accident
Comparison of Airbag Material and VS 90 [2]

All Accidents - Airbag Material					
	Urban	Suburban	Total	n.a.	Total
Motorway		48	48		48
Federal road	11	40	51		51
Country/State road	9	46	55	2	57
County road	2	23	25		25
Others	27	1	28		28
n.a.	15	4	19	21	40
Total	64 28%	162 72%	226 100%	23	249

Car-to-Car Accidents					
	Urban	Suburban	Total	n.a.	Total
Airbag Material	48 36%	85 64%	133 100%	4	137
VS 90	9,715 68%	4,507 32%	14,222 100%	778	15,000

n.a. = not ascertained

Almost three-quarters of the accidents reported took place outside built-up areas (Table 2). Depending on the kind of roads, the accidents were distributed more or less equally over motorways, federal roads or country and state roads. The accident rate on roads outside built-up areas is very high in the airbag material, as a comparison with the representative VdS accident material „Vehicle Safety 90" (VS 90; 15,000 car-to-car accidents with personal injury from 1990) [2] shows: thus in VS 90 only 32% of the accidents took place outside built-up areas. This is an indication of an altogether higher accident severity in the case of airbag-equipped cars and is also confirmed by the damage observed to these cars.

Degree of Damage

The distribution of the degrees of damage (DD) in the airbag and the VS 90 material (Table 3) shows that airbag-equipped cars were involved in accidents with a far higher accident severity. Thus 58% of the airbag-equipped cars were severely to totally damaged (DD 3-5), while this figure only applies to 38% of the cars in the VS 90 material. Examples of the classification into different degrees of damage are presented in Appendix I.

Table 3.
Degree of Damage
Comparison of Airbag Material and VS 90 [2]

Degree of Damage	Airbag Material		VS 90
	All	Car-to-Car (Front)	Car-to-Car (Front)
DD 1 - light			1,112 14%
DD 2 - moderate	77 39%	43 42%	3,904 49%
DD 3 - severe	86 44%	46 45%	2,182 28%
DD 4 - extreme	30 15%	11 11%	605 8%
DD 5 - total	3 2%	2 2%	129 2%
Total	196 100%	102 100%	7,932 100%
n.a.	53	18	8,700
Total	249	120	16,632

n.a. = not ascertained

On account of the, in some cases considerable, vehicle deformation there were problems in rescuing the occupants in 31 cases (13.5%): in 25 cases it was only a matter of the doors being stuck, but in six other cases the driver and/or the front passenger were trapped in the car and could only be freed by the rescue services.

Accident Opponents

In more than half of the accidents (59%) the airbag-equipped cars collided with a car (Table 4), 14% collided with a minibus or transporter, a truck or bus, and 13% of the cars ran into a fixed obstacle such as a tree or wall. In 15 cases (6%) the collision was with a guardrail - which the car very often drove under - and in ten further cases (4%) the airbag-equipped car did not collide directly with an accident opponent or an object, but left the road and ran into a ditch, an embankment or a field. In isolated cases a motorcycle, a trailer, an agricultural vehicle or even a truck wheel were the accident opponents; in 17 cases it was not possible to ascertain the accident opponent from the available records.

Table 4.
Accident Opponents

Airbag Car vs. Accident Opponent	Number	%
Car	137	59.1
Transporter, Minibus	13	5.6
Truck	17	7.3
Bus	2	0.9
Motorcycle	2	0.9
Deer (roe, wild boar)	3	1.3
Pole (tree, mast)	27	11.6
Wall	3	1.3
Guardrail	15	6.5
Ditch, Embankment, Field	10	4.3
Others (trailer, agricultural vehicle, truck wheel)	3	1.3
Total	232	100.0
n.a.	17	
Total	249	

n.a. = not ascertained

Car Types

In Table 5 all 249 airbag-equipped cars involved in an accident are listed. Cars from all classes were involved in the accidents, not only compact cars (e.g. Fiat Punto, Ford Fiesta, Renault Clio etc.) but also popular saloons (e.g. Audi 80/100, BMW 5 Series, Ford Mondeo, Mercedes E Class, Opel Omega etc.) and cars in the executive class (e.g. BMW 7 and 8 Series, Mercedes S Class, Toyota Lexus etc.). The relative proportion of the small and compact cars of almost one-third in the available

Table 5.
Car Types in Airbag Material

Manufacturer	Car Type	Number
Audi	80/90/Coupe	16
	A4	1
	100	12
	V8	1
BMW	3 Series	26
	5 Series	12
	7 Series	3
	8 Series	1
Chrysler	Dodge Dynasty	1
	Le Baron	1
	Voyager	2
	Jeep Cherokee	1
Fiat	Punto	3
	Tempra	1
Ford	Fiesta	5
	Escort	9
	Scorpio	1
	Mondeo	10
	Probe	2
Ford (US)	Taurus	1
	Mustang	1
	Others	1
General Motors	Buick Park Avenue	1
	Chevrolet Camaro	1
Hyundai	Lantra	1
Jaguar	XJ6	1
Lancia	Thema	1
Mazda	323	1
	626	1
	MX5	1
	Xedos 6	1
Mercedes	190	3
	C Class	4
	E Class	17
	S Class	7
	SL (Roadster)	1
Mitsubishi	Sigma	1
	3000 GT	1
Nissan	100 NX	2
Opel	Corsa	5
	Astra	17
	Vectra	6
	Tigra	1
	Omega	3
Porsche	911	2
Renault	Twingo	1
	Clio	5
	R19	3
	Laguna	1
Saab	9000	2
Seat	Ibiza	2
Toyota	Corolla	1
	Carina	1
	Camry	3
	Lexus	1
Volvo	850	4
VW	Polo	1
	Golf	22
	Passat	12
Total		**249**

accident material shows that the airbag is on the advance even in small, light vehicles. Nevertheless, larger and more powerful cars can be found far more frequently in the airbag material than in all the cars registered in Germany [3]; thus, for example, cars of more than 120 kW accounted for 23% compared with 4%, thus representing a far larger proportion (Figure 1).

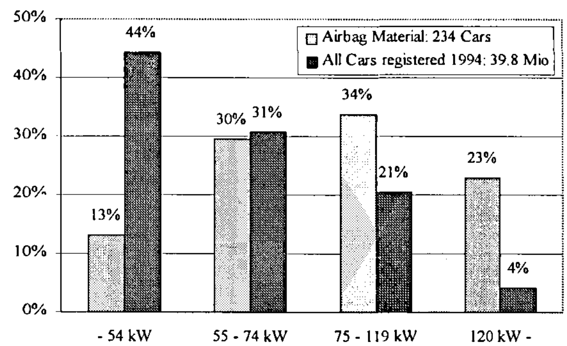


Figure 1. Engine power in kW. Comparison with all cars registered in Germany in 1994 [3].

On account of the "car with airbag" selection criterion a large number of the cars were relatively new: 88% of the cars were manufactured in the period 1992 to 1995 and at the time of the accident were only one or two years old.

Airbags and Belt Pretensioners Installed in the Cars

Of the altogether 249 cars 160 were equipped with a US or full-size airbag for the driver (Table 6), and 51 (32%) of them also with an airbag for the front seat passenger. A Euro-airbag was installed in 89 cars on the driver-side, and 55 (62%) of these also had a passenger-side airbag. Thus a total of 43% of the cars were equipped with an airbag on the passenger-side. It is noticeable that the cars with a Euro-airbag for the driver have a passenger airbag almost twice as frequently as those cars with a US airbag on the driver's side.

Table 6.
Rate of Equipment with Airbag

	Euro-Airbag	US Airbag	Total
Driver	89 100%	160 100%	249 100%
Passenger	55 62%	51 32%	106 43%

The rapid increase in the proportion of cars fitted with twin airbags is very well reflected in the available airbag material (Figure 2): while only 16% of the cars built in 1992 have twin airbags, as many as 45 out of 100 cars built in 1993 have airbags for both driver and passenger, with a upward trend for cars from 1994 and 1995.

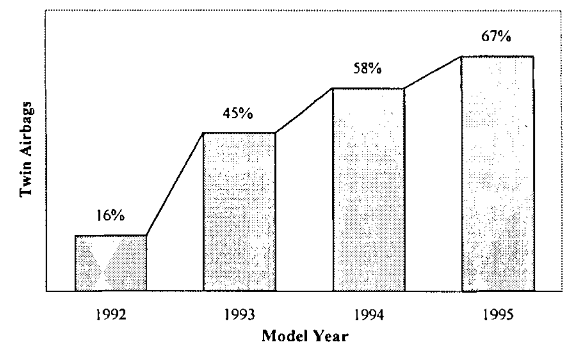


Figure 2. Percentage of cars fitted with twin airbags - airbag material.

90% of the airbag cars were fitted with safety belt pretensioning systems (buckle pretensioners or retractor pretensioners), and some models additionally had webbing grabbers.

Car Occupancy and Restraint Rates

The occupancy of the airbag-equipped cars and the restraint rates of the occupants are given in Table 7. The restraint rates in the case of the front

Table 7.
Car Occupancy and Restraint Rates Distributed over Adults and Children

Type of Restraint		Driver	Passenger	Rear left	Rear centre	Rear right	Total Rear
Adults	3-point belt	226 98%	92 98%	16		18	34 49%
	Lap belt				2		2 3%
	Unrestr.	5 2%	2 2%	1	1	7	9 13%
Children 0-11 Years	3-point belt			3			3 4%
	Lap belt				3		3 4%
	CRS			7	2	7	16 23%
	Unrestr.				1	1	2 3%
Total		231 100%	94 100%	27	9	33	69 100%

CRS = Child restraint systems

occupants were very high, almost 98%, and exceeded the values which are recorded by the Federal German Road Research Institute (FGRRI) at regular intervals [4] (September 1995: drivers 93%, front seat passengers 94%). Although the belt wearing rate on the rear seat was lower, 84%, than on the front seats, it was 20 percentage points above that observed by the FGRRI (September 1995: rear seat occupants 64%).

In the available accident material the front seats were occupied by adults only, so that at present it is not possible to say anything on the subject of "the risk of children in rearward facing child seats on the passenger side in the case of passenger airbag deployment" from real-life accidents. Altogether there were 24 children on the rear seats up to 11 years of age, 16 of them restrained by child restraint systems (CRS), while 6 were wearing 3-point or lap belts and two were unrestrained.

AIRBAG DEPLOYMENT

Deployment Frequency, Occupancy of the Front Passenger Seat

The available accident material of 249 airbag-equipped vehicles contains 188 cars in which the driver airbag fired; the passenger-side airbag was deployed in 82 out of 106 cars (in which a passenger airbag was installed). Both the driver and the passenger airbag thus fired in about three out of four accidents (Table 8).

Table 8.
Deployment Frequency

	Airbag Fired		Total
	Yes	No	
Driver	188	61	249
	76%	24%	100%
Passenger	82	24	106
	77%	23%	100%

In 34 out of 82 accidents with passenger airbag deployment there was no passenger in the car, which means that passenger airbag deployment occurred in 42% of the cases quite unnecessarily. These redundant deployments of the passenger airbag constitute - at least in the case of cars that are not a complete write-off - a not inconsiderable cost factor when they are repaired and should therefore be avoided, for example by passenger presence detection.

Impact Area/Overlap in the Event of a Driver Airbag Deployment

In 150 out of a total of 188 accidents with driver airbag deployment the impact area and the overlap was known (Figure 3). The front of the car was struck most frequently, in 92% of the cases, but in 8% of the accidents driver airbag deployment also occurred in side impacts, as there was considerable deceleration in the car's longitudinal direction.

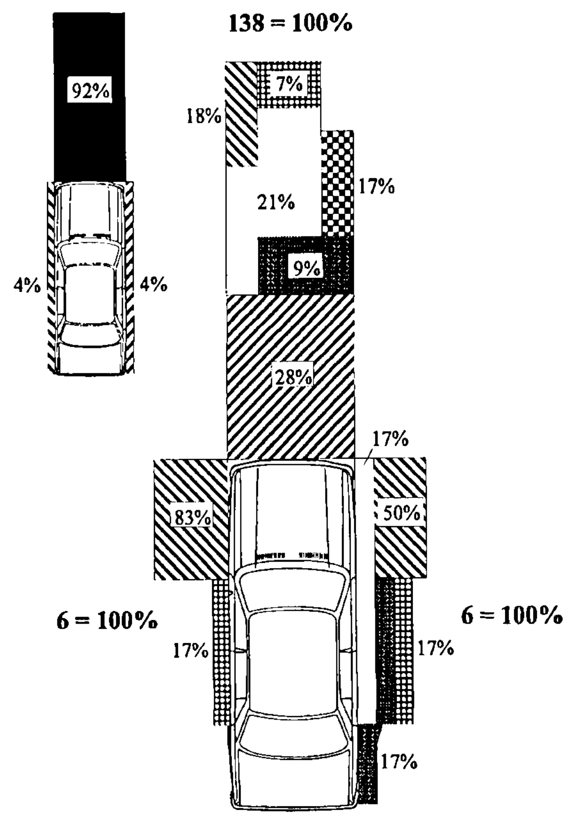


Figure 3. Impact area/overlap in the event of a driver airbag deployment.

How frequently individual areas of the car's front were struck is also shown in Figure 3. An overlap on the left side was most frequent with 39% (21% two-thirds left, 18% one-third left), followed by complete overlap (28%) and overlap on the right side (17% one-third right, 9% two-thirds right); the least frequent (7%) was an impact concentrated on the middle of the car's front (for example, in collisions with a tree or post).

Direction of Principal Force in the Event of a Driver Airbag Deployment

In theory the airbag deploys in impacts in which the direction of principal force is within a range of ± 30° to the car's longitudinal axis; between ± 30° and ± 60° there is a tolerance area in which a deployment is still possible, and above this airbag deployment does not occur.

These theoretical values were largely confirmed in the available material. The impact angle presented in Figure 4 gives the direction of principal force taking into account the car's mass, the collision type and the crash velocity. In 96% of the cases the airbag deployed in a range of 0° ± 45° to the car's longitudinal axis (11, 12 and 1 o'clock).

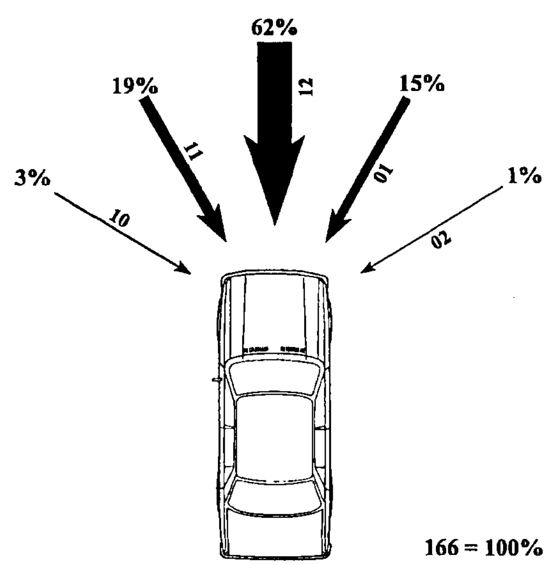


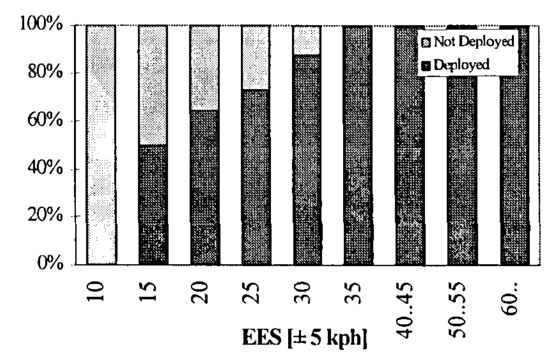
Figure 4. Impact angle in the event of a driver airbag deployment.

In 62% of the cases the direction of principal force even occurred in a very narrow sector of 0° ± 15°. Thus nearly all the deployments took place in the theoretically specified deployment area of ± 30° to the car's longitudinal axis, and only in a few cases was the impact angle larger.

Deployment of the Driver Airbag as a Function of the EES

According to information supplied by the manufacturers [5], the typical deployment threshold ranges between a Δv of 15 to 30 kph (frontal crash against a wall, 100% overlap). Since, however, Δv can only be ascertained from a complete reconstruction of the accident (which also presupposes detailed information on the accident opponent, which was not the case in the available accident material), for the airbag-equipped cars the energy screen method [6] was used to determine the EES [7], which is generally acknowledged to correlate well with the injury severity of the occupants [8], a very essential criterion in accident research.

Figure 5 shows for all cars within an EES class whether the driver airbag deployed or not. In the case of EES values of around 10 kph the airbag did not deploy in any of the cars, while from an EES of 35 ± 5 kph all the airbags were deployed. It is



EES ± 5 kph	10	15	20	25	30	35	40..45	50..55	60..
Number	2	30	31	26	24	16	14	7	6

Figure 5. Deployment of the driver airbag as a function of the EES.

noticeable that even in the case of cars that were only involved in relatively light accidents (EES between 15 and 20 kph) the airbags deployed in at least half of the cases. This early activation of the airbags despite an only slight accident severity can result in injuries (e.g. bruises, abrasions, burns) which would not have occurred if the airbag had not deployed and should therefore be avoided (see Appendix II). Also for reasons of cost (additional repair costs for the exchange of the airbag), a premature airbag deployment should be prevented.

Inadvertent Firing

The problems of inadvertent firing (while driving or stationary) is not dealt with in this paper, as only a few individual cases are at present available to the VdS accident research and they require analysis in greater depth.

INJURIES TO THE FRONT OCCUPANTS

Distribution of the Maximum Injury Severity MAIS

The distribution of the maximum injury severity MAIS [9] for drivers and front seat passengers, presented in Figure 6, shows a relatively low proportion of serious to critical injuries (MAIS 3-5) in the available accident material when the airbag deployed, but even fatal injuries (MAIS 6) can occur in individual cases. Only in extremely severe frontal collisions with considerable deformation of the passenger compartment and/or an oblique collision are the limits of the front airbag's protective effect reached, as an in-depth analysis of the three accidents with fatally injured front occupants showed.

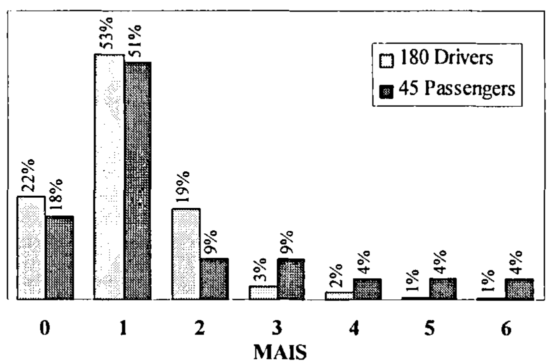


Figure 6. Distribution of the maximum injury severity MAIS (in case of airbag deployment).

In the accidents with airbag deployment serious to critical injuries to the passenger occurred more frequently than to the driver; this had already been observed in the VS 90 accident material [2] (belted occupants only). In the chapters 'Protection potential of the passenger airbag' and 'Individual injuries to the passenger with airbag deployment' this point is gone into in greater detail.

If only those accidents are considered in which there were two restrained occupants on the front seats of the car and both airbags deployed (42 cases with the same accident severity for driver and passenger), the tendency for the passenger to sustain more severe injury is again corroborated (Figure 7). If a disaster case is not taken into account (driver MAIS 4; passenger MAIS 6), the maximum injury severity for the driver shows only values between 0 and 2, while in eight cases serious

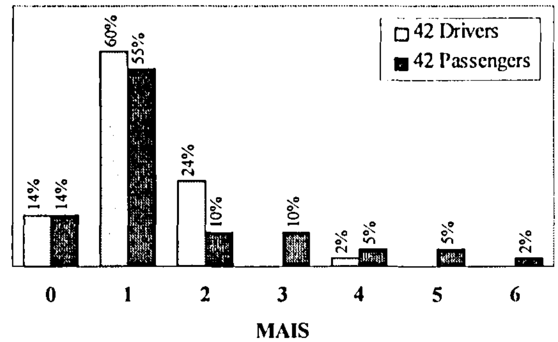


Figure 7. Distribution of the maximum injury severity MAIS of driver and passenger with deployed twin airbags.

to critical injuries (MAIS 3-5) occurred to the passenger, which in one case even proved fatal after four days.

Protection Potential of the Driver Airbag

To examine the protective effect of the complete system of "airbag plus 3-point belt" compared with "3-point belt only" it is essential to take the accident severity into consideration. Since the EES was not ascertained in the VS 90 large-scale body of material, for a comparison of the two bodies of accident material the "degree of damage" was used as a measure of the accident severity. It has to be taken into consideration in this comparison that most airbag-equipped cars were only one or two years old at the time of the accident and belonged to a newer generation of cars (87% built in 1992-95) than those in the VS 90 and therefore most of them had more rigid deformation structures. This means that the airbag-equipped cars with the same degree of damage were involved in an accident with a slightly higher accident severity than the cars in the VS 90.

The maximum injury severity MAIS to belted drivers in frontal impacts with airbag deployment as a function of the degree of damage is shown in Table 9. In a comparison with the accident material of the Vehicle Safety 90 (Table 10) it becomes clear that with the same car damage the drivers are far more rarely severely injured or killed when the airbag deploys. This becomes especially clear in the 3 to 4 range of degrees of damage, i.e. a relatively high accident severity. In the airbag material, only in the case of high car deformation (DD 4) did serious MAIS 3-4 injuries occur, and there were no fatal injuries at all with this injury severity. Only in

Table 9.
Airbag Material - Distribution of the Driver's MAIS as a Function of the Degree of Damage

Driver Airbag Deployed, Frontal Impact, Driver Belted								
Degree of Damage	**MAIS**							**Total**
	0	**1**	**2**	**3**	**4**	**5**	**6**	
DD 1 - light								
DD 2 - moderate	35%	53%	12%					100% = 34
DD 3 - severe	13%	67%	20%					100% = 64
DD 4 - extreme	8%	24%	44%	12%	12%			100% = 25
DD 5 - total			50%				50%	100% =2

Table 10.
VS 90 - Distribution of the Driver's MAIS as a Function of the Degree of Damage [2]

VS 90 Frontal Impact, Driver Belted								
Degree of Damage	**MAIS**							**Total**
	0	**1**	**2**	**3**	**4**	**5**	**6**	
DD 1 - light	52%	47%	1%					100% = 394
DD 2 - moderate	34%	63%	3%	0.2%				100% = 2,178
DD 3 - severe	12%	68%	16%	3%	0.7%	0.1%	0.3%	100% = 1,402
DD 4 - extreme	1%	28%	33%	20%	9%	3%	6%	100% = 391
DD 5 - total		4%	13%	9%	20%	7%	47%	100% = 54

so-called disaster cases with extreme intrusion into the passenger cell (degree of damage 5) was a fatally injured driver observed in the airbag material.

Individual Injuries to the Drivers when the Airbag Deploys

It emerges from Table 9 that 39 drivers sustained MAIS 2+ injuries in spite of airbag deployment; Table 11 gives the AIS values of the injuries to the individual parts of the bodies of these 39 drivers, and it also shows which individual injury led to the MAIS value.

In the MAIS 3-6 range (8 cases), only in one case (Case No. 108, Appendix III) was the MAIS value determined by the AIS 6 injury to the head, but another fatal injury to the vertebral column also occurred. In this case, as already described above, there was extreme intrusion into the passenger cell, making survival practically impossible. This means that the airbag can largely prevent serious to critical injuries to the belted driver, and only in the case of the most serious frontal collisions is the limit of the protective effect reached.

In the area of moderate MAIS 2 injuries (31 cases), in five cases I° skull/brain trauma (concussion) with an AIS of 2 determined the maximum injury severity. The reason for the head injuries were striking the side window (oblique impact from the left front) and two further contacts with hard parts of the passenger cell during a rollover following the collision and when the car drove under a truck. In these three cases the deployed steering wheel airbag could not have saved the driver from the head injury. In two of the five cases described the precise cause of concussion could not be ascertained.

Thorax injuries determined the MAIS value of altogether 12 of the 39 drivers. Typical injuries were in particular sternum fractures (AIS 2) and fractured ribs (AIS 2-3). A close analysis of the 12 cases established that especially in the case of older drivers injury to the chest predominates; thus 10 of the 12 drivers were aged 50 or above. This shows that the system of airbag plus 3-point belt is still not ideally adapted to the possible driver population.

Table 11.
Individual Injuries to the Drivers which Determined the MAIS

Driver Airbag Deployed, Frontal Impact, Driver Belted

Case No.	108	129	55	201	36	72	71	185	237	235	171	62	56	210	208	59	48	143	139	132
MAIS	**6**	**4**			**3**				**2**											
Body region	**AIS Value**																			
Head	6	2		3			1		2	2	2	2	2	1	1	1				
Neck	6	1					1	1	1		1		1				1	1	1	
Chest		4	4		3		1		1		1			2	2	2	2	2	2	2
Pelvis/Abdomen		3		4										1						2
Shoulder		2						2	1				1		1				1	
Upper arm		2				3	3		1											
Lower arm							1		1											
Hand		1		1						1						1				
Upper leg	3		3																	
Knee			1		1	1	1	1	1					1						1
Lower leg	3	3		2				3								1				1
Foot						1	1	2						1						
T/L Spine					2				1		1		1							
Age	33	24	69	45	51	43	35	58	31	30	47	39	44	59	51	56	50	50	64	35

Case No.	182	144	198	74	104	170	202	51	82	15	29	115	21	220	189	100	25	66	64	
MAIS	**2**																			
Body region	**AIS Value**																			
Head	1		1	1	1	1					1	1				1			1	
Neck					1	1													1	
Chest	2	2	1										1	1				1	1	
Pelvis/Abdomen			2			1										1				
Shoulder				2		1							1							
Upper arm				1	2			1			1		1					1		
Lower arm				2	2	2		1												
Hand				1			2	2	2	2		1		1		1				
Upper leg																				
Knee	2		1			1					2	2		1				1	1	
Lower leg											2		2							
Foot			2								2	2	1	2	2	2	2	2		
T/L Spine		2																	2	
Age	54	70	25	27	47	45	47	31	52	45	29	42	56	65	48	25	19	48	57	

For this reason the interplay of airbag and belt in conjunction with belt pretensioners and belt-force limiters still has to be optimised in the future.

It was, however, injuries to arms and legs which by far most frequently determined the MAIS value in the case of the belted driver and airbag deployment. The arms mostly sustained fractures of the hand and the forearm, while in the case of the feet it was most frequently foot fractures that were observed. Through the additional protective effect of the driver airbag, especially in the region of the head, a new weak point thus appears in the passive safety of cars, namely the still inadequate protection of the extremities.

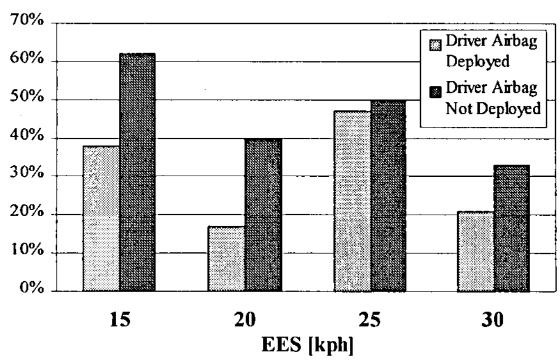


Figure 8. Frequency of CVC distortions dependent on deployment/non-deployment of airbag.

The neck injuries presented in Table 11 are - with the exception of one fatal fracture - exclusively cervical vertebral column (CVC) distortions (AIS 1) and thus only in one case was the MAIS value determined by the injury to the neck. As the light CVC injuries are frequently observed with a low accident severity (low EES), the influence of the airbag on the frequency of CVC injuries (AIS 1) were examined for an EES between 15 and 30 kph. Figure 8 shows for each EES range how frequently the drivers sustained a CVC distortion. It turned out that when the airbag deployed slight CVC AIS 1 injuries occur more rarely. Intercepting the head by the airbag apparently prevents a hyperflexion of the cervical and vertebral column and thus reduces the occurrence of these relatively harmless injuries, which, nevertheless, result in considerable discomfort for those affected.

Protection Potential of the Passenger Airbag

In Figures 6 and 7 it has already been established that the passenger in cars with twin airbags is more often severely injured than the driver; in the following chapter a comparison will be made between "passengers who were protected by belt and airbag" and "passengers who were only belted " [2] (Tables 12 and 13). In contrast to the obvious protective effect of the driver airbag (see Tables 9 and 10), Tables 12 and 13 do not show that the passenger airbag provides any clear protection. In accidents with a degree of damage of 3-4, i.e. a relatively serious accident severity, 32% of the passengers were, for example, seriously to fatally injured (MAIS 3-6) in spite of airbag deployment, whereas in cars without an airbag (Table 13) only about 10% of the passengers sustained injuries of such severity. Here, however, it has to be taken into consideration that the airbag material contains only a small number of severe cases.

In the following chapter a detailed analysis of the individual injuries is carried out to gain a clearer insight into the principal injuries and the possible protective effect of the passenger airbag.

Table 12.
Airbag Material - Distribution of the Passenger's MAIS as a Function of the Degree of Damage

Passenger Airbag Deployed, Frontal Impact, Passenger Belted								
Degree of Damage	**MAIS**							**Total**
	0	**1**	**2**	**3**	**4**	**5**	**6**	
DD 1 - light								
DD 2 - moderate	17%	83%						100% = 6
DD 3 - severe	11%	50%	17%	11%	6%	6%		100% = 18
DD 4 - extreme	20%	20%	10%	20%	10%	10%	10%	100% = 10
DD 5 - total								

Table 13.
VS 90 - Distribution of the Passenger's MAIS as a Function of the Degree of Damage [2]

VS 90 Frontal Impact, Passenger Belted								
Degree of Damage	MAIS							Total
	0	1	2	3	4	5	6	
DD 1 - light	37%	63%						100% = 71
DD 2 - moderate	25%	70%	5%	0.1%	0.1%			100% = 680
DD 3 - severe	8%	65%	21%	4%	1%	0.2%	1%	100% = 539
DD 4 - extreme	3%	36%	33%	13%	7%	5%	3%	100% = 147
DD 5 - total			15%	30%	22%	11%	22%	100% = 27

Individual Injuries to the Passenger with Airbag Deployment

Table 14 shows for the passenger, just as Table 11 shows for the driver, which individual injuries determine the MAIS value. In 12 out of 13 MAIS 2+ cases the individual injuries to the passenger were known, and only in one case (No. 201), which was a disaster case, was the information simply "passenger killed".

On account of the absence of "steering wheel injury risk" on the passenger side, the MAIS value in the case of the passenger is less frequently determined by head injuries. In the available accident material there was only one such case (No. 25, Appendix IV). The AIS 5 head injury

Table 14.
Individual Injuries to the Passengers which Determined the MAIS

Passenger Airbag Deployed Frontal Impact, Passenger Belted													
Case No.	201	25	142	226	220	182	144	59	100	79	52	33	78
MAIS	**6**	**5**		**4**		**3**				**2**			
Body region	**AIS Value**												
Head		5	1						1				
Neck		1		1			1	1		1			
Chest		4		4	4	3	3	3		2	2	2	1
Pelvis/Abdomen			5		3	1		1	1				
Shoulder		1	2										
Upper arm				3									
Lower arm		2										2	
Hand			2		1				1				2
Upper leg		3							3				
Knee													
Lower leg		2											
Foot													
T/L Spine			2		2			2					
Age	82	35	34	66	63	54	58	53	26	56	n.a.	59	47

n.a. = not ascertained

came about because the belted passenger moved past the deployed full-size airbag towards the centre of the car as a result of a strong impact against a tree on the left side and struck the console between the seats and the instrument panel with considerable force, sustaining III° skull and brain trauma, a lung contusion as well as fractures to one arm and both legs. The driver, however, escaped with a simple ankle joint fracture (AIS 2).

In the second MAIS 5 case (No. 142, Appendix V) very severe injuries in the region of the pelvis and abdomen occurred. The slightly-built 34-year-old female passenger (160 cm, 50 kg), restrained by a 3-point belt, sustained a ruptured small intestine (AIS 5) and colon (AIS 4), which was only diagnosed after two days and then treated in intensive care. In [10], too, comparable so-called "concealed internal injuries" following accidents with airbag deployment are reported.

What is striking about Table 14 is that very frequently thorax injuries determine the MAIS value. These injuries are, as noticed in the case of the drivers too, on the one hand, due to the still inadequate adjustment of the seat belt (the absence of belt-force limiters) and airbag, and, on the other, to the advanced age of the passengers and the reduced biomechanical loading in the thorax area. Thus all two male and six female passengers in which injuries to the thorax determined the MAIS value were aged between 53 and 66. The following individual injuries (AIS 3-4) occurred: more than three rib fractures on either side (AIS 4, n=2), lung contusion (AIS 4, n=1), haematothorax (AIS 3, n=2) and cardiac contusion (AIS 3, n=1).

Injuries to the lower and upper extremities determining the MAIS value were of minor importance.

Injury Risks to Spectacle Wearers and Smokers when the Airbag Deploys

In altogether 188 accidents in which airbag deployment occurred 74 drivers and nine passengers were wearing spectacles, one driver contact lenses, and another was smoking (Table 15). In the available material the smoker was not injured, although in individual cases serious eye injuries are possible, as is shown in the case of a pipe smoker described in [11]; in tests, too, [12] evidence of a certain risk of burn injuries to smokers was provided. In the case of spectacle wearers the injury severity was limited exclusively to slight AIS 1 injuries, most of which were bruises.

Table 15.
Injury Risks to Spectacle Wearers and Smokers when the Airbag Deploys

Airbag deployed	Driver		Passenger	
Special Risk Factors	Number	Injuries Caused	Number	Injuries Caused
no	94	-	31	-
Wearing Spectacles	74	12	9	4
Wearing Contact lenses	1			
Smoking Pipe				
Smoking Cigar/Cigarette	1			
Phoning (Handy)				
Others				
n.a.	18	-	8	-
Total	188	12	48	4

n.a. = not ascertained

Injuries (AIS 1) of Spectacle Wearers	12 Driver (100%)		4 Passenger (100%)	
	Number	Rel. Share	Number	Rel. Share
Eye injury	1	8%		
Contusion at eyes surface	3	25%	2	50%
Nasal bone fracture	2	17%		
Nose contusion	6	50%	1	25%
Tear wound/cuts	2	17%	1	25%

Repeated Injuries Possible

Only one case was, however, rather more critical in that a metal ear piece broke off and entered the eye - but even this injury was only rated as AIS 1. Here, too, the results of the study are basically confirmed by crash tests [12], in which no negative effects were observed for spectacle wearers from an airbag deployment. In very rare cases, however, more serious injuries than only bruises can occur, as the above-mentioned case with the broken metal ear piece shows.

Injuries in the Event of Non-Deployment of the Airbag

In all 61 cases in which the airbag did not deploy (cases with slight accident severity and accidents with an overturn and slight deceleration in the direction of the car's longitudinal axis) both the driver and the passenger were restrained. More than 70% of the drivers and passengers remained uninjured or were only slightly injured (MAIS 1). Critical, life-threatening as well as fatal injuries (MAIS 5-6) occurred in none of the cases (Figure 9).

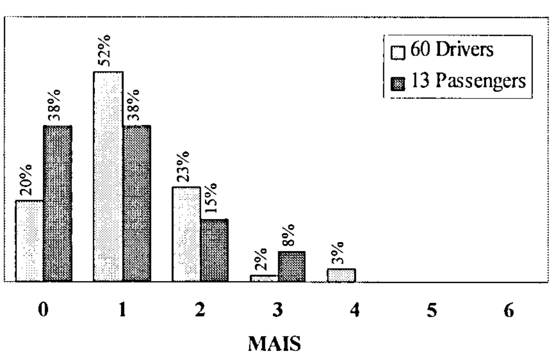


Figure 9. Distribution of the maximum injury severity MAIS in the event of non-deployment of the airbag.

In three cases with a driver's injury severity of MAIS 3-4 the impact took place at a very oblique angle, outside the ± 30° range to the car's longitudinal axis, so that the airbag did not fire (Appendix VI). A careful analysis showed that any protective effect of an airbag deployment would have been very doubtful or extremely unlikely, as one car rolled over once and the other several times. The primary causes of the injuries were head contact with the roof and the restraining forces of the seat belt, which did, however, prevent the occupants from being ejected from the car and thus possibly from being fatally injured.

In another case (passenger MAIS 3) the accident severity just failed to reach the deployment threshold in a frontal collision. The slightly-built woman passenger (160 cm, 50 kg) sustained belt-induced injuries to the chest area, which would very probably not have been prevented even if airbag deployment had occurred, as is confirmed by the findings of the study of accidents with airbag deployment, in which a high proportion of thorax injuries were observed to both front seat occupants in spite of airbag deployment.

In Table 16 the maximum injury severity of restrained drivers is presented as a function of the EES in accidents with a frontal impact but without airbag deployment. More than three-quarters of the drivers remained uninjured or were only slightly injured (MAIS 0-1), and almost one-quarter sustained moderate injuries (MAIS 2). An in-depth analysis of the individual AIS 2 injuries showed in three cases belt-induced sternum fractures, which could only have been largely prevented by belt-force limiters and a better interplay of seat belt and airbag. In the other five cases the driver's head struck the steering wheel, resulting in concussion and, in one case, a small woman driver sustained nosebone and cheekbone fractures. The contact of the head with the steering wheel, in spite of relatively slight accident severity (EES ≤ 30 kph), would suggest a forward sitting position with very little space between the driver and the steering

Table 16.
Maximum Injury Severity in the Event of Non-Deployment of the Driver Airbag as a Function of the EES

Driver Airbag Not Deployed Frontal Impact, Driver Belted								
EES	MAIS							
± 5 kph	0	1	2	3	4	5	6	Total
10		2						2
15	1	7	5					13
20	3	5	2					10
25		5						5
30		2	1					3
35								
40...45								
50...55								
≥ 60								
Total	4	21	8					33

wheel. Airbag deployment in such cases, which are under circumstances to be described as 'out of position', could result in at least equally severe injuries to the driver, unless an intelligent airbag could detect the short distance between the occupant and the steering wheel and gently intercept the head.

SUMMARY AND CRITICAL EVALUATION OF THE ESSENTIAL FINDINGS

249 real-life accidents in which airbag-equipped cars were involved in Germany are the basis of this paper. The available material comprises a broad spectrum of different car classes, from compact cars to executive limousines, and provides indications of the weak and strong points of current airbag systems and of areas for future improvements.

The protective effect of the airbag as a passive restraint system to supplement the seat belt was undoubtedly confirmed for the driver; it proved to be especially effective in accidents with a relatively high degree of accident severity (Appendices VII and VIII). For the belted and airbag-protected front seat passenger, on the other hand, the available accident material did not provide any clear evidence of a reduction in the injury risk. A direct comparison between the belted front occupants (altogether 42 cases in which deployment of both airbags occurred) also showed a noticeably higher risk of injury for the passenger.

In frontal collision with airbag deployment, the maximum injury severity (MAIS 2+ cases) of belted drivers was on the whole determined by injuries to the lower, in some cases the upper, extremities, and less often by injuries to the head or chest. Only in "disaster cases" were severe to critical head injuries observed. With the passenger it was frequently the chest injuries that determined the MAIS value. The relatively advanced age of the passengers was a contributory cause of this result. This could be remedied, on the one hand, by belt force limiters and, on the other, by a better attunement of the system of "airbag plus 3-point belt".

In the available cases, the limits of the protective effect of front airbags were reached only where there was extreme deformation of the passenger compartment and a high accident intensity, but also a strongly angled impact or an overturn. In these cases, the airbag cannot - or can no longer - develop its protective effect both for the driver and the passenger, as it either does not fire at all or cannot take effect because there is no space left in which to survive. Severe injuries caused by strongly angled or side impacts could be reduced by side airbags in the head and thorax areas.

In the present material eight frontal collisions occurred without airbag deployment (EES $\leq$ 30 kph), in which the belted driver sustained moderate injuries (AIS 2). AIS 3+ injuries to the belted driver without airbag deployment were not recorded in the material. Most of the AIS 2 injuries were concussion resulting from head contact with the steering wheel. Although the EES was between 15 and 30 kph, i.e. the accident severity was relatively slight, the head shifted so far forward that impact with the steering wheel occurred, which can only be explained by the fact that the driver must have been sitting very close to the steering wheel. Under these circumstances an airbag deployment (with an airbag of the present generation) would tend to be an added risk to the driver rather than protection from concussion. Here only an intelligent airbag which can detect the driver's sitting position could lead to a reduction in injuries.

In 42% of the cases in which the passenger airbag fired there was no passenger on the front seat, so that the airbag deployed quite unnecessarily. In the future, intelligent airbag systems (with passenger presence detection) should avoid these unnecessary deployments. An intelligent passenger-side airbag should also be able to detect a rearward-facing child seat on the front passenger seat and deactivate the airbag as required.

Almost all airbags fired in the specified deployment range of $\pm$ 30° to the car's longitudinal axis, and not one single case was observed above an EES of 30 kph in which airbag deployment did not occur; above this accident severity the airbag fired every time. But the airbag also fired in half of the very slight accidents (EES = 15 $\pm$ 5 kph). This premature firing led in some cases to injuries, such as burns, which would not have occurred to a belted driver if the airbag had not deployed. From this point of view, but also for reasons of cost, the deployment threshold should, therefore, be around 25 to 30 kph in future car models. This relatively high deployment threshold can, however, only be achieved if the airbag is adjusted to the **belted** occupants.

But quite independently of the height of the deployment threshold, the development of future airbag systems should concentrate on occupants who are wearing seat belts and not on unbelted occupants.

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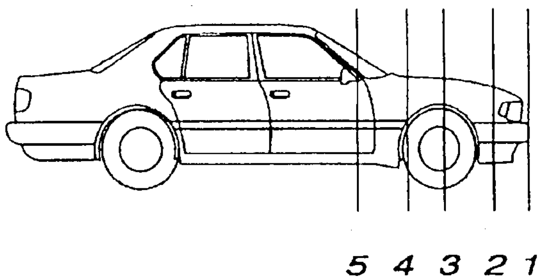
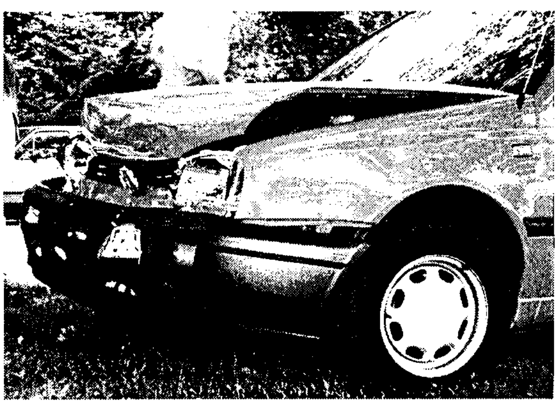
Appendix I

Examples of Degree of Damage Classification

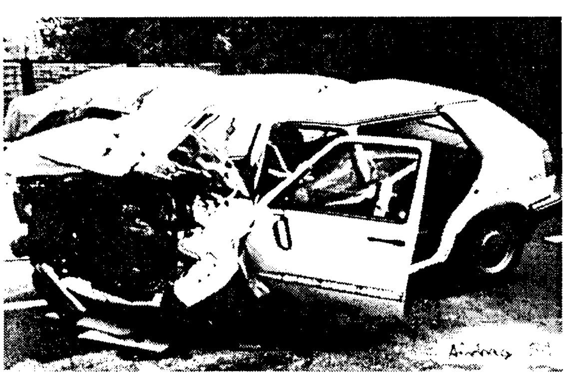


Degree of damage 1 = light
(corresponds to minor scratches, dents, etc.)

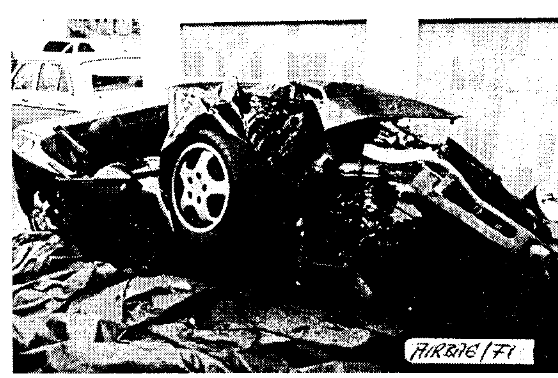


Degree of damage 2 = moderate

Degree of damage 3 = severe



Degree of damage 4 = extreme

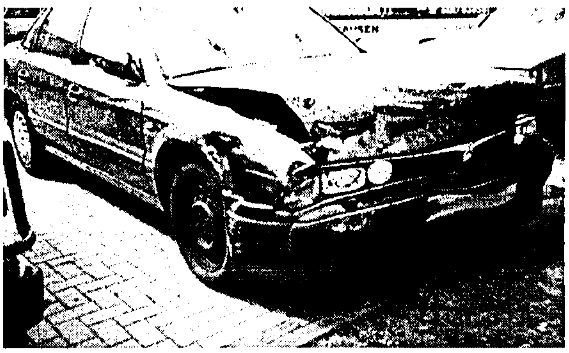


Degree of damage 5 = total

Appendix II

"Airbag Material"
Case No.: 127

Airbag car: Mitsubishi Sigma, '94 model
Driver airbag (full size)



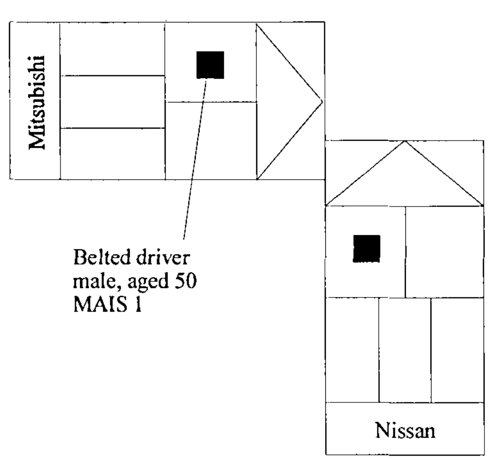
EES = 15 ± 5 kph
Driver airbag fired

Injuries to the belted driver:

- Neck/spine whiplash AIS 1
- Thorax contusion AIS 1
- Burn to the left hand AIS 1

Accident opponent: Nissan Bluebird

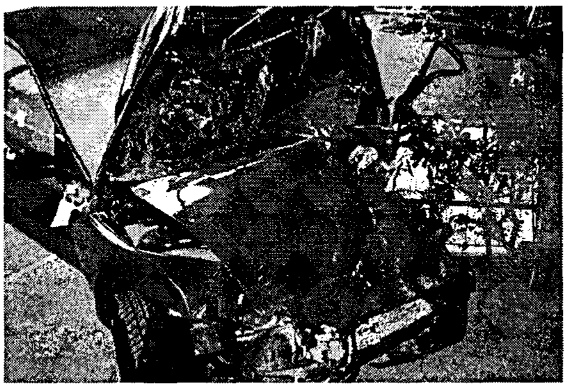
Collision position:

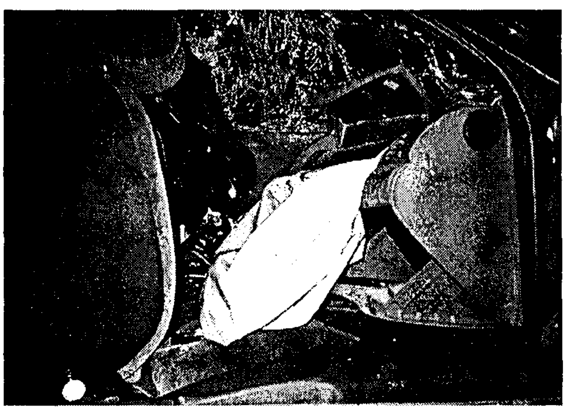


Appendix III

"Airbag Material"
Case No.: 108

Airbag car: Ford Mondeo, '94 model
Driver and passenger airbags (small size)





EES = 60 ± 5 kph
Driver and passenger airbags fired

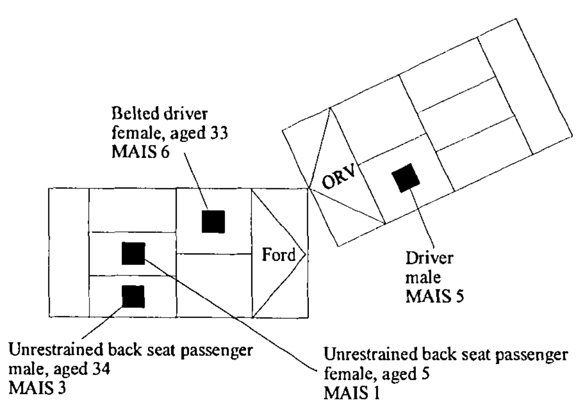
Injuries to the female, belted driver:

• Fatal head injuries	AIS 6
• Fractured neck	AIS 6
• Multiple thigh fractures on both sides	AIS 3
• Multiple lower leg fractures on both sides	AIS 3

Accident opponent: Off-Road Vehicle (ORV)

Injuries to the driver:
- Paraplegia, MAIS 5

Collision position



Appendix IV

"Airbag Material"
Case No.: 25

Airbag car: BMW 325 tds, '94 model
Driver and passenger airbags (full size)

EES = 65 ± 5 kph
Driver and passenger airbags fired.
Single car accident in which car struck a tree.

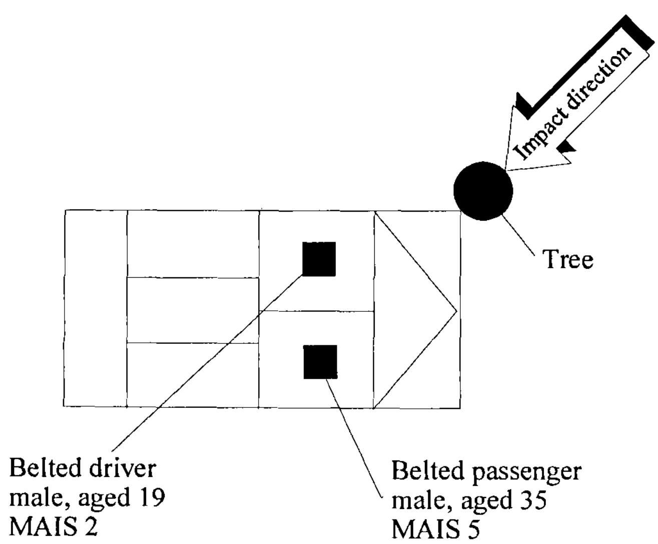
Injuries to the belted driver:

• Fractured ankle joint	AIS 2

Injuries to the belted passenger:

• III° skull-brain trauma	AIS 5
• Facial bruises	AIS 1
• Contusion of neck/spine	AIS 1
• Lung contusion	AIS 4
• Shoulder bruises	AIS 1
• Fractured lower arm, left	AIS 2
• Dislocated comminuted fractures of both thighs	AIS 3
• Fractured fibula, left	AIS 2
• Died after 4 days	

Collision position:



Appendix V

"Airbag Material"
Case No.: 142

Airbag car: Opel Astra, '94 model
Driver and passenger airbags (full size)

EES = 50 ± 5 kph
Driver and passenger airbags fired

Injuries to the belted driver:

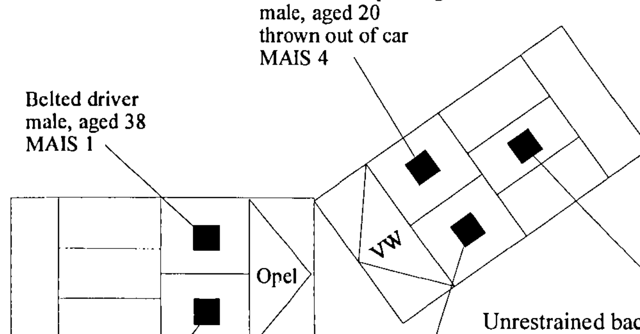
• CVC whiplash	AIS 1
• Pelvic bruise	AIS 1
• Shoulder bruise, left	AIS 1
• Hand bruise, left	AIS 1
• Knee bruise, left	AIS 1
• Foot bruise, left	AIS 1

Injuries to the female, belted passenger:

• Burns to the facial skin	AIS 1
• Facial bruises	AIS 1
• Double rupture of small intestine	AIS 5
• Colon rupture	AIS 4
• Concomitant pancreatitis	
• Concomitant pneumonia	
• Fractured collarbone	AIS 2
• Fractured upper plate of the thoratic vertebral body 11	AIS 2
• Fractured metacarpal, left	AIS 2

Accident opponent: VW Scirocco
EES = 60 ± 5 kph

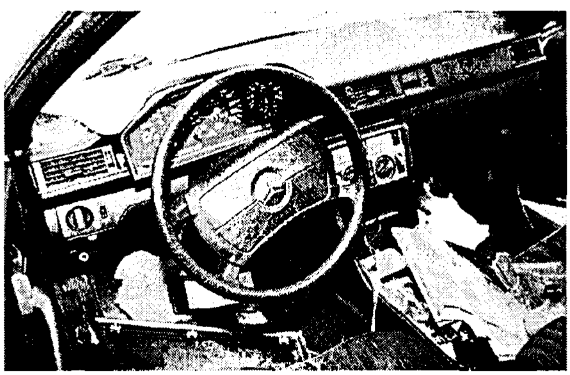
Collision position:



Appendix VI

"Airbag Material"
Case No.: 5

Airbag car: Mercedes 300 CE, '89 model
Driver airbag (full size)



Single car accident with side impact and several roll-overs.
Airbag did not fire.

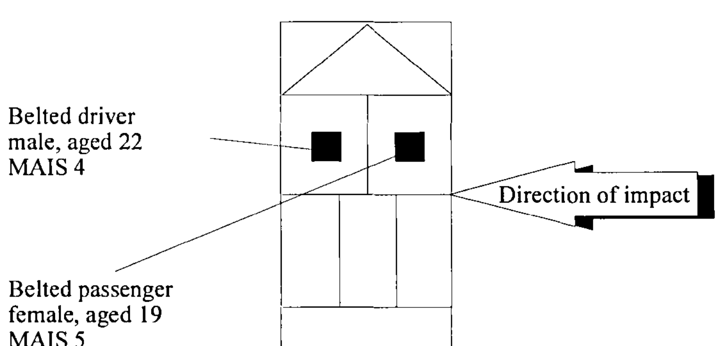
Injuries to the belted driver:

• III° skull-brain trauma	AIS 4
• Brain haemorrage	AIS 4
• Fractured collarbone, left	AIS 2
• Haematoma on right thigh	AIS 1

Injuries to the belted passenger (no airbag installed):

• III° skull-brain trauma	AIS 5
• Pneumothorax with two collapsed lungs	AIS 4
• Open fracture of the upper arm	AIS 3

Collision position:



Appendix VII

"Airbag Material"
Case No. : 115

Airbag car: BMW 320i, '92 model
Driver airbag (full size)

EES = 65 ± 5 kph
Driver airbag fired

Injuries to the belted driver:

• Nose bruise	AIS 1
• Fractured finger, left hand	AIS 1
• Fractured kneekap	AIS 2
• Fractured heel bone	AIS 2

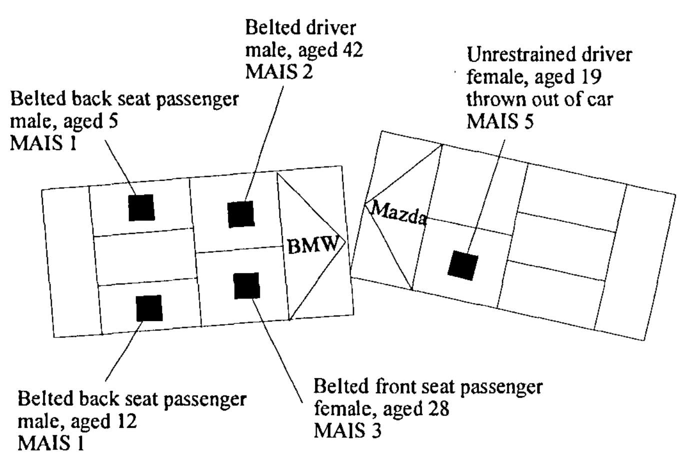
Injuries to the female, belted passenger (no airbag installed):

• Concussion	AIS 2
• Lacerated and bruised eyebrow	AIS 1
• Lacerated and bruised left hand	AIS 1
• Lacerated liver	AIS 3
• Ruptured colon	AIS 3

Accident opponent: Mazda 323, '83 model

Injuries to the unrestrained driver:
- Unrestrained driver was thrown out of the car, MAIS 5

Collision position:



Appendix VIII

"Airbag Material"
Case No.: 198

Airbag car: VW Golf GTI, '94 model
Driver and passenger airbags (small size)

EES = 65 ± 5 kph
Driver and passenger airbags fired

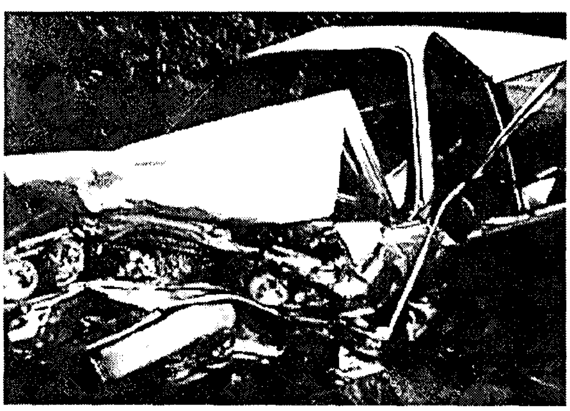
Injuries to the belted driver:

• Chin laceration	AIS 1
• Thorax contusion	AIS 1
• 4 fractures of the pelvis	AIS 2
• Abdominal bruises	AIS 1
• Laceration to both knees	AIS 1
• Metatarsal fracture, right	AIS 2

Injuries to the belted passenger:

• Haematoma to the right eye	AIS 1
• Chest bruises	AIS 1
• Haematoma to the left knee	AIS 1

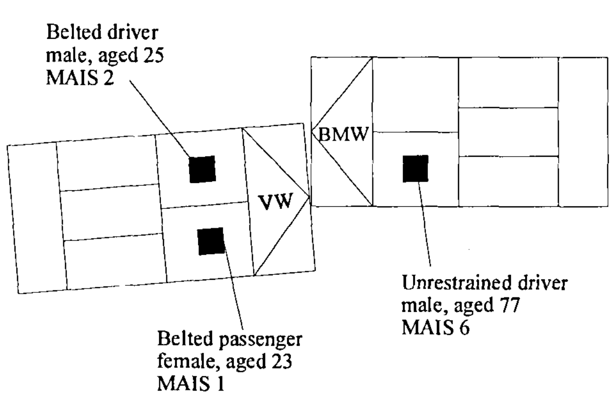
Accident opponent: BMW 320i, '84 model



EES = 60 ± 5 kph

Injuries to the unrestrained driver:
- Driver died at the scene of the accident, MAIS 6

Collision position:



AIR BAG DEPLOYMENT CRASHES IN CANADA

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ABSTRACT

In the fall of 1993 Transport Canada initiated a major field accident study to examine the injury experience of occupants protected by supplementary air bag systems. While the initial findings of this study confirm that belted drivers are afforded added protection against head and facial fracture injury in moderate to severe frontal collisions, the findings also suggest that these benefits are being negated by a high incidence of bag-induced injury. Most bag-related injuries consist of AIS 1 facial injuries and AIS 1 to 3 upper extremity injuries. However, they can include AIS ≥3 injuries to other body regions if the occupant is close to the deploying air bag. The incidence of bag-induced injury was greatest among female drivers.

To further quantify the benefits and drawbacks afforded by air bag systems, particularly as a function of collision severity, additional analyses were carried out using US field accident data. Both Canadian and US data examined suggest the protection afforded belted drivers by air bag systems would be greatly enhanced if deployment thresholds were increased. Far greater attention to the protection requirements of female drivers needs to be given in federal regulations addressing restraint system performance.

INTRODUCTION

In response to regulatory initiatives and consumer pressure, the majority of passenger cars and an increasing proportion of multi-purpose passenger vehicles are fitted now with supplementary restraint systems (SRS) in the form of "air bags" in the front outboard seating positions. As of September 1997, the fitment of such devices will be mandatory for all passenger cars sold in the US. By September 1998 this requirement will apply also to all light-duty vehicles (vans, pickups, etc.). While a similar hardware requirement has not been introduced in Canada, a number of major changes to the technical basis of Canada Motor Safety Vehicle Standard 208 have been proposed. The revised performance requirements have only been satisfied consistently by vehicles fitted with supplementary air bag systems [1,2]. Consequently, it is anticipated that most, if not all, manufacturers will elect to meet the proposed Canadian requirements through the fitment of supplementary air bags.

As in the United States, future assessments of vehicle performance will be based on responses measured on two 50th percentile male Hybrid III dummies in a 48 km/h frontal barrier crash test. In Canada only a belted test is proposed. The proposed Canadian performance requirements, however, are more stringent than those in the US. Nevertheless, given the highly integrated nature of the automotive industry in North America, most if not all air bag systems sold in Canada can be expected to be of US design. Such systems may be at odds with the protection requirements of belted occupants from the standpoint of both the deployment threshold and the level of air bag aggressiveness.

One limitation shared by both the US and proposed Canadian requirements is that each front outboard seating position is tested with a dummy of 50th percentile male dimensions in one well-defined seating posture. Consequently, the performance levels achieved in the test may not be indicative of the levels of protection likely to be afforded to occupants of different stature. Of particular concern are possible adverse air bag-occupant interactions if the seat is located forward of the mid seat position. There is evidence from laboratory testing that the proximity of an occupant to the air bag module has a strong influence on the response of the neck and the chest [3,4]. Furthermore, the intervention of the air bag can be expected to introduce a variety of new injury mechanisms such as facial injuries from "bag slap", upper extremity fractures, either directly from the deploying air bag module or from arm flailing, and thermal burns to the face and arms.[5,6].

In order to gain an understanding of the field performance of supplementary air bag systems in Canada, Transport Canada has recently initiated a directed study devoted to documenting the injury experience of occupants involved in crashes resulting in the deployment of an air bag system.

CANADIAN STUDIES

The data collection methodology adopted for this study is similar to that used in the Fully Restrained Occupant Study (FROS) where the emphasis was on evaluating the collision performance of three-point seat belt systems [7]. Each participating collision investigation team is assigned a defined area of operation and case selection criteria. Prompt notification of cases meeting the selection criteria is provided through arrangements with investigating police officers, towing companies, or insurance agency personnel. Once identified, such a collision is rejected only if the capture of a minimum set of data elements cannot be achieved. In this manner, a convenience sample of criteria air bag crashes is produced.

Physical evidence from the collision scene is gathered and used in reconstruction of the vehicle dynamics and collision configuration. The external damage to each of the involved vehicles is documented photographically. Vehicle damage patterns are characterized by assigning Collision Deformation Classifications (CDC) [8], and specific measurements of the crush profiles are obtained. When possible, the equivalent barrier speed (EBS) and the change in velocity (delta-v) for each vehicle are computed as measures of collision severity. EBS is calculated using a variant of the CRASH program [9,10], while delta-v is obtained from CRASH and/or by means of momentum-based calculations.

PRINCIPAL FINDINGS

Transport Canada's Air Cushion Restraint Study (ACRS) was initiated in the fall of 1993. A description of the different phases of the study and associated selection criteria is given in Table 1.

The initial selection criterion was any collision which resulted in the deployment of an air bag system, regardless of injury outcome. This selection criterion was changed subsequently to include only cases where an air bag deployed and the associated occupant was reported by the police to have sustained a "major' injury (transported to a hospital). Additional air bag cases were investigated under the Special Collision Investigation Programme and as part of the pilot/transition phase which preceded the change in the selection criterion noted above. Except for descriptions of selected case studies, the present paper will focus on the injury experience of occupants included in the initial phase of the programme (any SRS deployment). This subset of cases included 409 drivers and 36 front right passengers who experienced deployment of an air bag. Of these individuals, 380 drivers and 32 front right passengers were belted, giving a combined seat belt usage rate of 93%.

Table 1.
Canadian ACRS Study

Study	Selection Criteria
ACRS	Any SRS deployment
ACR2	Pilot/Transition: ACRS to ACR3
ACR3	SRS deployment with "major" injury
ASFS, ASF2	None - Special Investigation Cases

Air bag sensors are designed to respond to longitudinal vehicle deceleration. Frontal crashes accounted for approximately 87% of crashes with air bag deployment. Side impact collisions accounted for another 12% of the cases. The remaining deployments resulted from top and undercarriage impacts.

Estimates of velocity change were available for 304 of the sampled vehicles with belted drivers, while equivalent barrier speeds were available for 359 vehicles with belted drivers. These estimates are summarized in Table 2. From the data presented it can be seen that the majority of cases involved low speed collisions. About 74% of the cases involved an EBS or a delta-v of 25 km/h or less. EBS or delta-v values in excess of 40 km/h accounted for less than 5% of the cases investigated.

Table 2.
Cumulative Distributions of Case Vehicles by Estimated Equivalent Barrier Speed and Delta-V

Velocity Change (km/h)	EBS Cumulative %	Delta-V Cumulative %
≤15	20.9	24.0
≤20	53.5	52.3
≤25	74.4	72.7
≤30	85.2	83.6
≤35	92.8	90.8
≤40	95.0	93.8
≥ 41	100.0	100.0

Table 3.
Distribution of Individual Injuries Sustained By Belted Drivers in Collisions Which Resulted in the Deployment of the Driver-Side Air Bag System

Body Region	Anatomic Structure	AIS1 %	AIS2 %	AIS>=3 %	Total** %
Head	Organs	-	-	1.09	1.09
	Head-LOC*	0.31	0.78	0.31	1.40
	Skeletal	-	-	0.47	0.47
	Skin	0.93	0.16	-	1.09
Face	Organs	0.78	-	-	0.78
	Skeletal	0.62	0.16	-	0.78
	Skin	24.30	-	-	24.30
Neck	Skin	2.34	-	-	2.34
Thorax	Organs	-	0.31	0.62	1.09
	Skeletal	1.09	0.31	0.31	1.71
	Skin	7.79	-	-	7.79
Abdomen	Organs	-	0.62	-	0.62
	Skin	2.34	-	-	2.34
Spine	Organs	6.07	-	-	6.07
	Skeletal	-	0.31	-	0.31
	Skin	0.16	-	-	0.16
Upper Extremity	Organs	0.93	-	-	0.93
	Skeletal	2.34	1.25	0.47	4.05
	Skin	23.68	-	-	23.68
Lower Extremity	Organs	0.47	-	-	0.47
	Skeletal	1.56	0.93	0.31	2.80
	Skin	15.26	-	-	15.26
Total		91.30	4.80	3.60	100.00

* Loss of consciousness.
**Injuries of unknown severity account for .31% of total.

Driver Injury Experience

The belted drivers ranged in age from 16 to 84 years. Approximately two thirds of the belted drivers were males. Of the 250 belted male drivers in the sample, 145 (58%) sustained some degree of injury. The injury rate among belted female drivers was significantly higher. Of the 130 belted female drivers, 101 (78%) were injured.

As would be predicted by the high representation of low severity collisions in the present sample, most of the injured belted drivers sustained minor injuries (AIS 1) as measured by the Abbreviated Injury Scale (AIS) [11]. The percentage distribution of individual injuries by body region and severity is shown in Table 3. The most frequently injured body regions were the upper extremities, followed by the face, and the lower extremities. Only 8.4% of the individual injuries recorded among belted drivers were rated as AIS ≥2. The latter injuries typically consisted of head injuries involving a loss of consciousness (LOC), followed by fractures of upper extremities and fractures of lower extremities.

The findings noted above show good agreement with those of other recently published studies on the field performance of air bag systems. The incidence of injury to the upper extremities of occupants in air bag crashes was examined by Huelke [6] based on collisions investigated primarily by the University of Michigan Transportation Research Institute (UMTRI). The UMTRI data also showed upper extremity injuries are a common consequence of air bag deployments. As in the present study, the injuries documented in the UMTRI study ranged from minor (AIS 1) contusions, abrasions and burns, to AIS 3 fractures of the forearm. An elevated rate of facial abrasion among belted drivers in air bag deployments was also observed in a recent University of Virginia study [12]. The study showed that drivers protected by both the seat belt and air bag experienced a higher incidence of facial abrasions than occupants protected only by seat belts.

Table 4.
Distribution of Belted Drivers with Air Bag Deployment as a Function of MAIS and EBS

EBS (km/h)	No Injury %	MAIS AIS 1 %	MAIS AIS >=2 %	Total %
0 - 19	56.3	44.4	26.9	47.4
Over 19	43.8	55.6	73.1	52.7
Total	100.0	100.0	100.0	100.0

In addition to the relatively low representation of head injuries in the present sample, another striking feature of the injury distribution in the present sample is the low incidence of facial injury at the AIS 2-3 level. Historically such facial injuries accounted for over 20% of all AIS ≥2 injuries observed among belted drivers [7]. In the present sample, they account for less than 2% of the injuries observed at this severity interval.

A further striking feature is the high representation of AIS ≥2 injuries in very low speed impacts. As can be seen from the findings presented in Table 4, approximately 27% of all MAIS ≥ 2 cases in the present sample were sustained at equivalent barrier speeds under 20 km/h.

Paired Comparisons - A comparison of injury outcomes for all cases where a belted driver protected by an air bag was accompanied by a belted right front passenger not protected by an air bag is presented in Table 5. Initial comparisons of injury outcome were based on whether the occupants escaped injury, were injured or were killed. If both occupants in the vehicle sustained non-fatal injuries the comparisons were based on the maximum AIS value assigned to the injured occupant. A parallel analysis was carried out using field

Table 5.
Injury Outcome (IO) Comparison Driver vs. Front Right Passenger

Driver vs. Right Front Passenger	ACRS Study (%)	PCS Study (%)
Driver IO = RF Pass. IO	63	53
Driver IO > RF Pass. IO	24	19
Driver IO < RF Pass. IO	12	28

accident data compiled as part of Transport Canada's Passenger Car Study. [13] In this case, the pairings comprised belted (without air bags) drivers accompanied by belted right front passengers in single event frontal collisions. Consequently, in these analyses the right front passenger serves as a "control" subject.

From these results it can be seen that in 28% of the pairings contained in the PCS database, the injuries sustained by drivers were less severe than those of the accompanying right front passenger. In the ACRS database, this was found to be the case in only 12% of injury outcome pairings. As noted in earlier publications, this negative outcome reflects the fact that at lower collision severities driver injury risk is increased by deployment of the air bag [14,15]

Canadian Case Studies

An appreciation of the advantages afforded by air bag systems, as currently designed, as well as their limitations and drawbacks can be gained from individual case studies.

While manual seat belts alone provide good protection against head and facial injury in minor to moderate collision severities, the risk of head contact increases greatly when delta-v exceeds of 40 km/h. Under such circumstances, the intervention of a compliant structure such as an air bag can greatly reduce the magnitude of the forces applied to the head. As the loads are also more distributed, the risk of facial fracture is reduced greatly. These benefits are illustrated in the following cases:

ACRS-1113: The driver of a 1989 Oldsmobile Cutlass Ciera lost directional control and his vehicle spun across the roadway into the path of the on-coming case vehicle, a 1994 Pontiac Grand Am. The front of the Grand Am (12FDEW2) struck the right side of the Cutlass. The EBS for the Grand Am was estimated at 45 km/h. The fully restrained driver (male/39 years) braced his hands on the steering wheel prior to the impact. He sustained a fracture to the proximal fifth metacarpal of the right hand (752002.2,1), and minor contusions to the face and chest.

ACRS-1127: The right-front end of a 1993 Ford Taurus (12FZAA6) impacted the left-rear corner of a 1989 International tanker truck (06BLMW1). A maximum crush of 146 cm was measured at the right front bumper of the Taurus, while the frame of the truck was severely distorted. The driver of the Taurus was a 39 year old, female. She was fully restrained and her air bag deployed. Rescue personnel took over an hour to extricate the driver from the wreckage. She suffered fractures to the right tibia (852200.2,1) and right foot (853406.2,1) She also sustained multiple minor contusions, lacerations, and abrasions to her face, chest, arms, and legs.

Each deployment of an air bag exposes the occupant to risk of injury from the air bag itself. The risk of bag-induced injury is influenced greatly by the proximity of the occupant to the air bag module at time of deployment. In low speed collisions the severity of bag-induced injuries can greatly exceed the severity of any injuries which would have been sustained in the absence of deployment as illustrated by the following case studies:

ACR2-1102: The driver of a 1995 Oldsmobile 98 failed to negotiate a left turn, ran off the right side of the roadway and struck a multi-stemmed tree. Only minor damage resulted; the maximum crush was 21 cm, measured at the bumper (12FLEN1). The 75 year old, male driver and a 78 year old, female, right-front passenger were fully restrained, and both front air bags deployed. Both front seats were adjusted forward of the mid-seating positions. The driver was unconscious on admission to hospital, and suffered from amnesia and confusion when he recovered (MAIS 3). He sustained fractures to the right ulna, the eighth rib on the right side, and the nose, in addition to multiple minor contusions, lacerations, and abrasions. The right-front passenger received a fracture of the distal end of the left ulna, a hemorrhage to the right cornea, and multiple minor contusions, lacerations, and abrasions (MAIS 2).

ACRS-1118: Apparently under the influence use of alcohol and prescription drugs, the operator of a 1993 Plymouth Sundance fell asleep and drove through a red traffic light at low speed. The front of the case vehicle struck the side of a 1989 Ford Thunderbird. The collision produced only minor damage to the front of the Sundance (12FYEW1). The belted driver (female/32 years) had the seat positioned fully forward. The driver was unconscious at the collision scene and, on arrival at hospital, was responsive only to pain (160899.3,0). Contact to the driver's neck by the deploying air bag resulted in an abrasion and swelling which closed off the airway (442699.3,4). At the hospital she was placed on a respirator for three days. She also sustained a chipped upper molar (251404.1,8) and minor contusions and abrasions to the arms and chest.

ASF2-1802: The case vehicle, a 1994 Plymouth Sundance, drifted off the right side of the roadway striking a wooden utility pole. Vehicle damage was minimal and was concentrated on the vehicle's front bumper and grille (12FREN1) with maximum crush of 18 cm. The 58 year old female driver had her seat located fully forward with the seat back upright. The crash occurred directly outside a hospital where she was immediately examined and was pronounced dead. The autopsy indicated minor contusions to the right upper arm, left arm and the left chest. Death resulted from a 0.6 cm tear to the root of the left main pulmonary artery with cardiac tamponade (421006.3,4). The pathologist noted that there was no pre-crash degeneration of the arterial tissue. The driver's blood alcohol content was found to be at the legal limit. This fact, in combination with the shallow angle of departure from the roadway, the lack of evasive action, and the injury pattern to the thorax, leads to the conclusion that the driver had fallen asleep and was slumped directly on top of the air bag module at the time of collision.

US FIELD ACCIDENT DATA

In order to examine in more detail a number of the issues raised by the Canadian accident data, additional analyses were carried out drawing on data compiled under the National Accident Sampling System (NASS) - Crashworthiness Data System (CDS) maintained by the US National Highway Traffic Safety Administration (NHTSA). The NASS-CDS database represents a probability-based sample of police-reported light vehicle accidents investigated at selected sites across the US. Weighting factors provided as part of the database permit nationally representative estimates to be made based on criterion accidents. The NASS-CDS database contains only accidents involving at least one light vehicle that was towed away from the scene. Complete vehicle, occupant, and injury data are provided only for towed CDS-applicable vehicles.

In order to compare the performance of manual three-point seat belts and air bags, an injury risk/harm model was developed as a basis for analyzing occupant injuries and the following five performance indicators were defined:

- AIS ≥1, including injuries of unknown severity;
- AIS ≥2, excluding injuries of unknown severity;
- AIS ≥3, excluding injuries of unknown severity;
- mean number of injuries per involved occupant, including uninjured occupants; and
- mean level of "harm" per involved occupant, including uninjured occupants.

"Harm" values as used in the present paper are expressed in terms of normalized "equivalent fatality units" which were derived by dividing the estimated dollar cost of each individual injury by the estimated dollar cost of an individual AIS 6 (Maximum) injury. Harm levels were calculated using dollar values assigned to each individual injury as a function of body region and injury severity. The dollar values used and the

Table 6.
Composition of US NASS (1988-94) Sample by Delta-V

Delta-V	Unbelted, No SRS Fitted	Unbelted, SRS Fitted	Belted, No SRS Fitted	Belted, SRS Fitted
00-15 km/h	14.4%	6.9%	27.2%	31.1%
16-23 km/h	39.8%	32.8%	41.3%	45.9%
24-31 km/h	21.3%	34.0%	21.3%	15.9%
32-39 km/h	14.9%	16.3%	6.8%	4.6%
40+ km/h	9.5%	10.0%	3.4%	2.5%
Total	100.0%	100.0%	100.0%	100.0%

corresponding calculated equivalent fatality units are presented in Tables A.1 and A.2 of Appendix A. The total level of harm assigned to each injured occupant was calculated by summing the equivalent fatality units assigned to each individual injury, including injuries of unknown severity which were assigned a harm value corresponding to the lowest AIS 1 body region injury ($3/$644 = .0047 equivalent fatality units). Note that derived harm values are intended to denote a measure of bodily insult which takes into account the number of individual injuries sustained and their severity. It is not intended to denote an index of net monetary loss.

The initial series of analyses upon which the present paper is based examined the injury experience of drivers of passenger cars fitted with conventional body-mounted three-point seat belt assemblies, with or without an SRS. In the case of vehicles not equipped with air bags, only 1982 to 1986 vehicle models are considered here. This was done to exclude all automatic seat belt systems and to minimize distortions in the vehicle size and class composition of the "baseline" manual-three-point-seat-belt-only fleet. Starting with the 1987 model year, the number of vehicle models in the US that were fitted with automatic restraint systems increased steadily in response to the phase-in requirements of FMVSS No. 208.

Additional criteria for inclusion in the subset of cases analyzed included that vehicles had been inspected by a NASS investigator and, when injuries were sustained, that at least one injury was rated as other than unknown. For vehicles not fitted with an SRS, it was also necessary to know whether the driver had used the seat belt. For vehicles equipped with air bags, information on seat belt use and deployment of the SRS had to be recorded.

In order to generate a US sample of cases with collision characteristics similar to those in the Canadian ACRS sample, only frontal type crashes were included in the analysis. Case selection was based on the CDC code assigned to describe primary damage to the case vehicle. All cases where the general area of damage involved the front of the vehicle (GAD1=F) were included. All side impacts (GAD1= R or L) were included if the direction of force was between 11 o'clock and 1 o'clock. Side impacts where the direction of force was either from the 10 o'clock direction or the 2 o'clock direction were also included if the impacted area was ahead of the passenger compartment (SHL1 = F).

The sample assembled from NASS for calendar years 1988 to 1994 comprised a total of 6,354 drivers (representing 2,451,060 drivers, when weighted) with the following restraint use and fitment characteristics :

- Unbelted, No SRS fitted 2,260 (610,302)
- Belted, No SRS fitted 3,173 (1,521,829)
- Unbelted, SRS fitted 249 (52,723)
- Belted, SRS fitted 666 (266,206)

A description of the sample by delta-v (where available) is provided in Table 6. From the data presented it can be seen that marked differences exist between the collision experience of unbelted and belted drivers in terms of their representation at different collision severity intervals. The former can be seen quite clearly to be over-represented in more severe collisions. These differences preclude the derivation of reliable estimates of injury risk reduction or harm reduction as a function of belt use, controlling for the fitment of the air bag, based on the overall injury and harm rates presented in Table 7. Such estimates could only be derived by reweighting the data to take into account collision severity differences. By way of example, the unadjusted data presented in Table 7 would yield a seat belt use alone effectiveness against AIS ≥2 injury of 64%. When the injury rate for belted occupants was recomputed using the combined delta-v distribution of all belted and belted drivers, the severity-adjusted rate was 18.2%. When the

Table 7.
Overall Injury Risk and Harm Rates as a Function of Use of Seat Belt Use and Fitment of SRS System
(Unadjusted for Collision Severity)

Performance Measure	Unbelted, No SRS Fitted	Unbelted, SRS Fitted	Belted, No SRS Fitted	Belted, SRS Fitted
Injury Probability AIS >= 1	73.5%	82.3%	44.5%	48.0%
Injury Probability AIS >=2	19.4%	22.3%	7.1%	6.9%
Injury Probability AIS >=3	4.8%	9.2%	1.4%	1.8%
Mean Number of Injuries	2.880	2.816	1.290	1.421
Mean Level of Harm	0.0405	0.0432	0.0133	0.0130

injury rate for unbelted occupants was recomputed using the combined delta-v distribution of all belted and belted drivers, the severity-adjusted rate was 26.9%. The severity-adjusted level of risk reduction afforded by the seat belt alone against AIS >2 injury translates to approximately 45%, a value which is consistent with published seat belt effectiveness estimates [16]

On the other hand, reasonable estimates of injury risk reduction and harm reduction as a function of the fitment of an air bag system, controlling for belt use, can be obtained directly from the data presented in Table 7. The two belted populations (with and without SRS fitment) show similar collision severity distributions, as do the two unbelted populations. Although the purpose of the present analysis is to focus on the injury experience of belted drivers, from Table 7 it can be observed that unbelted drivers in the non-SRS sample typically showed lower rates of injury than unbelted drivers in the SRS sample. The mean harm rate for the non-SRS sample was also lower (.0405 vs. .0432). The results for belted drivers were more mixed. Belted drivers in the SRS sample showed a higher overall injury rate, a higher AIS ≥3 injury rate, and a higher mean number of injuries than belted drivers in the non-SRS sample. Belted drivers in the SRS sample showed a lower AIS ≥2 injury rate and a marginally lower overall mean level of harm. Note that all of the rates in Table 7 are calculated independently of whether the air bag deployed, to avoid the introduction of a collision severity bias in the comparisons.

A more detailed breakdown of the harm distribution among belted drivers by body region grouping is provided in Table 8. While the mean level of harm to the head in the SRS sample can be seen to be substantially lower than that in the non-SRS sample these head benefits are offset by an increased level of overall harm to the face and upper extremities. This trade-off is consistent with the Canadian findings described in previous sections of this paper.

The mean harm levels for the two belted driver populations as a function of collision severity, represented by delta-v, are summarized in Table 9. Here we can see that belted drivers in the SRS sample showed a greatly reduced mean harm level at collision severities in excess of 39 km/h, but showed higher overall mean harm rates in the under 24 km/h and the 24-39 km/h delta-v intervals. It is apparent that the benefits achieved at higher collision severities by the intervention of the air bag are being negated by deployments in low and moderate speed collisions. This trade-off is again consistent with the Canadian findings.

Table 8.
Mean level of Harm by Body Region Grouping

Body Region Grouping	Mean Level of Harm		% Change in Mean Harm Given SRS Fitment
	Belted, No SRS Fitted	Belted, SRS Fitted	
Head	0.0026	0.0009	-65.4%
Face, Upper Extremities	0.0038	0.0065	71.1%
Other	0.0068	0.0056	-17.6%
All	0.0133	0.0130	-2.3%

Table 9.
Mean level of Harm as a Function of Delta-V

Collision Severity Grouping	Mean Level of Harm		% Change in Mean Harm Given SRS Fitment
	Belted, No SRS Fitted	Belted, SRS Fitted	
Under 24 km/h	0.0063	0.0112	77.8%
24-39 km/h	0.0215	0.0270	25.6%
40+ km/h	0.1287	0.0944	-26.7%

How the injury experience of belted drivers varies as a function of gender was also examined. These results are summarized in Tables 10 and 11. As would be predicted on the basis of the Canadian experience, belted females in the SRS sample showed higher injury and harm rates than belted males in the SRS sample. They also showed consistently higher injury and harm rates than belted females in the non-SRS sample. The differences are most pronounced with respect to the incidence of AIS ≥ 3 injuries, which in turn, were found to be linked to the elevated AIS 3 injury rate to the upper extremities among belted females in the SRS sample in low severity crashes in the SRS sample. A detailed breakdown of the NASS data on the incidence of AIS 3 upper extremity injury among belted females is presented in Table 12.

The NASS subset of frontal cases examined in the present paper contains a total of 13 belted females in the SRS sample who sustained AIS 3 upper extremity fractures. All sustained their injuries in collisions which produced deployment of the air bag. Six belted females, out of an unweighted sample of 74, sustained their injuries at collision severities under 24 km/h. By way of comparison, none of the 443 belted females in the

Table 10.
Overall Injury Risk and Harm Rates among Belted Male Drivers as a Function of SRS Fitment

Performance Measure	Belted, No SRS	Belted, SRS Fitted	% Change Given SRS Fitted
Injury Probability AIS >= 1	37.8%	39.9%	5.5%
Injury Probability AIS >=2	6.5%	5.7%	-12.9%
Injury Probability AIS >=3	1.1%	1.3%	10.4%
Mean Number of Injuries	1.038	0.957	-7.8%
Mean Level of Harm	0.0112	0.0099	-11.6%

Table 11.
Overall Injury Risk and Harm Rates among Belted Female Drivers as a Function of SRS Fitment

Performance Measure	Belted, No SRS	Belted, SRS Fitted	% Change Given SRS Fitted
Injury Probability AIS >= 1	50.7%	57.4%	13.2%
Injury Probability AIS >=2	7.6%	8.2%	8.5%
Injury Probability AIS >=3	1.7%	2.5%	41.1%
Mean Number of Injuries	1.521	1.937	27.4%
Mean Level of Harm	0.0152	0.0166	9.2%

Table 12.
AIS 3 Upper Extremity Injury Experience of
Belted Female Drivers as a Function of SRS Fitment and SRS Deployment

Collision Severity	Restraint Configuration	NASS Weighted % Injured AIS =3	NASS Unweighted % Injured AIS =3	Unweighted Populations
< 24 km/h	Belted, No SRS	0.0 %	0.0 %	0/443
	Belted, SRS, No Deployment	0.0 %	0.0 %	0/33
	Belted, SRS, Deployment	8.8 %	8.1 %	6/74
>= 24 km/h	Belted, No SRS	0.7%	2.2 %	11/491
	Belted, SRS, No Deployment	0.0 %	0.0 %	0/8
	Belted, SRS, Deployment	4.4 %	4.0 %	3/75
Unknown	Belted, No SRS	0.3 %	1.1 %	7/640
	Belted, SRS, No Deployment	0.0 %	0.0 %	0/46
	Belted, SRS, Deployment	0.8 %	5.0 %	4/80

unweighted non-SRS sample sustained AIS 3 injury to this body-region grouping at collision severities under 24 km/h. Another three belted females in the SRS sample sustained AIS 3 upper extremity injuries at collision severities at or above 24 km/h. The remaining four cases in the SRS sample sustained injury in collisions without a coded delta-v. Examination of the CDC coded for the damaged vehicle indicates at least 3, if not all 4, of the latter victims sustained their injuries in low speed collisions.

From the data presented it can be seen that the elevated upper extremity injury rate in the SRS sample among belted females is not a product of distortions associated with any NASS weight factor assigned to these cases. The weighted injury frequencies for the two delta-v intervals, however, are inflated by virtue of the exclusion of cases with unknown delta-v estimates. The latter subset yields a weighted injury frequency, given deployment of the SRS, well below those yielded for the subset of cases where delta-v was reported.

It is also interesting to note that, given SRS deployment, the AIS 3 injury rate in low severity collisions is twice that observed in higher severity collisions. This is not unexpected since maximum opportunity for direct injury from the module cover itself as well as from arm flailing would be expected to occur during turning manoeuvres when the forearm is centred over the module. This is far more likely to occur at lower driving speeds. To further test this hypothesis, the analysis was repeated for all AIS ≥2 upper extremity injuries. These results are presented in Table 13.

Here again we see the rate of upper extremity injury at the AIS 2 or greater level among females in lower severity collisions, given deployment of the SRS, is twice

Table 13.
AIS >=2 Upper Extremity Injury Experience of
Belted Female Drivers as a Function of SRS Fitment and SRS Deployment

Collision Severity	Restraint Configuration	NASS Weighted % Injured AIS ≥2	NASS Unweighted % Injured AIS ≥2	Unweighted Populations
< 24 km/h	Belted, No SRS	1.3 %	2.3 %	10/443
	Belted, SRS, No Deployment	0.0 %	0.0 %	0/33
	Belted, SRS, Deployment	20.1 %	16.2 %	12/74
>= 24 km/h	Belted, No SRS	4.8 %	10.8 %	53/491
	Belted, SRS, No Deployment	0.0 %	0.0 %	0/8
	Belted, SRS, Deployment	9.9 %	12.0 %	9/75
Unknown	Belted, No SRS	1.1 %	4.2 %	27/640
	Belted, SRS, No Deployment	1.4 %	2.2 %	1/46
	Belted, SRS, Deployment	2.6 %	11.3 %	9/80

Table 14.
Injury Experience of Belted Drivers as a Function of SRS Fitment and Delta-V

Performance Measure	Delta-V : 0 - 23 km/h		Delta-V : 24 - 31 km/h		Delta-V : 32 - 39 km/h	
	Belt Used, SRS NotFitted	Belt Used, SRS Fitted	Belt Used, SRS NotFitted	Belt Used, SRS Fitted	Belt Used, SRS NotFitted	Belt Used, SRS Fitted
Mean Number of Injuries Per Involved Occupant :	0.7930	1.2760	2.1390	2.1340	2.9640	2.4440
Mean Level of Harm Per Involved Occupant :						
Head	0.0014	0.0001	0.0020	0.0008	0.0042	0.0000
Face, Upper Ext.	0.0019	0.0074	0.0059	0.0067	0.0106	0.0231
Other	0.0030	0.0037	0.0105	0.0126	0.0168	0.0274
Total	0.0063	0.0112	0.0183	0.0202	0.0315	0.0506
AIS >= 3 Count Occupant Level (Unweighted) :						
Head	5	0	5	0	2	0
Face, Upper Ext.	0	6	4	1	3	4
Other	7	2	18	1	21	5
Total Sample	887	197	523	85	225	40

that observed at higher collision severities. While the weighted frequencies may be inflated by the NASS weighting factors associated with these cases and by the exclusion of cases where delta-v is unreported, the increased risk of upper extremity injury associated with deployment of the air bag is evident.

The collective findings presented above show close agreement with a recent US study which examined the relationship between injury risk and deployment frequency using both insurance claim frequency data and NASS data [17]. The study compared the injury experience of two popular midsize car models, one with a high frequency of air bag deployment (50 per 100 frontal crashes) and one with a low frequency of deployment (20 per 100). The study observed the former produced a 50% higher injury rate in low speed collisions. The body regions most likely to be adversely affected were the face and the upper extremities. The authors also observed that females showed a higher rate of bag induced injury in low speed collisions than males, particularly with respect to the upper extremities.

Deployment Threshold

Further analyses were performed to examine what would constitute an appropriate threshold for air bag systems in a jurisdiction comprised largely of belted drivers, such as Canada. From the analysis already presented, raising the threshold to 24 km/h can be seen to afford a number of advantages. This would reduce the number of air bag deployments currently experienced by some 75 %. In addition to the monetary savings in reduced vehicle repair costs, this would achieve a significant harm reductions, particularly among female drivers. The number of catastrophic injuries associated with out-of-position occupants could also be expected to be reduced by 75 percent.

The data previously presented in Table 9 suggest that additional harm benefits could be achieved by increasing the air bag threshold beyond 24 km/h. To explore this possibility, the injury experience of belted drivers in the non-SRS and SRS samples was quantified for three delta-v intervals under 40 km/h. These results are summarized in Table 14.

From the findings presented, it can be seen that the injury rates for the two driver populations, both in terms of the mean number of injuries sustained and the mean level of harm, initially converge over the delta-v interval from 24-31 km/h, but again diverge over the interval from 32-39 km/h. The divergence in the mean harm rate can be seen to be associated with the elevated incidence of AIS 3 upper extremity injury in the SRS sample. Conversely, the data also suggest that the SRS benefits over the 24-31 km/h delta-v interval are being negatively influenced by the NASS weighting factors since the SRS sample can be seen to be outperforming the non-SRS sample in terms of raw AIS ≥3 injury frequencies.

The injury experience of the individual belted drivers in the non-SRS sample represented in the 24-31 km/h interval is informative. Included in this subset of cases were a total of 17 drivers who sustained injury at the AIS ≥3 level at delta-v severities between 26 and 29 km/h. Nine of these victims were over the age of 60 years (they ranged from 63 to 89 years in age). Seven of the nine victims sustained either head or chest injuries from contact with the steering wheel or chest injuries attributed to the seat belt. Further investigation revealed that a total of 39 belted drivers, 60 years of age or older, are represented in the non-SRS NASS sample for the delta-v interval of 26-29 km/h. Of these drivers, 9 (23%) sustained no injury, while the injuries to another 15 were confined to the AIS 1 level. Thus, while it can be seen that the majority of elderly belted drivers represented in this collision severity interval escaped with little or no injury, it is possible that the intervention of an air bag could have reduced the severity of the injuries sustained of 7 (12%) of the 39 individuals in this subset of collisions.

Only 4 belted drivers 60 years of age or older are represented in the SRS sample in the delta-v interval of 26-29 km/h. While all four individuals were injured, none suffered injuries in excess of AIS 2 and all bag-related injuries were confined to AIS 1. Of the 32 belted drivers 60 years or older in the SRS sample in <u>all collisions</u> under 30 km/h delta-v with deployment, only 4 (12.5%) escaped with no injury while 3 (9%) sustained bag-induced injuries at the AIS 3 level (to the upper extremities).

From the above injury frequencies, 26 km/h represents the point at which, in exchange for an increase in overall injury probability (AIS ≥1) and an increased likelihood of sustaining a non life-threatening AIS 3 upper extremity injury, deployment of the air bag <u>may offer</u> a reduced risk of sustaining a possible life-threatening AIS ≥3 injury among the segment of the population at highest injury risk in a moderate speed collision. This trade-off would be more favourable than indicated above, if the risk of an AIS 3 upper extremity injury in moderate speed collisions is actually less than that in low speed collisions, as suggested by the earlier analysis. This trade-off would certainly be rendered more favourable if air bag designs were rendered less aggressive.

DISCUSSION

The analyses of both Canadian and US field accident data provide considerable insights into the benefits afforded by supplementary air bag systems as well as their present limitations and drawbacks. On the benefit side, the ability of air bags systems to substantially reduce the risk of serious head injuries and facial fractures is reflected very clearly in both Canadian and US collision data. However, both sets of data indicate that these gains are being accompanied by a substantial increase in the risk of injury in minor and more moderate severity collisions. While the majority of bag-related injuries take the form of AIS 1 facial and AIS 1-3 upper extremity injuries, they can include AIS ≥3 injuries to other body regions if the occupant is close to the deploying air bag. The NASS data show a much higher rate of bag-induced upper extremity injury at the AIS 3 level than would be predicted by the Canadian data. Both Canadian and US data agree that female drivers are at higher risk of sustaining bag-induced injuries.

Historically, if a safety device was found to be effective in reducing the risk of fatal injury, it was also found to be effective in preventing serious non-fatal injuries. Owing to the high incidence of bag-induced injuries, particularly to the upper extremities, the present findings suggest that this is not likely to prove true in the case of air bags, given current design and deployment practices.

Since the benefits and drawbacks of air bag systems as currently designed can be seen to be drawn largely from mutually exclusive portions of the collision severity spectrum, a significant improvement in the overall level of protection afforded belted occupants by air bags could be achieved by increasing the deployment threshold. In jurisdictions such as Canada which have achieved high levels of seat belt use, the vast majority of air bag deployments in low speed collisions serve no useful purpose: injury outcome is either unchanged or it is adversely affected.

Although air bags are marketed as supplementary equipment to the seat belt, the current generation of air bag systems are a product of the US automatic restraint ruling and reflect design practices dictated by the requirements imposed by the US unbelted test. Even so, air bag systems have consistently demonstrated exceptionally good performance in belted tests performed by Transport Canada. Nor is there any evidence to suggest that the protection interests of "average" belted male drivers is not being served properly by current air bag designs in high severity crashes. The same cannot be said for other segments of the driving population. Moreover, current regulatory practices fail to ensure that optimum benefits are achieved over the range of collision severities that are represented in the field.

As reflected by the injury experience of females in both the Canadian and US collision data, the segment of the driving population who place their seats ahead of the mid seating position cannot be overlooked in regulations governing occupant protection systems. A number of possible regulatory options to remedy this situation are presently being evaluated by Transport Canada. One option is to add a second belted 48 km/h test using 5th percentile female Hybrid III dummies with the seats set in the full forward position.

Another option is to introduce a low speed offset frontal crash test, again using 5th percentile dummies with the seats set in the full forward position. The test speed could be set to correspond closely to what, in a general sense, would constitute an appropriate speed to deploy an air bag deployment for a belted occupant. Vehicle manufacturers would then have the option of satisfying the associated performance requirements either through deployment of the air bag system or by other means as deemed appropriate. Such a test condition would encourage consideration of countermeasures such as belt-pretensioners, compliant steering wheel rims, padded hubs and padded A-pillars to provide low speed protection. The use of air bags could then be limited to higher severity collisions where the benefits in terms of reduced head injury and facial fracture potential can more reasonably be expected to outweigh the risk of any bag-induced injury.

Neither of the above options would necessarily preclude the fitment of common air bag systems in Canada and the US if the development of "smart" air bag systems, which vary deployment threshold and inflation characteristics as a function of belt status and seat position, proves successful [18].

ACKNOWLEDGMENTS AND DISCLAIMER

The authors would like to acknowledge the contributions of the collision investigation teams at the Technical University of Nova Scotia, University of New Brunswick, École Polytechnique de Montréal, Ryerson Polytechnic University, University of Western Ontario, University of Saskatchewan, University of Calgary, and the University of British Columbia in providing the field data for this study. The assistance afforded by Floyd Dempsey in the creation and maintenance of the collision databases is similarly acknowledged.

The conclusions reached, and opinions expressed, in this paper are solely the responsibility of the authors. Unless otherwise stated, they do not necessarily represent the official policy of Transport Canada.

APPENDIX A

Table A.1
Average Cost of Injury as a Function of Body Region and Injury Severity Level
(1988 $ US x1000 Dollars)

Body Region	AIS Level						
	AIS 1	AIS 2	AIS 3	AIS 4	AIS 5	AIS 6	UNK
External	$ 3	$ 16	$ 45	$ 73	$ 106	$ 644	$ 3
Head	$ 4	$ 19	$ 78	$ 180	$ 636	$ 644	$ 3
Face	$ 4	$ 19	$ 78	$ 103	$ 211	$ 644	$ 3
Neck	$ 4	$ 19	$ 78	$ 103	$ 211	$ 644	$ 3
Chest	$ 3	$ 16	$ 45	$ 73	$ 106	$ 644	$ 3
Abdomen-Pelvis	$ 3	$ 16	$ 45	$ 73	$ 106	$ 644	$ 3
Spine	$ 3	$ 16	$ 105	$ 905	$ 1,082	$ 644	$ 3
Upper Extremity	$ 4	$ 28	$ 66				$ 3
Lower Extremity	$ 3	$ 28	$ 84	$ 124	$ 211		$ 3

Table A.2
Converted Average Harm Value as a Function of Body Region and Injury Severity Level
(Equivalent Fatality Units)

Body Region	AIS Level						
	AIS 1	AIS 2	AIS 3	AIS 4	AIS 5	AIS 6	UNK
External	0.0047	0.0248	0.0699	0.1134	0.1646	1.0000	0.0047
Head	0.0062	0.0295	0.1211	0.2795	0.9876	1.0000	0.0047
Face	0.0062	0.0295	0.1211	0.1599	0.3276	1.0000	0.0047
Neck	0.0062	0.0295	0.1211	0.1599	0.3276	1.0000	0.0047
Chest	0.0047	0.0248	0.0699	0.1134	0.1646	1.0000	0.0047
Abdomen-Pelvis	0.0047	0.0248	0.0699	0.1134	0.1646	1.0000	0.0047
Spine	0.0047	0.0248	0.1630	1.4053	1.6801	1.0000	0.0047
Upper Extremity	0.0062	0.0435	0.1025				0.0047
Lower Extremity	0.0047	0.0435	0.1304	0.1925	0.3276		0.0047

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VEHICLE OCCUPANT RESTRAINT SYSTEM PERFORMANCE

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Paper Number 96-S1-O-06

ABSTRACT:

The early 1990s was a period which saw the rapid introduction of improved occupant restraint systems into Australian vehicles. However, even in 1995, by which date nearly all US passenger vehicles were equipped with single or twin airbags, most Australian vehicles only offered a single airbag, and in most cases this was optional equipment. A few models offered twin airbags, most also optional equipment.

There has therefore been a limited opportunity to examine the difference in results between otherwise identical vehicle models with and without airbags in NCAP tests.

It has also become evident that crash dummy head to knee impacts are much more common in the Australian NCAP than in the US NCAP. This is mainly due to the difference in the rate of airbag fitment noted above, and again there is a valuable opportunity to gain useful knowledge about occupant restraint systems by examining the circumstances of the head to knee impacts experienced in the Australian NCAP.

This paper examines the difference in NCAP injury scores between otherwise identical vehicle models with and without airbags, and analyses the reasons for dummy head to knee contacts in NCAP full frontal tests. From these analyses some suggestions are made to improve occupant restraint performance.

INTRODUCTION

The introduction of airbags into Australian vehicles has been much slower than in the US. By 1995 virtually all passenger vehicles and the majority of light trucks in the US were fitted with airbags, often twin airbags systems.

In 1996 in Australia there were few mass market cars with airbags as standard equipment, and there are many models on the market which either have no airbag available at all, or only a driver airbag option.

The Australian New Car Assessment Program (NCAP) was established to provide information to new vehicles buyers on the relative occupant protection afforded by new vehicles in frontal crashes. The program follows the crash test procedures laid down for the 56 km/h full frontal test by the US National Highway Traffic Safety Administration (NHTSA) and the European Experimental Vehicle Committee (EEVC) for the 64 (formerly 60) km/h offset test.

Australian NCAP selects vehicle models to test based on the model's market share and the most popular variants within that model. The option is sometimes taken to test a "benchmark" vehicle which is accepted as performing well in the test to explore levels of best practice in the industry (eg Volvo 940).

COMPARISONS OF MODELS WITH AND WITHOUT AIRBAGS

Partly due to the above guidelines only the Subaru Liberty (Legacy in other markets) and the Toyota Camry models have been tested both with and without airbags. The Liberty has a twin airbag system as an option while the Camry has only a driver airbag option in Australia, although a passenger airbag is fitted to the US model.

The test results for these two vehicles are shown in Table 1 in the Appendix. It can be seen that the HICs from the airbag positions in full frontal tests were consistently lower than those from the non-airbag equivalents. In the offset test, which concentrates the vehicle structure deformation on the driver's side of the vehicle, the driver HIC difference was 560. The passenger HIC was higher in the airbag-equipped model than the non-airbag version by 150 but the passenger HIC figures in both variants in the offset test were well below the level of concern. Other measurements show no consistent trends.

While the difference in HIC results is expected, we might also have expected some consistent reduction in chest compression and/or chest "g", due to the support given by the airbag to the upper body as well as the head. In the full frontal tests this does not appear to be the case.

Some effect on chest injury parameters might be expected particularly in the case of the Subaru Liberty (Legacy) which is sold in several international markets. This model (and several others from a range of manufacturers) has been shown in NRMA low speed impact damage tests to have an airbag triggering impact speed below 16 km/h. This is a common speed selected for the US market where manufacturers try to protect occupants not wearing seat belts by triggering airbags at comparatively low impact velocities. In Australia, where the seat belt wearing rate averages above 95%, this calibration simply serves to increase repair costs.

Australian-manufactured vehicles have their airbag triggering point set at an equivalent barrier impact speed of about 25 km/h, which is approximately where belted occupants begin to gain some benefit from airbag deployment.

Reduced driver chest compression of 10mm shown by the airbag model in the Subaru offset test was 24% lower the non-airbag specification which supports a hypothesis that in more serious crashes with greater structural deformation the airbag supplies some support to the thoracic region in addition to that provided by the seat belt.

While the comparison data are limited, the difference between the chest injury parameters in full frontal and offset tests suggests that modification to airbag deployment characteristics now being researched for "smart" airbags could also be effective in reducing occupant thoracic injury in full frontal-type crashes. Wider use of seat belt pre-tensioners would also be likely to help in better controlling upper body movement.

ANALYSIS OF HEAD TO KNEE IMPACTS

A summary of NCAP test results from Australian vehicles showing head to knee impacts is shown in the Appendix in Table 2. For comparison the results from a range of vehicles not showing head to knee impacts is included as Table 3 also in the Appendix.

While the Hybrid 3 dummy knee design is not fully biofidelic so the HIC figure cannot be relied upon, the HIC result has been included for completeness.

Referring to Table 2 the impact of a safety upgrade as a running change can be seen in the Toyota Camry's performance. Its earlier specification allowed 98 mm of passenger seat belt reel out compared with the later model which allowed only 43 mm, with benefits in reduced head and upper body excursion. While the HIC figures cannot be totally relied on as noted above, they indicate that reducing thoracic and head movement can reduce injuries.

A summary of the results is included here.

Vehicle Make & Model	*	Max seat belt reel out - mm	HIC	Chest compress-ion -mm	Chest g m sec^{-2}	Left leg kN	Right leg kN
Average for head impact vehicles	P	82	1745	45	48	2.3	2.3
Average for non head to knee impact	P	77	1100	40	51	1.8	2.2
Δ head impact - non head impact	P	5	646	4	-4	0.5	0.1

* P - Passenger

Comparing the average passenger test results between the vehicles which had a significant head strike and a range of vehicles which did not have a significant head strike shows a major difference only in HIC, of 646. However, HIC is not reliable for head to knee impacts. There was only a minor difference in average belt reel out which shows that reel out alone is not the determinant of a low injury score.

This is further demonstrated in the case of the driver airbag and seat belt pre-tensioner-equipped Volvo 940 in which both passenger and driver belt reel outs were among the highest of all vehicles, yet the full frontal test injury figures were the best yet recorded in the Australian NCAP program. In this case the comparatively large reel out was balanced by the slack taken out of the belt system by the pretensioners.

Examining the results from the models with the lowest injury indices, which are listed in Table 4 we can see that they all have belt reel out towards the lower end of the scale except the Volvo 940 which, as noted above, mitigates the high reel out with pretensioners. These results emphasise the need to prevent the occupants moving forwards excessively and striking the interior of the vehicle. This is increasingly being achieved by the use of webbing clamps and/or pretensioners.

CONCLUSIONS

- The inclusion of airbags significantly reduces head accelerations but have little effect on the dummy chest parameters measured in full frontal crash testing.
- The low equivalent barrier test speed of less than 16 km/h commonly adopted as a triggering point for US market airbag installations is lower than necessary to protect occupants, and significantly increases repair costs in low speed crashes.
- The incidence and likelihood of significant passenger head to knee impacts cannot be determined simply by seat belt reel out. Comparing US and Australian NCAP experience shows that fitting passenger airbags almost completely eliminates head to knee impacts.
- Recent model vehicles scoring low injury risk figures had low belt reel out figures, emphasising the need to reduce the movement of the head and upper body to reduce injuries in frontal crashes.

APPENDIX

Table 1.
Crash Test Results - Australian NCAP Vehicles

Full Frontal Test 56.3 km h^{-1}

Vehicle Make & Model	*	HIC	Chest compress-ion - mm	Chest g, m sec^{-2}	Left leg kN	Right leg kN
Subaru Liberty current model, non airbag	D	1040	48	51	3.0	2.8
	P	1110	38	49	0.7	1.6
Subaru Liberty current model, twin airbag	D	710	45	51	1.2	3.2
	P	920	39	53	1.3	1.4
Δ non airbag-airbag (+ve means improvement with airbag)	D	330	3	0	1.8	-0.4
	P	190	-1	-4	-0.4	0.2
Toyota Camry current model with belt grabbers, non airbag	D	1360	45	44	0.5	1.5
	P	@	54	43	0.7	1.4
Toyota Camry before safety upgrade, driver airbag	D	1040	37	61	0.6	1.9
	P	@	41	39	#	1.4
Δ non airbag-airbag (+ve means improvement with airbag)	D	320	8	7	-0.1	-0.4
	P	@	13	4	-	0.0
Offset crash test results						
Subaru Liberty current model, non airbag	D	960	42	41	3.4	2.9
	P	300	35	41	n/i	n/i
Subaru Liberty current model, twin airbag	D	400	32	42	1.9	3.3
	P	450	34	40	n/i	n/i
Δ non airbag-airbag (+ve means improvement with airbag)	D	560	10	0	1.5	-0.4
	P	-150	1	1	-	-

* D - Driver, P - Passenger
\# no data, equipment failure
n/i not instrumented
@ head-to-knee impact

Table 2.
Vehicles Showing Significant Head to Knee Impact - Australian NCAP Vehicles - Full Frontal Test 56.3 km h^{-1} - Passenger results

Vehicle Make & Model	Max seat belt reel out - mm	HIC	Chest compress-ion -mm	Chest g, m sec^{-2}	Left leg kN	Right leg kN
Nissan Pulsar 91-96	113	1530	43	57	2.0	2.5
Ford Laser current model	34	2625	55	55	4.7	4.7
Toyota Camry before safety upgrade	98	1350	41	39	#	1.4
Toyota Camry current model	43	1070	54	43	0.7	1.4
Toyota Corolla 93-96	95	1220	36	49	3.2	1.0
Hyundai Excel 95	111	2680	42	45	1.1	3.0
Average for head impact vehicles	82	1746	45	48	2.3	2.3

* D - Driver, P - Passenger

\# no data, equipment failure

Table 3.
Vehicles Not Showing Significant Head To Knee Contact - Australian NCAP Vehicles Full Frontal Test 56.3 km h^{-1}

Vehicle Make & Model	*	Max seat belt reel out - mm	HIC	Chest compress-ion -mm	Chest g, m sec^{-2}	Left leg kN	Right leg kN
Subaru Liberty current model non airbag	D	46	1040	48	51	3.0	2.8
	P	46	1110	38	49	0.7	1.6
Subaru Liberty current model twin airbag	D	67	710	45	51	1.2	3.2
	P	53	920	39	53	1.3	1.4
Ford Mondeo current model driver airbag	D	24	870	51	51	2.8	4.6
	P	23	900	50	44	2.9	3.3
GM Commodore 91-93 driver airbag	D	20	1170	41	51	1.2	3.2
	P	19	1110	45	53	1.3	1.4
Ford Falcon current model driver airbag	D	30	910	59	74	#	7.4
	P	33	1280	48	56	6.1	1.7
Volvo 940 92-95 driver airbag	D	135	490	48	50	4.0	2.3
	P	136	600	46	46	1.4	0.4
Mitsubishi Magna current model	D	126	1140	51	60	3.4	3.8
	P	105	1580	45	58	0.9	1.1
Nissan Pintara 89-92	D	101	1750	44	64	1.3	2.4
	P	107	890	40	67	0.8	0.7
Honda Accord 90-93	D	96	1500	51	58	2.8	3.1
	P	98	1330	37	50	2.7	4.4

Table 3 (cont).

Vehicle Make & Model	*	Max seat belt reel out - mm	HIC	Chest compress-ion -mm	Chest g, m sec^{-2}	Left leg kN	Right leg kN
Daihatsu	D	83	1000	50	70	2.1	2.7
Charade 93-94	P	94	1260	41	50	1.3	3.0
Ford Laser	D	104	1900	42	68	8.6	2.8
92-94	P	108	1790	#	70	2.7	3.9
GM Barina	D	71	1000	54	59	3.9	1.6
91-94	P	60	1150	38	47	1.4	2.3
Honda Civic	D	87	1460	52	63	3.0	3.2
GL 91-94	P	92	1100	41	55	1.6	1.5
Hyundai Excel	D	115	1320	36	54	0.5	3.6
90-94	P	138	860	37	44	2.0	1.3
Mazda 121	D	143	1530	34	61	4.7	4.4
current model	P	123	1070	34	47	1.0	2.1
Subaru Impreza	D	36	1110	47	54	0.8	3.7
current model	P	39	890	41	51	0.9	2.5
Daewoo Cielo	D	48	900	56	49	5.0	1.0
current model	P	36	860	34	43	1.4	4.3
Average for non	D	78	1164	47	58	3.0	3.3
head to knee impact	P	77	1100	40	51	1.8	2.2
Δ head impact-non head impact	P	5	646	4	-4	0.4	0.1

* D - Driver, P - Passenger

\# no data, equipment failure

Table 4.
Vehicles Showing Lowest Injury indices in Australian NCAP Testing

Vehicle Make & Model	*	Max seat belt reel out - mm	HIC	Chest compress-ion -mm	Chest g, m sec^{-2}	Left leg kN	Right leg kN
Volvo 940	D	135	490	48	50	4.0	2.3
92-95 driver airbag	P	136	600	46	46	1.4	0.4
Daewoo Cielo	D	48	900	56	49	5.0	1.0
current model	P	36	860	34	43	1.4	4.3
Subaru Liberty	D	46	1040	48	51	3.0	2.8
current model non airbag	P	46	1110	38	49	0.7	1.6
Ford Mondeo	D	24	870	51	51	2.8	4.6
current model driver airbag	P	23	900	50	44	2.9	3.3
GM Commodore	D	20	1170	41	51	1.2	3.2
91-93 driver airbag	P	19	1110	45	53	1.3	1.4

* D - Driver, P - Passenger

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OPTIMIZING SEAT BELT USAGE BY INTERLOCK SYSTEMS

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ABSTRACT

Seat belts are known to be very effective, reducing the risk of injury by approximately 50% when used. Such high effectiveness is, however, based on the fact that all car occupants use the available belts. In several studies it has been shown that, in severe accidents, the seat belt use was less than 50%.

In order to increase the wearing rate more drastic solutions than information, legislation etc. have to be used. A Swedish group, representing government, research, insurance companies, car and restraint systems industry has approached the problem by proposing a smart system that will force car occupants, that normally are unbelted, to use the seat belts by systems that will interfere with the normal use of the car. Different technical approaches, which not in any way will interfere with the normal belt user, will be put forward and evaluated. The problem will also be discussed from a cost-effectiveness point of view and the potential of saving lives in an international perspective will be analyzed. It is shown that more than 6.000 lives could be saved per year in the European Union if the existing seat belts were used.

INTRODUCTION

For a couple of years there has been a concern within the Swedish road safety community about the fact that the safety potential of the seat belts is not fully used.

One of the first alarms came in a report from 1992 where fatally injured car occupants in Stockholm were studied. (Kamrén, 1992). The belt use in this group was only 40% compared to 80% in the general population.

At the last ESV Conference preliminary thoughts on a seat belt interlock system were presented by the Folksam Research Group. (Kamrén, 1994)

Another Swedish study of the belt use and the injuries showed again that the belt usage rate among severely injured was 50% in rural and 33% in urban accidents. (Bylund 1995)

In a Finnish study (Rathmayer, 1994) a clear pattern of the behavior of non-users was observed. Seat belt users committed one traffic offense in every 13 km on highways while the non-users committed one offense every 5,5 km. In urban traffic the distance between offenses was 9 km for the belted and 2,5 km for the unbelted. Non-users also drove faster and had driving histories with longer violation records than the control drivers.

Seat belt usage rates are generally observed in daylight. It can be assumed the rates are lower in darkness and in other situations where the risk for being caught without a belt is lower.

There are also a number of international research results that confirms these findings regarding the belt use and the non-user.

Being aware of that further campaigns and enforcement could only have a limited effect and that technical solutions like automatic belts were not realistic the Swedish National Road Administration last year formed a group of people representing the administration, research, insurance and car industry in order to analyze the situation and propose solutions. This paper reflects the thoughts of that group so far.

THE US EXPERIENCE

Because of various delays in introducing mandatory automatic protection in the USA in the beginning of the seventies the starter interlock requirement was introduced for the period August 15, 1973 until August 15, 1975 for vehicles without automatic protection produced during that period. These systems were connected to both front seats in such a way that if any front seat belt in an occupied seat was not locked, the starter was disabled. If a buckle was opened later a buzzer-light system was activated. All 1974 model year cars sold in the United States came with this ignition interlock except a few thousand GM models that came with airbags that met the automatic protection requirement.

In March 1974 NHTSA described the public reaction to the ignition interlock as follows:" Public resistance to the belt-starter interlock system currently required has been substantial with current tallies of proper lap-shoulder belt usage at or below the 60% level. Even that figure is probably optimistic as a measure of results to be achieved, in light of the likelihood that as time passes the awareness that the forcing systems can be disabled, and the means for doing so will become more widely disseminated,....". There were also speeches on the floor of both houses of Congress expressing the public's anger at the interlock system. On October 27, 1974 President Ford signed into law a bill that prohibited any Federal Motor Vehicle Safety Standard from requiring or permitting the use of any seat belt interlock system. NHTSA then deleted the interlock option from October 31, 1994.

Thus the interlock systems were required in the USA for 14½ months instead of the 24 months that were originally intended. (Kratzke. 1995)

LESSONS TO BE LEARNED FROM THE US EXPERIENCE

The failure of the interlock systems in the USA 1974 can be explained by the following factors:

- Many people felt that it was an infringement of personal freedom. This is probably a typical US reaction that may not be valid for e.g. the European market.
- The voluntary seat belt use was very low.
- The belt systems that were used in the USA at that time were usually difficult to use and had a bad fit.
- The interlock system itself was too unsophisticated. It did not allow low speed maneuvers or sitting in the car with the engine idling.

NEW APPROACHES

Basic principles

Some basic principles for a new system have been established:

- The normal seat belt user shall not notice the system.
- It shall be more difficult and cumbersome to cheat on the system than using the belt.
- Permanent disconnection of the system shall be hard to make.
- The system must be very reliable and have a long lifetime.
- All seating positions in the car shall be covered by the system.
- The accident risk must not increase by any malfunctions in the system.
- Retrofit systems for old cars should be available.

Detection and processing

One input to any interaction system is the situation in the car. Which seats are occupied and are the belts properly used on these seats?

The basic sensor for an occupied seat is a contact that will detect a certain load on the seat. This concept can give false signals from e.g. luggage on the seats. Modern techniques with photocells, IR-detectors, inductance, pattern recognition and load measurements on the seatback can be used to overcome most of the problems.

To determine if the belts are properly used is maybe more complicated. The US experience showed that simple systems like a switch in the buckle or measuring the amount of webbing coming out from the retractor could easily be tampered with. There is more information available from the belt system that could be used e.g. angles and forces at anchor points. Also sophisticated systems like pattern recognition or transponders in the webbing could be used.

Information from the doors, the seats and from the belt system can be combined and analyzed in such a way that the proper conclusions can be made.

Other problems that must be considered are how child restraint systems will work in this new environment and how to handle the situation when a passenger disengages his belt during travel. In that case it is probably not possible to influence the behavior of the car other than gradually and after some proper warnings to the driver.

Interaction

Several ideas for interaction could be considered, of which some are presented here.

The starter-interlock as used in the USA is the most aggressive solution. As mentioned above there are several shortcomings with this system so it is not on the agenda for the new approaches.

External visual signals is a new concept that is worth considering. By flashing the headlights or the hazard warning flashers the surrounding traffic (and the police) will notice the vehicle with non-belt users. The social pressure and the risk of being caught ought to be a good incentive to use the seat belts.

Internal light and sound warnings are used already in cars today but they can be made more aggressive and more directed to the individual non-user.

Interactions with comfort and audio systems is another approach that is discussed. This is a "soft" countermeasure but by disabling the radio, the air-condition, opening the windows etc. some users may get the message.

Throttle pedal feedback can also be used so that the force on the pedal will increase at a certain speed. This will make it possible but very tiresome to exceed that speed. Another solution may be to introduce severe vibrations in the pedal at a certain speed level.

Maximum gear level makes it impossible to put in any gear than number 1 and reverse. This takes care of one of the main faults with the US starter interlock which made it impossible to garage the car without using the seat belt. It also makes it possible to remove a stuck car from e.g. a railway crossing or a burning garage.

Maximum speed is a similar solution to the maximum gear level. The limit that is discussed so far is 30 km/h.

The final solution may be a combination of these systems i.e. the sequence can start with a visual and audible warning and then increase in intensity and finally reduce the maximum speed.

POTENTIAL EFFECTS

Sweden has got one of the highest seat belt use rates in the world with a front seat use of about 88% in observational studies. Other countries in Europe have a marginally higher use with UK in top with 91%.

In Sweden, the 88% use is to be compared to the less than 50% use among fatalities. The following table on the number of fatalities can be derived from the present situation in Sweden.

Table 1.
Seat belt use among fatally injured car occupants in Sweden 1994, based on a sample of 32 cases, and estimated number of fatalities with 100% belt use

	1994	with 100% belt use
Seat belt used	155	272
Seat belt not used	234	-
Total	389	272

	1994	with 100% belt use
Saved lives in relation to current seat belt use in Sweden (50% effectiveness)		117
Saved lives in relation to 0% seat belt use in Sweden (50% effectiveness)	155	272

The potential number of savings is 272 fatalities per year, but we have only come to a level where we have used 57% of the potential savings. This also shows that we have a higher benefit per user from the last 15% than we have had from the 85% seat belt use that we have today. This is different from other areas where the major benefits comes from the first part of an investment and with a decreasing marginal benefit. The relation between the seat belt usage rate in the population and the potential effect based on the Swedish situation can be described by the curved line in the following curve. A low usage rate gives a very limited effect since these individuals drive very safe anyway. The last 10% probably represents the most accident prone group so this is where we find the largest benefits from the belt use. The straight line describes the common belief that there is a linear correlation between the usage rate and the effect.

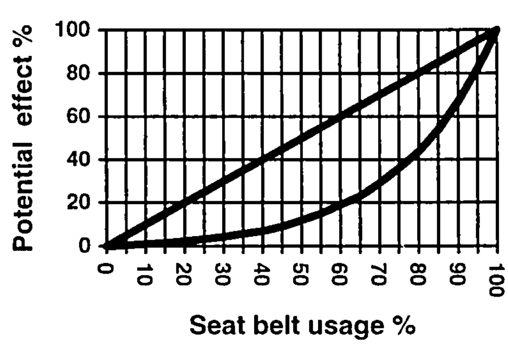

Figure 1. Correlation between seat belt usage and the potential effect on the fatalities.

If we use the Swedish figures and assume that the total European situation is not better, 15.200 unbelted occupants are killed every year in Europe. With a 100% seat belt use and a 50% injury reducing effectiveness, the total number of savings is around 7.600 per year. Given the Swedish situation, this is probably not an overestimation, although the potential savings may vary from country to country. It must be remembered that these figures apply only if the whole vehicle fleet is equipped with an interlock system.

A recent study from the European Transport Safety Council (ETSC) shows similar results with a potential reduction of 5.570 fatalities by a 95% belt usage rate.

Table 2.
ETSC estimations of seat belt use potential

Country	Killed car occupants 1993	Belt usage % Front seat 1991-95	Potential number of saved lives
Austria	747	70	175
Belgium	1050	55	277
Denmark	254	92	58
Finland	274	87	53
France	6168	85	1243
Germany	6128	92	1097
Greece	781	63	199
Ireland	187	53	51
Italy	3931	~55	998
Luxembourg	54	71	14
Netherlands	615	73	139
Portugal	1140	~63	234
Spain	3606	~75	834
Sweden	389	90	69
United Kingdom	1835	91	329
EU Total	27159	80	5570
USA	21987		

(IRTAD 1993) (ETSC 1996)

Only fatalities are discussed in this paper. An interlock system will of course also have a similar effect on the number of severely injured which is about 10 times larger than the number of fatalities.

ALTERNATIVE MEASURES

Preliminary calculations of the cost-effectiveness of an interlock system show that this is a very effective measure compared to some other ones.

By using the Swedish calculations of the willingness-to-pay for risk reductions, it is possible to calculate the possible economic benefits for an interlock system. It can be estimated that the savings from interlock in Sweden is in the region of more than 5 billion SEK/year (~700 million US$/year). With the medium age of cars that we have in Sweden for the moment , the cost that can be spent on each car for an interlock system is therefore approximately 20.000 SEK (~3.000 US$). With an anticipated cost of 200 SEK (~30 US$) per car for an interlock system, the ratio between benefit and cost is 100:1, which by margin is higher than for any other known safety measure. As an example, a 100% fitting of airbags from now on in Sweden would save approximately 50-60 lives annually, but for a cost that is ten times higher than for the interlock, still leaving us with a positive balance between cost and benefit, but serving as an indicator of the extreme benefits of interlock.

ATTITUDES

Preliminary results from a study made by the Swedish National Road Administration in 1995 based on interviews with 5914 persons aged 15-84 years show that there, in general, is a positive attitude for introducing interlock systems.

Table 3.
Swedish interviews

Do you agree or disagree that cars should not be able to run faster than 30 km/h if the driver is not using the seat belt?

%	Male	Female	Total
Strongly agree	24,1	36,6	30,2
Agree	17,7	19,8	18,7
Neither / or	14,1	15,1	14,6
Disagree	19,4	13,7	16,6
Strongly disagree	24,8	14,8	19,9
Total	100,0	100,0	100,0

Table 4.
Swedish interviews

Do you agree or disagree that cars should be equipped with buzzers and lights to warn that someone is not using the seat belt?

%	Male	Female	Total
Strongly agree	37,1	51,8	44,5
Agree	25,4	26,4	25,9
Neither / or	13,4	9,4	11,4
Disagree	12,5	7,2	9,8
Strongly disagree	11,6	5,3	8,4
Total	100,0	100,0	100,0

These two tables show that women are more positive than men and that the less aggressive buzzer-light system is preferred.

This investigation also shows that older persons are more positive to interlock systems than younger persons.

An alarming fact is that of those who state that they seldom or never buckle up in the front seat on rural roads we can find that 77% disagree or strongly disagree on a 30 km/h speed limiting interlock. 55% of this group are also against the buzzer and light warning system. This is actually our target group so we need to find out how to change their attitudes and how to prevent them from disconnecting the interlock system.

INCENTIVES FOR INSTALLATION

Since a legislation on a national level is difficult or impossible after Sweden has become a member of the European Union, other ways to have these systems installed in new and existing cars have been discussed.

- A majority of new cars in Sweden are bought as company cars. There is a possibility to lower the tax liability of the benefit in kind on cars if they are equipped with interlock systems.

- The general vehicle tax can also be moderated depending on the safety equipment of the car.
- Insurance companies are discussing to adopt the premiums along these lines.
- Another proposal is that drivers that are caught without using the belt will be obliged to install an interlock device in their cars.

FUTURE ACTIVITIES

Attitudes

During the spring of 1996 a survey of non-users will be made in Sweden. In cooperation with the police, non-users will be stopped and interviewed at 11 locations spread over Sweden. The interviews will concentrate on seat belts in general and reasons for non-wearing in particular.

System specification

A technical specification, probably in the format of a draft ECE-Regulation will be made during 1996. This draft will allow the car manufacturers to use different options but the goal will be a 99% belt use rate. Also retrofit systems for existing cars will be considered.

An interesting alternative to have detailed technical specifications is to measure the actual belt use rate in traffic for the different cars model years. The tax and other benefits could then be applied with about a one year delay from the introduction of a new car. This alternative will give the vehicle manufacturers free hands to do whatever they want to increase the use in their vehicles. If the coupling between the usage rate and the benefits for the car industry and the car user are strong enough this approach could lead also to a voluntary installation of interlock systems in the existing car fleet.

Implementation

Since this concept has been very positively accepted by the Swedish Ministry of Transport, discussions will go on in order to find the proper tax and other incentives to have some kind of system implemented on the Swedish market as soon as possible.

The car industry may object to this since they do not want to have different equipment on different markets. An international standard would of course be better for everybody but considering the time it will take - and the number of unbelted people killed during that time - our position is that we ought to do something wherever it is possible to get something done quickly in this field.

CONCLUSIONS

At least 6.000 lives can be saved in the European Union annually if the seat belts that are already in the cars are used by 100% of the occupants. The only way to reach this level is to have a technical solution that will make it impossible or very cumbersome to use the car without using the seat belts. There are several technical solutions available that could be implemented in a short time. The main obstacle to reach this goal is probably of a political nature.

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IMPROVED OCCUPANT PROTECTION THROUGH ADVANCED SEAT DESIGN

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ABSTRACT

Through a contract from National Highway Traffic Safety Administration (NHTSA), EASi Engineering in conjunction with Johnson Controls Inc.(JCI) is working to conceive and develop an advanced integrated structural seat that meets the current FMVSS requirements and significantly improves occupant protection for frontal, rear, side and rollover accidents and contributes to passenger compartment intrusion resistance. This work is a cooperative effort between the government and industry, bringing together the strengths of impact biomechanics, computer aided engineering and seat systems engineering and manufacturing.

This paper summarizes the advanced integrated structural seat criteria used, the design concepts evolved and adapted thus far as part of this ongoing research, the evaluation of the design concepts using various computer aided engineering (CAE) methodologies, and the resulting changes in occupant crash protection. Concept level models were created primarily through use of the MADYMO software to establish potential benefits. Further design evolution and evaluation were achieved via detailed finite element models and coupled models using LS-DYNA3D and LS-DYNA3D/MADYMO coupling. The design concepts studied include rollover-sensing seat belt pretensioners and extended head rest frames for improved rollover protection, belt load limiters for improved frontal crash protection, energy absorbing dual recliners, strengthened seat back wing structures for improved side impact protection and side intrusion resistance. This study does not include seat mounted side airbags as they have been explored already (Pilhall et. Al.) and are in production.

INTRODUCTION

The advanced integrated structural seat (AISS) is aimed at enhancing occupant protection in all of the four basic crash modes (frontal, rear, rollover and side crashes) primarily by modifications of the seat structure and by addition of seat mounted safety/restraint features. By focusing on the seat structure modifications, it is hoped the resulting designs will be simple and cost-efficient. Also, the seat system is designed to function with the body structure to resist passenger compartment intrusion in side and rollover crashes. This seat system design is being developed starting from an existing integrated structural seat design. Integrated seats have the belt anchorages on the seat itself as opposed to conventional seats where the shoulder belt upper anchorage is located on the car upper body structure. The belt fit is considerably improved regardless of the seating position. Also, the assembly of the seat in the car becomes much easier with this design as the belts are part of the seat. An integrated structural seat was chosen as the baseline seat since it is expected to enhance occupant protection.

The criteria matrix shown in Table 1, lists the loading conditions and evaluation criteria for the seat in various crash modes. This matrix is established based on current regulations and a sample of industry design practice. The matrix should not be considered a statement of NHTSA's future regulatory intentions.

Specifically, this paper focuses on design concepts studied for the frontal and rear impact protection. The concepts, their evaluation procedure and detailed design are described in the following sections.

FRONTAL CRASH PROTECTION

Load Limiter

Recent work done in evaluating injury reduction in frontal crashes suggests that belt restraints and airbag restraints may not interact in a way which achieves optimal occupant protection (Mertz at al, 1995). Ideally, at frontal collision speeds below the threshold of air bag deployment, torso belt forces should be limited to those levels required to prevent occupant impact against compartment interior surfaces such as steering wheels and instrument panels. Additionally, for the AISS (Advanced Integrated Structural Seat) design, the torso belt should sustain loads capable of retaining the occupant within the compartment during rollovers and side crashes. Such reduced torso belt load limits are far below current practice (Figure 1). The upper anchorage of the torso belt on the seat back structure of current integrated seats is the source of the greatest seat back bending moment and shear load on the seat structure. As

Table 1 Criteria Matrix

	Criteria	Loading: Speed/ Location	Loading: Load	Characteristics of Interest	Dummy	Back Position	Remarks
COMPLIANCE	Rollover	30 mph	Drop from rollover dolly (FMVSS 208)	Head excursion; Neck injury; Shoulder belt loads on occupant; Failure mode	50th & 95th %ile male Hybrid III belted	Design	Consideration for 5th %ile female head excursion relative to torso
	Rear impact	35 mph	301 crash pulse extrapolated	Shoulder belt loading; Neck injury; Ramp up; Back collapse; Rebound	50th & 95th %ile male Hybrid III belted	Design	Hybrid III may not be adequate; Check for 5th %ile
	Side impact	33.5 mph	MDB	TTI; Pelvic injury; Head injury; Head Excursion	SID belted	Design	Head excursion relative to torso
	Frontal impact	30 mph	30 mph pulse	Head injury; Chest g; Anti submarining Rebound	50th & 95th %ile male Hybrid III belted	Design	Airbag interaction to be considered; Check for 5th %ile female; Submarining
PRACTICE	Torsional Rigidity	Seat back corner	2260 in-lb about H-point	Permanent set		Design	
	Abuse load	Seat back crossbar	6400 in-lb rearward about H-point	Seat Integrity		Design	
	Submarine loads	Cushion frame	Simulated	Seat Integrity			
OTHER	Cost estimate	N.A.	N.A.	Market acceptable range			
	Ingress/Egress	N.A.	N.A.	Ease			
	Styling	N.A.	N.A.	Acceptable practices			
	Manufacturing	N.A.	N.A.	Mass production feasible			
	Vibrational characteristic	N.A.	N.A.	Away from discomfort range			
	Weight	N.A.	N.A.	Within market range			

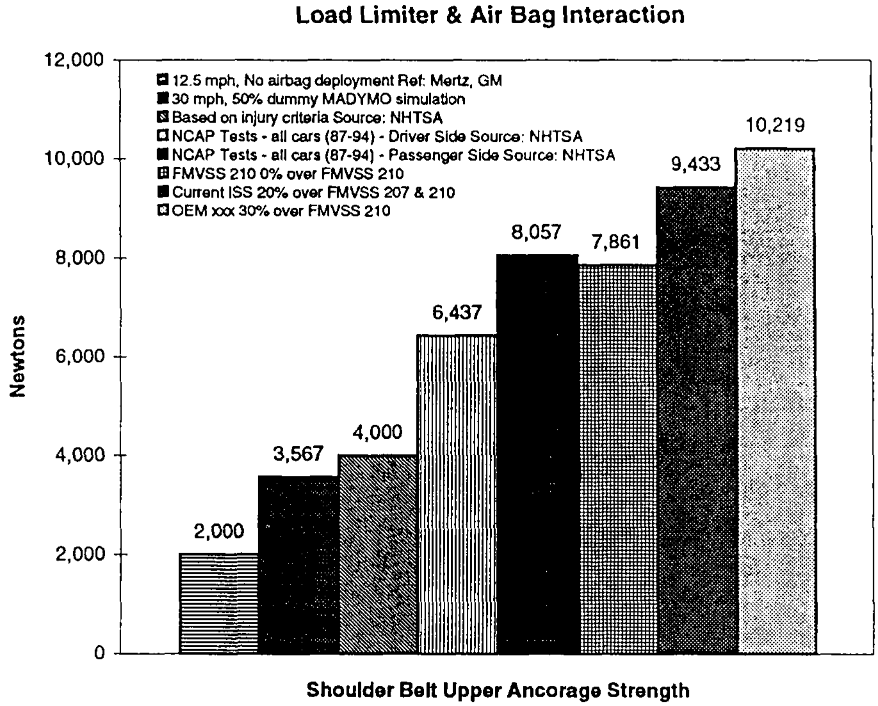
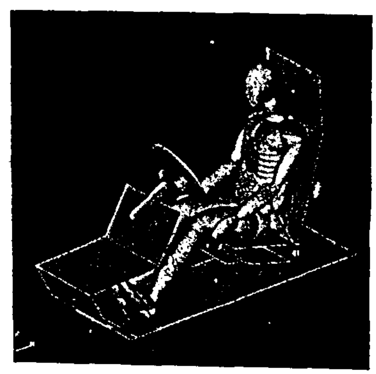


Figure 1 Shoulder belt loads seen in different studies

a result, limiting the torso belt loads allows weight reduction of the seat back structure and reduced floor pan shear while reducing occupant injuries at higher crash severities. At present, significant re-designing of the vehicle floor pan is required to adapt it for an integrated structural seat. Introduction of load limiter may reduce the extent of redesign required to replace a conventional seat (shoulder belt upper anchorage on the vehicle structure, mostly the b-pillar) with an integrated structural seat.

Mertz et al. (1995), have shown vast improvement in occupant injury parameters for 50th percentile occupant by limiting the seat belt loads to 2000 N. They have shown a 27 percent reduction in chest acceleration and 67 percent reduction in chest compression. The risk of AIS $\geq$ 4 was reduced from 14.5 percent to 0.4 percent, and the risk of AIS $\geq$ 3 was reduced from 94 percent to 19 percent. The 95th percentile occupants although, were studied only at low speeds (15 mph) in non-deploy situations.

In the current study, a rigid body MADYMO model validated using a front impact sled test, is used for evaluating the torso belt load limit. The model is set up for Ford Taurus environment with an existing integrated structural seat design (Figure 2). The seat model includes the seat back joint stiffness, seat cushion stiffness, anti-submarining plane and a 3-point seat belt. Eight percent nominal belt stiffness is used. Three different impact



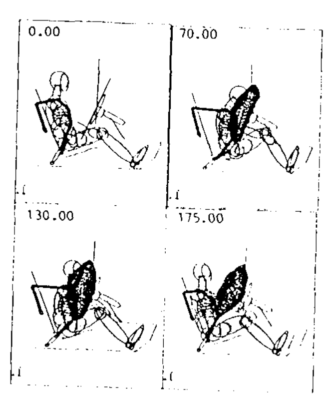


Figure 2 The validated MADYMO model and the crash sequence for 30 mph frontal impact.

velocities of 12, 30 and 40 mph are studied, with and without the presence of a load limiter. The airbag and the inflator model are assigned characteristics taken from a production airbag. A small (40 liter) and a large (80 liter) airbag are used for the study.

A load limit of 4000 Newton is found to be ideal based on a new research on rib fracture by Kallieris et al. In the current study, no head to wheel contact is seen for a 50th percentile Hybrid III dummy, at a 30 mph frontal impact with a 4000 N load limit. Figure 3 shows the belt payout for a range of load limits for 30 mph impact pulse. The load limit for a 95th percentile male, that produces no wheel to head contact is found to be 4500 Newtons.

LOAD LIMITER FORCE & BELT PAYOUT FOR 30 MPH

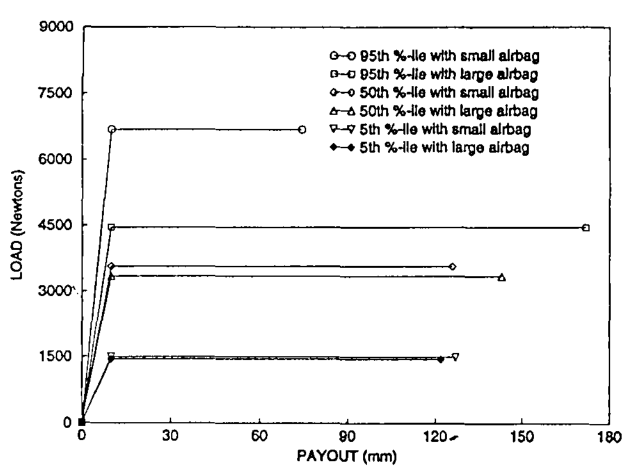


Figure 3 Belt payout for several load limits at 30 mph

Figure 4 and 5 show the bar charts for the HIC and chest compression for the 5th, 50th and the 95th percentile occupants respectively, for 30 mph frontal impact. With the 4000 N load limit substantial reduction in HIC values and neck loads is seen. HIC is reduced by 51%, neck loads by 15% and chest compression by 14% for a 50th percentile occupant with a 40 liter airbag. This translates to a 30 percent reduction in the risk for AIS ≥ 3 thoracic injury.

Vast reduction in belt loads due to the introduction of load limiter (Figure 6), would make one expect significant reduction in chest compression values. But, from Figure 5 it is clear that chest compression reductions are only moderate. This is explained by the influence of airbag in frontal impact. Limiting of the belt loads causes the airbag to pick up the loads. Optimum design to minimize injury indices would involve concurrent tuning of the seat belts, the load limiter and the airbag. This is further illustrated by the vast differences in injury parameters between the three sizes of occupants, for the two sizes of airbags.

If a load limiter, that causes increased belt payout, is activated in an impact mode other than frontal, e.g. rollover, the risk of injury to the occupant could increase. However, rollover simulation performed by EASi as part of this research show shoulder belt loads much below 4000 N. Lap Belt with retractor and pretensioner significantly limits the motion of the occupant in rollover and hence lower loads on the shoulder belt are seen.

Several load limiters based on different concepts such as, stitch tearing, torsion rod, shearing/extrusion etc. are available in the market. Most of them are capable of limiting the load at 4000 N.

HIC

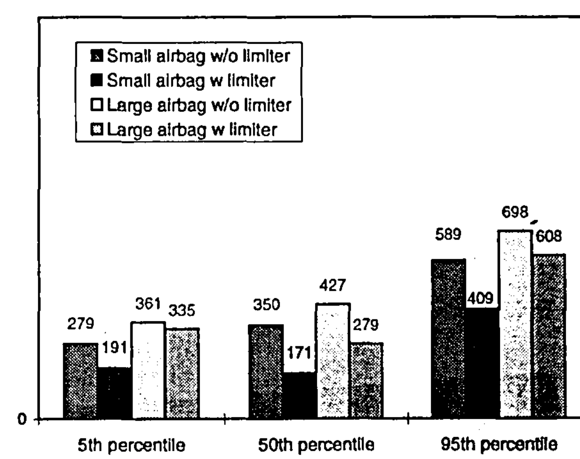


Figure 4 HIC values for 30 mph frontal impact

Chest Compression
(mm)

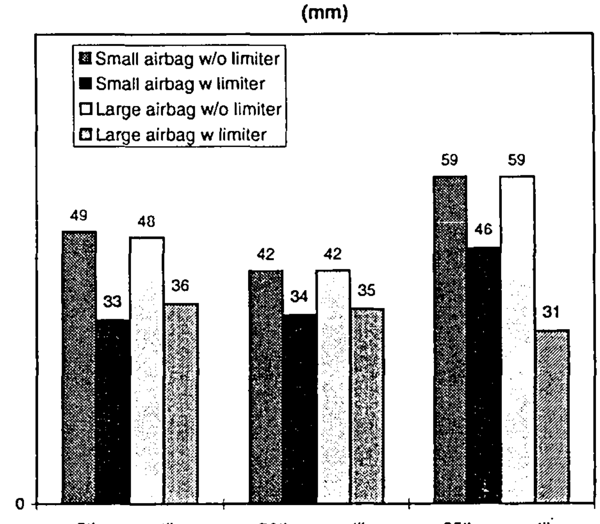


Figure 5 Chest Compression (mm) values for 30 mph frontal impact.

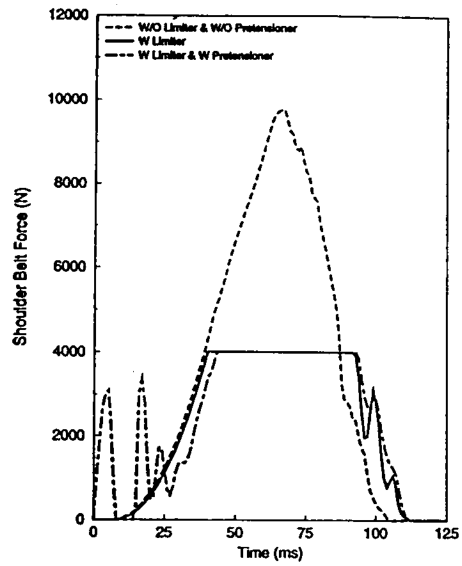


Figure 6 Shoulder belt load vs. Time for 30 mph

<u>**Pretensioner**</u>

To realize the effectiveness of shoulder belt load limit, belt slack and loose webbing wrap should be minimized. This can be achieved with the device called pretensioner, which takes up belt slack early in the collision by pulling on the belt at the buckle or the retractor location. This induces energy absorption during the early forward travel of the occupant in a frontal impact. This is illustrated in Figure 6 which compares the belt loads vs. time for the cases of : (1) no load limiter, (2) with load limiter and (3) load limiter with pretensioner in 30 mph frontal impact. Figure 7 shows the bar chart for injury numbers with and without the pretensioner in the presence of a load limiter for 50th percentile dummy at 30 mph.. More than 10 percent reduction is seen for all the injury parameters, with the resultant chest acceleration (3MS) reducing by 50%.

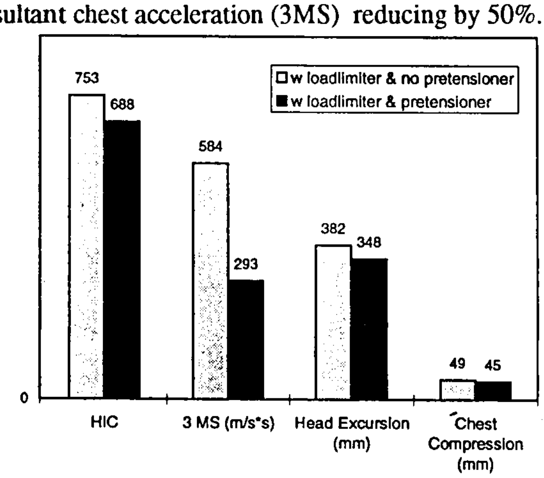


Figure 7 Effect of pretensioner on injury parameters (50th percentile dummy at 30 mph)

Work done as part of this study has shown improved occupant protection by the use of pretensioner in rollover crashes. More work is in progress and will be reported at a later time. Introduction of belt pretensioner has also been shown to significantly reduce the risk of submarining (Haland et al., 1993). The pretensioner (buckle mounted) helps prevent submarining by reducing the slack (Leung et al., 1982) and by pulling the buckle downwards, which narrows the opening for pelvis to slide through. A buckle mounted seat belt pretensioner with specifications as shown in Table 2 is used in this study.

Table 2 Specifications for the Belt Pretensioner

Pulling Distance	80 mm
Pulling Time	9.5 msec.
Pulling Force	<1000 N (Depending on Test Set-up)
Opearting Temparature	-40°C to 100°C
Weight	450 grams
Type	Pyrotechnic, Buckle Mounted

REAR IMPACT PROTECTION

Neck injuries with risk of permanent disability are frequent in low severity rear-end collisions (Carlsson et. al., 1985). Studies by Langweider (1981) and Kahane (1982) suggest that of those occupants injured in rear impact accidents 80 to 90 percent suffer neck injury. Because such accidents are common, they cause significant human suffering and high societal costs, despite the fact that the injuries are usually classified as "minor" (AIS 1) in the Abbreviated Injury Scale (AIS) (Nygren, 1984 and Nygren et. al., 1985). Analysis of CO-operative Crash Injury Study (CCIS) database (Renouf, 1991) has indicated that 95 percent of neck injuries to front seat occupants are recorded as AIS 1. The importance of certain seat (specially seat back) characteristics on rear impact and criteria relevant to minimize the related injuries are discussed below. Seat back bending stiffness strongly influences occupant response in rear impact. Seatback rotation can be beneficial from an energy absorption standpoint. A seat back that collapses without absorbing energy is not desirable. A study on protection against rear end accidents (Thomas et al., 1982) suggests that failure of the seat back or mountings has a greater effect on cervical spine injury than the head restraint. However, a small number of cases exist where rear seat occupants have been killed by the front seat collapsing on to them (Lowne et al., 1987). However, in an integrated structural seat, large rotation angles will cause greater demands on the shoulder belt in restraining the occupant from sliding backwards. At the same time, excessive rotation will encroach on rear seat occupant space. Therefore 30 degrees seat back rotation from the design position is selected as the maximum allowable for the 95th percentile dummy under a 30 mph rear impact crash pulse. This seat back rotation angle is consistent with the current industry practice.

In contrast to a seat back that deforms too easily, a rigid seat back may cause occupant rebound (Partyka et al.). The elastic springback energy stored in the seat is sufficient to throw the occupant far enough to hit the steering wheel or the dash. It may also cause the occupant to ramp up, which may lead to partial or complete ejection; increasing head to neck torque. Due to ramp up of the occupant, the head may rise above the headrest leaving no support to stop the head from tilting rearward. This leads to "whiplash" related injuries. Seat back design should aim at minimizing occupant ramp up and rebound, and at the same time contain seat back rotation. In this study the maximum seat back rotation was restricted to 30° from the design position. Effect of the various seat characteristics described above, on the occupant, is reflected in the occupant injury numbers, such as - HIC, neck extension, head-to-neck flexion torque etc. While a seat needs to address the management of energy transfer to the occupant in severe rear crashes, biomechanical responses need to be below tolerance levels and proportionately lower with decreasing crash severity for overall injury prevention.

Some form of limited and controlled deformation of the front seats is therefore desirable. In this way energy could be absorbed by the seat, reducing the risk of injury to the front seat occupants, without endangering rear seat occupants. In this study several design features are explored to address the above injuries and to reduce them below the human threshold limit. Table 3 relates the design features to the different requirements The evaluation methodology and the results for various design features are described below.

<u>Seat Back Structure</u>

Torsional resistance of the seatback was considered as part of this project. A second recliner was added to the inboard side of the existing seat to consider the effect of that configuration on torsional rigidity. The existing baseline seat design has a single linear recliner on the outboard side and shows twisting in 30 mph rear crash. The analytical models used to evaluate the existing seat with and without dual recliners are described below.
To study the performance of the existing seat in rear impact, a nominal static design load that the seat must withstand is first estimated from a series of MADYMO simulations. The seatback is modeled in MADYMO as a pivoted structure which is restrained by a resistive torque function. The pivot is located at the recliner position. The torque function at the pivot defines the elasto-plastic bending stiffness characteristic of the seatback. Figure 8 shows the MADYMO model and crash sequence in rear

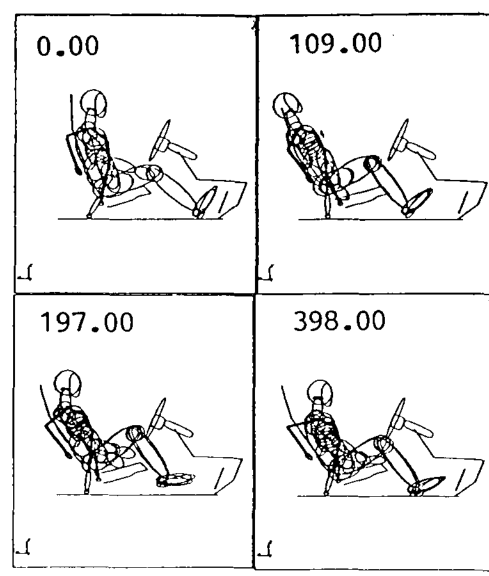
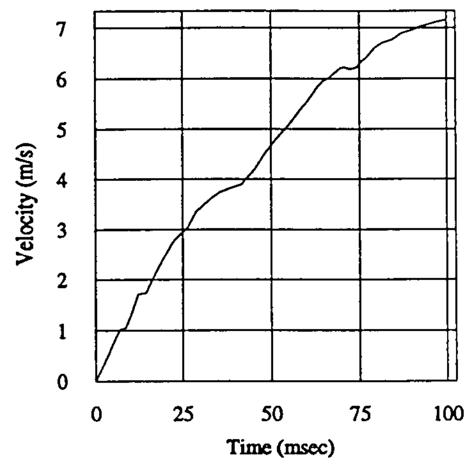


Figure 8 Crash sequence in rear impact

Table 3 Participation of the proposed design features in addressing various rear impact performance criteria

	Dual Recliner	Modified Seat Back	Energy Absorber	Inflatable Headrest	Integrated Seat Belts
Whiplash			X	X	
Excessive Seat Back Deformation	X	X			
Ramp Up			X		X
Rearward Ejection	X	X			X
Excessive Rebound			X	X	

crash for a 50th percentile occupant. Similar models were developed incorporating the 95th percentile and 5th percentile Hybrid-III dummy models. A 30 mph rear impact crash pulse is used in the model. This pulse (Figure 9) is obtained from a FMVSS 301 (fuel integrity) moving barrier crash test at 30 mph. The resulting change in velocity of the struck vehicle is 20 mph.



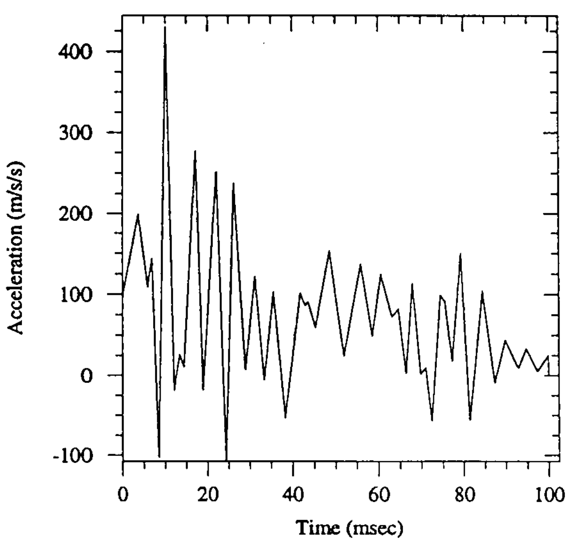


Figure 9 The acceleration pulse used for the study

A series of MADYMO simulations with different seat back stiffness characteristics were performed until satisfactory response, meeting the criterion of a maximum seat back rotation of 30 degrees for the 95th percentile dummy is achieved. Figure 10 shows the torque vs. Seat back rotation characteristic for the seat back that produces the desired maximum seat back rotations shown in Table 4.

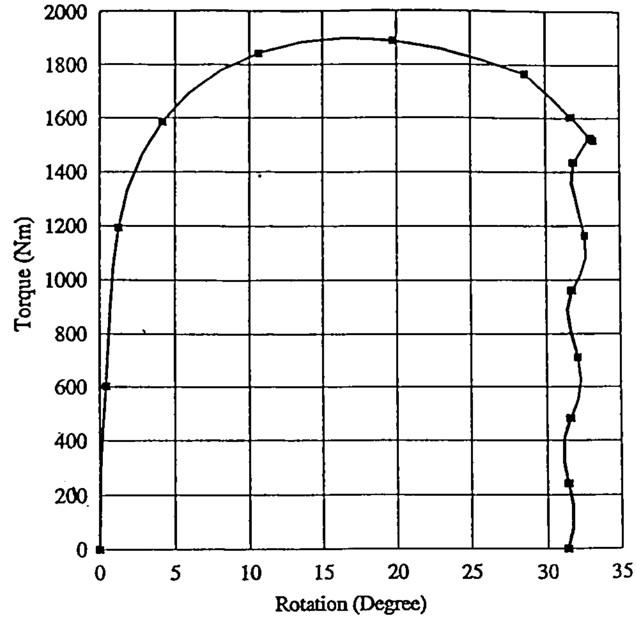


Figure 10 Torque vs. rotation curve for the seat back

Table 4 Maximum seat back angle with respect to the seat bottom.(Initial seat back angle = 23°)

	5th percentile	50th percentile	95th percentile
Rotation of the seat back	10°	22°	29°
Peak seat back angle (wrt. vertical)	33°	45°	52°

Equivalent torque applied to a finite element model of the existing seat produces a seat back twist of approximately 15°. A 17° rotation is seen using a LSDYNA/MADYMO coupled model for a 95th percentile dummy in 30 mph rear crash. This rotation is decreased to 3 degrees for the modified seat with a second recliner added on the inboard side.

Recliner With Energy Absorber

Occupant rebound and ramp up can be minimized by designing a seat back that deforms plastically in a controlled manner. Rebound is caused primarily by the elastic energy stored in the seat back during rearward deformation, which is imparted to the occupant during the forward travel. In order to obtain the necessary compliance in the rearward direction a mechanical energy absorbing element was added in series to both the recliners on either side of the seat.

The design of this device (described later) is such so as to modify the Torque vs. Angle function of the seat back to approximate the desired curve originally obtained from the MADYMO model (Figure 10). It should be noted that the Torque vs. Theta curve has a very sharp rise followed by a relatively flat region. Upon unloading at any point along the curve, the drop in force is also very sharp. This implies that the amount of elastic energy stored in the seat back is very small compared to the amount of energy absorbed by the energy absorber. A typical existing seat back would have a Torque vs. Theta curve which has a much lower initial slope. This translates to a higher proportion of stored spring back energy compared to the dissipated energy. The difference between these two cases is illustrated in Figure 11.

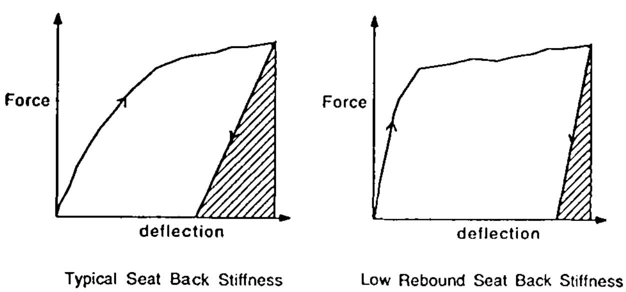


Figure 11 Energy absorption of a typical seat back and low rebound seat.

In order to quantify the amount of forward excursion due to seat rebound, MADYMO simulations, with and without seatback rebound, were performed using the rear impact model described earlier. Figure 12 summarizes some of the results, and graphically illustrates how the forward excursion of the 50th percentile dummy is affected by rebound as a function of rear impact velocity. Head excursion is defined as the distance of the head from the initial pre-impact position to the forward most rebound position.

To check the performance of the modified design in frontal impact, a non-linear finite element analysis is performed to simulate a 4,000 N load limited shoulder belt load for a frontal crash. The seat withstands this loading condition very well. The seat back rotation is less than 5°.

<u>Design for Energy Absorber</u>

This section describes the development of an energy absorbing recliner which serves the purpose of deforming the seat back in a controlled manner. The energy absorber is expected to deform by about 50 mm to produce the desired amount of rotation (maximum of 30°) of the seat back. Design evolved in this study utilizes the support plates of the existing recliner with appropriate modifications. The energy absorption is primarily achieved by the shearing of metal as the recliner pin traverses down a tapered slot created in the supporting metal plates.

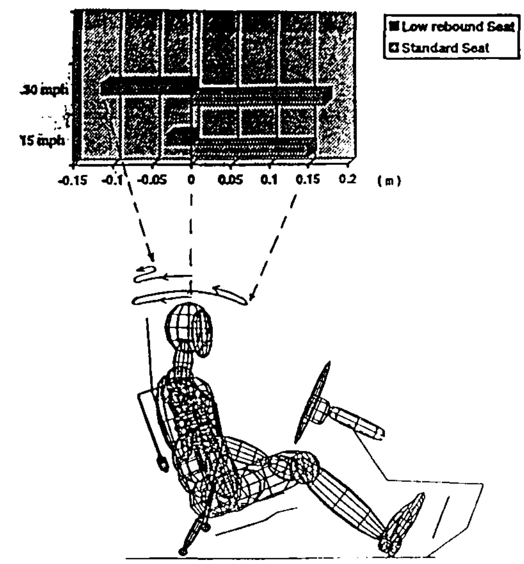


Figure 12 Head excursion due to seatback rebound

For modeling purposes, a portion of the seat is cut to isolate the recliner and the surrounding area. Tapering slots are created in the recliner support plates. The recliner diameter is 15 mm and it drives down the metal slots which decrease in width from 90% of recliner diameter (at the top) to 40% (at the bottom). About 50 mm of crush space is made available. Since slot width is smaller than the diameter of recliner, resistance is offered by the slot as recliner tries to drive down through it. Metal shearimg occurs in the process and energy is absorbed. The recliner is assumed rigid for this purpose. The basic layout of the modified recliner and the existing recliner is shown in Figure 13.

A mechanical device is added to the energy absorber to (1) avoid activation of shearing mechanism for low speed crashes and (2) to avoid rattling which would occur if there were no firm support to the recliner. The force-displacement curve for this metal element 1 mm thick is shown in Figure 14. A sharp increase in the strength can be noticed during initial part of the of the simulation.

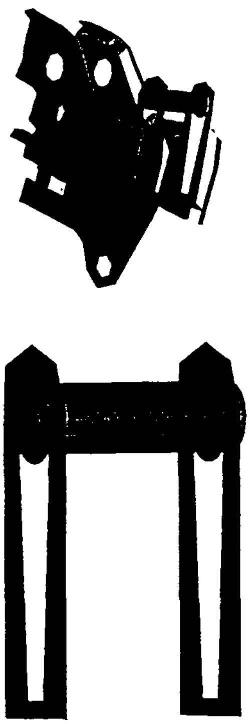


Figure 13 Layout of the modified recliner

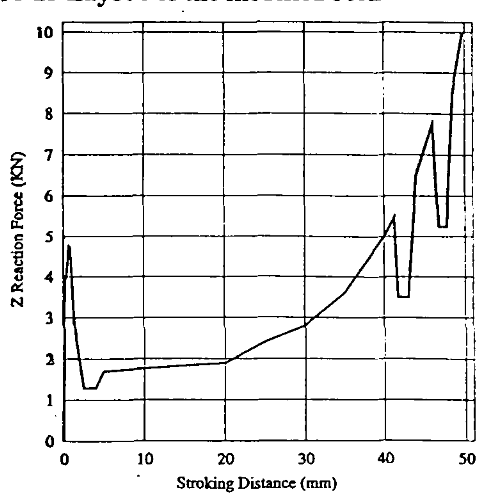


Figure 14 Torque-theta curve with the metal element

This would prevent the activation of the energy absorber at low speed crashes.

The energy absorber design is developed and tested using a very refined finite element model of the recliner support plates and the surrounding areas. Due to a very small time step of this refined model, analyzing a full seat model is computationally very expensive. Hence, the seat structure represented by beam elements and the dummy by lumped masses, is attached to the refined recliner model. Once the design for the energy absorber was finalized, then it was further tested using a LS-DYNA3D/MADYMO coupling method described below.

DESIGN VERIFICATION

The various design features for rear impact protection were finally tested using a detailed model. A detailed finite element seat model, incorporating the Dual Recliner, Modified Seat Back and the Energy Absorber, is coupled with the MADYMO model of the 50th percentile Hybrid III dummy. The dummy model used has been enhanced by EASi for greater biofidility. The hip joint has been released. The neck to upper torso joint stiffness has been modified to represent the rearward extension of the neck (Kolita et al.). The characteristics for the EA are obtained from the detailed EA model and represented in the full seat model as a spring element, in series with the recliner.
The coupled simulation is carried out for 200 msec. with the same rear impact crash pulse as used in the MADYMO study (Figure 9). Figure 15 shows the set up for the coupled simulation. The results are presented in Table 5. Table 5 lists the results for the 5th, 50th and

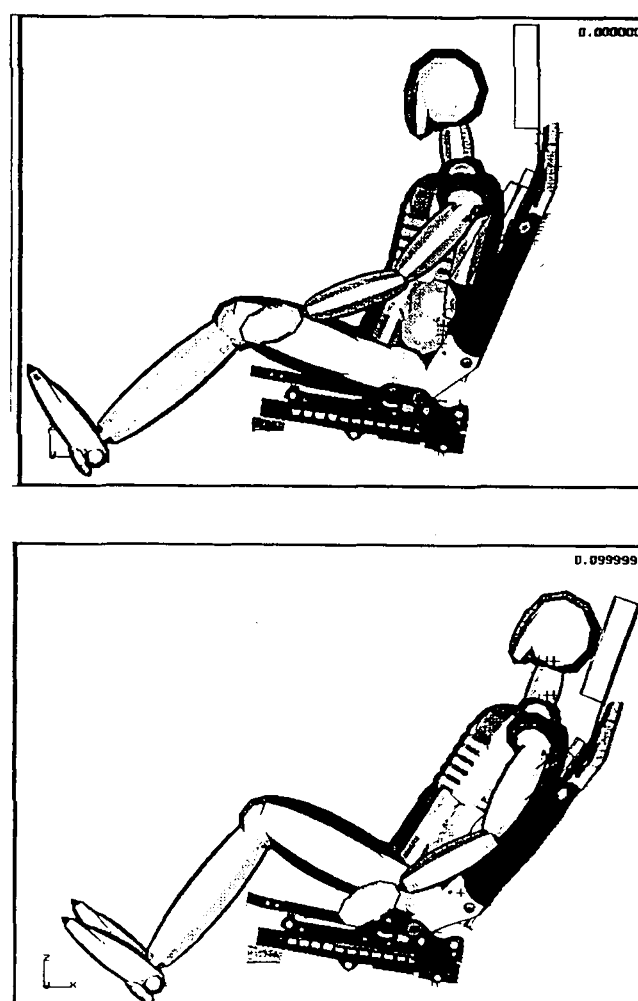


Figure 15 Model set up for the coupled simulation

Table 5 Injury numbers for 5th, 50th and 95th percentile dummy models

	50th %-ile Coupled	*Tolerance Levels*	*5 %-ile MADYMO*	*50 %-ile MADYMO*	*95 %-ile MADYMO*
DISPLACEMENT (M)					
Chest compression	0.0016	0.0760*	0.0011	0.0087	0.0074
Head excursion			-0.0858	-0.1531	-0.2299
Left shoulder			0.2143	0.2908	0.3409
Left seat corner			0.1506	0.2156	0.2377
ACCELERATION (M/S**2)					
Lower Torso	173		289	210	275
Chest	155		175	242	261
Head	248		343	612	620
FORCE (N)					
St. back-lower torso			5648	5863	10824
St. back-left shoulder			444	110	543
St.back-right shoulder			426	115	554
Shoulder belt			891	1352	1330
FLEXION TORQUE (N.M)					
Head-neck	10	60*	27	24	17
CONSTRAINT FORCE (N)					
On head from neck	258		791	1071	1179
On neck from upper torso	338		736	1159	1228
INJURY					
3 MS (m/s**2)	94	588	167	220	223
HIC		1000	43	259	240
ANGLE (DEGREES)					
Seat back angle w.r.t. vertical (initial=23)	43		37.6	44	46

* Armenia-Cope at al., 1993.

95th percentile dummy obtained from the MADYMO models. The table also lists the results for the 50th percentile dummy obtained using the LSDYNA/ MADYMO coupled model. The coupled model has a finite element representation of the seat structure, which is more accurate compared to the rigid body assumptions of the MADYMO model. Table 5 also lists the human tolerance values for some of the injury parameters. The injury numbers for the AISS design are well below the human tolerance levels. The injury parameters are compared with the human tolerance values. The final design was also tested for the 5th and the 95th percentile dummies using a MADYMO simulation. These results are reported in Table 5 and compared with the human tolerance values.

FUTURE WORK

- The analytical results presented here are to be verified with prototyping and testing in the future.
- Incorporating the design concepts proposed here into production will be based on cost versus benefit analysis.

SUMMARY

- 4000 N load limit is found to be favorable for 30 mph frontal impact for a 50th percentile Hybrid III dummy. A 30 percent reduction in the risk of AIS≥4 thoracic injury is seen.
- Belt pretensioner increases the effectiveness of shoulder belt load limiter in frontal impact. 50 percent reduction in 3MS (measure of chest injury) is seen with the use of pretensioner. Belt pretensioner is also found to be very effective in rollover crashes by reducing head excursion.
- Design for a energy absorber in series with a linear recliner is successfully developed.
- The energy absorber meets the maximum seat back rotation guideline for the 5th, 50th and the 95th percentile dummies.
- The occupant kinematics and injury parameters are very favorable for the 50th percentile dummy, calculated using a LS-DYNA/MADYMO coupled simulation.
- Work is currently in progress to develop designs for concepts such as: extended head rest for rollover protection, inflatable head rest for rear impact protection and seat back mounted side wing structures for improved side impact protection.

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NHTSA'S IMPROVED FRONTAL PROTECTION RESEARCH PROGRAM

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ABSTRACT

In the United States, within the next few years air bags will be required in all passenger cars and light trucks under Federal Motor Vehicle Safety Standard (FMVSS) No. 208, Occupant Crash Protection. Even after full implementation of driver and passenger air bags as required by FMVSS No. 208, frontal impacts will still account for up to 8,000 fatalities and 120,000 moderate to critical injuries (.i.e., injuries of AIS $\geq$ 2) [1]. The National Highway Traffic Safety Administration (NHTSA) has an ongoing research program to address these fatalities and injuries and provide a basis for the possible future upgrade of FMVSS No. 208. This effort includes developing supplementary test procedures for the evaluation of occupant injury in higher severity crashes, developing improved injury criteria including criteria for assessing injuries to additional body regions, and evaluating the injuries associated with occupant size [2-4].

More recently, in monitoring the fleet performance of current air bag systems, NHTSA has identified aggressive air bag deployment as a potential cause of injuries and fatalities of occupants in minor severity crashes. Accordingly, the agency has added new activities to investigate this finding in its frontal crash protection research program.

This paper presents an overview of the agency's overall research program. Selected results from the testing conducted to date are discussed. Finally, a discussion is presented toward improving occupant protection systems in frontal crashes.

INTRODUCTION

In the United States, air bags with lap and shoulder belts are specifically required by legislation (i.e., the National Highway Traffic Safety Administration Authorization Act of 1991) for both front outboard seating positions in all passenger cars manufactured after September 1, 1997. They are also required in all light trucks, multipurpose passenger vehicles (e.g., vans, utility and sport vehicles), and buses with a gross vehicle weight rating of 3,846 kilograms (8,500 pounds) or less and an unloaded vehicle weight of 2,489 kilograms (5,500 pounds) or less manufactured after September 1, 1998. However, with current high consumer demand for these systems, it is expected that manufacturers will install them several years earlier than the mandatory deadlines. Using sales data from Automotive News' 1995 Market Data Book and projected sales from DRI/McGraw-Hill's publication, Review of the U.S. Economy, the agency projects that almost 100 percent of the new 1995 passenger cars will have driver systems and 87 percent will have passenger side systems; and almost 85 percent of the new light truck vehicles will have driver systems and 23 percent will have passenger side systems.

The detailed performance requirements for these systems are contained in Federal Motor Vehicle Safety Standard (FMVSS) No. 208, Occupant Crash Protection. The main dynamic performance requirements in FMVSS No. 208 involves successful crash testing into a rigid barrier with a 50th percentile adult dummy at all speeds up to 48 kilometers per hour (30 miles per hour) at all angles between perpendicular and 30 degrees to either side of perpendicular. The tests can be run both with the dummy being unbelted and with the belts on. "Successful" crash testing requires that the dummy Head Injury Criterion (HIC) be at or below 1,000, the dummy chest deceleration be at or below 60 G's, and the dummy femur loads be at or below 10,000 Newtons. If a Hybrid III dummy is used, the chest deflection must be less than 75 millimeters.

Even after full implementation of driver and passenger air bags as required by FMVSS No. 208, it has been estimated that frontal impacts will still account for up to 8,000 fatalities and 120,000 moderate to critical injuries (i.e., injuries of AIS $\geq$ 2). The objective of this research program is to address these fatalities and injuries and provide a basis for the possible future upgrade of FMVSS No. 208 injury criteria and test devices, and the development of supplementary test procedures for the evaluation of occupant injury in higher severity crashes.

A detailed definition of the remaining safety problem for frontal impacts, subsequent to the full implementation of the current dynamic frontal crash protection standard, has been initiated. Research is underway for investigating the real world crash environment and projecting the occupant injuries that will occur for an all air bag fleet. This includes summarizing the human loading and injury tolerances relevant to frontal crashes. Also, test surrogates and injury criteria are being recommended for use in the crash testing.

Defining the problem includes identifying general laboratory test conditions that can be used to replicate the safety performance of air bag vehicles in use. Then, evaluating the performance of a variety of production vehicles under those preliminary crash conditions, comparing their performance, and conducting potential benefits assessments to guide the agency for the "final" selection of a test procedure(s).

This paper presents an overview of the agency's overall research program. Selected results from the testing conducted to date are discussed. Finally, a discussion is presented toward improving occupant protection systems in frontal crashes.

CRASH ENVIRONMENT

For projecting the occupant injuries that will occur and identifying general laboratory test conditions that can be used to analyze the safety performance of baseline vehicles, the agency's National Accident Sampling System (NASS) files for the years 1988-94 were used. The NASS is a statistical sample of the United States accidents investigated in detail. About 5,500 accidents per year are investigated. The NASS files for these years differ from those of previous years in that only the more serious accidents qualified for inclusion into the files. Also, the files include a variable for the description of vehicle intrusion which allows for the identification of up to ten intruding components per vehicle. Finally, a coding change occurred during this time frame to redefine a parameter that could be used to calculate the frontal percent of overlap due to direct contact with the struck vehicle.

Crashes involving air bag-equipped vehicles have been increasing along with the increasing installations. Between 1988 and 1994, the NASS teams investigated 35,146 crashes, representing an estimated 15.4 million crashes and 14.0 million injured vehicle occupants nationwide. In these crashes, 1,352 air bag deployments were investigated, representing an estimated 415,543 air bag deployments that occurred during that time frame. When comparing drivers with air bags to those without air bags both moderate and serious injury risk is about the same with belts used, not used or "as used." (Figure 1, and Table 1.) However, for fatalities air bags have significantly lower rates for no belts and "as used" and about the same for belts (Figure 2.) Figure 3 shows the risk of injury by body region in frontal crashes for occupants of air bag equipped vehicles and

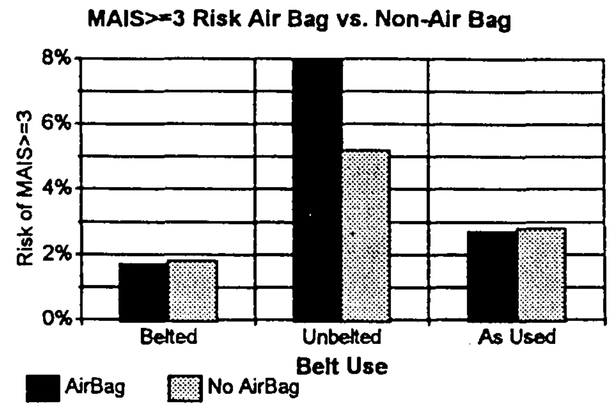

Figure 1. Serious Injury Risk by Restraint

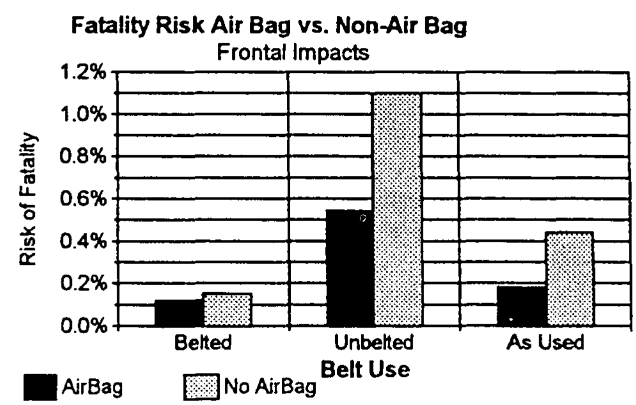

Figure 2. Fatality Risk by Restraint

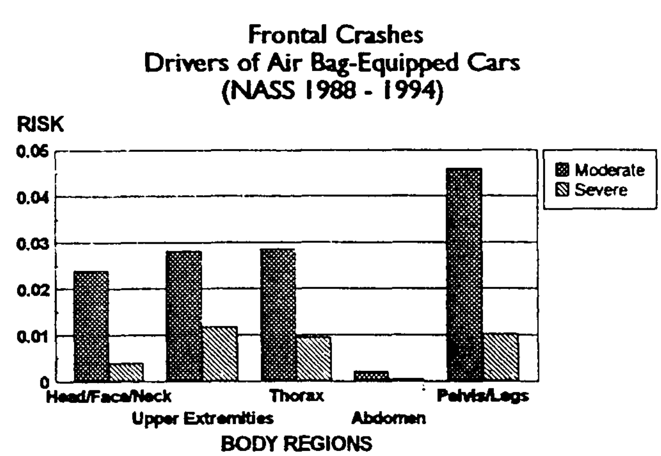

Figure 3. Injury Risk by Body Region

seated in the driver's position. In examining the data for the moderate injuries (i.e., AIS ≥ 2), it is seen that the pelvis/leg region has almost twice the risk as that for the other body regions. For the severe injuries (i.e., AIS ≥ 3), the highest risk is nearly equivalent among the upper extremities, the thorax, and the pelvis/leg regions.

Traditionally, fatality reduction has been the emphasis of the agency's research program. More recently, however,

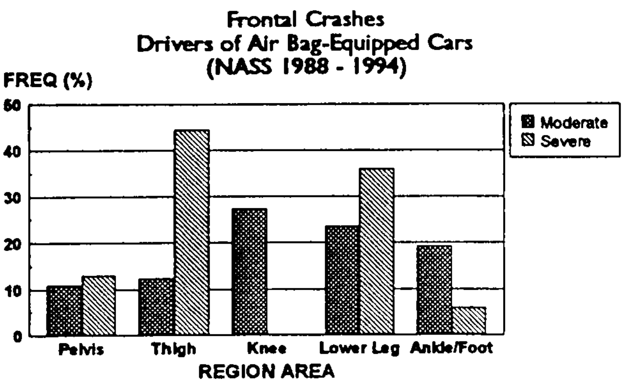

Figure 4. Injury Frequency for Leg Injuries

attention has been focused toward injury reduction, particularly for those injuries which lead to life long disabilities. This added focus includes the role of lower extremity and pelvic injuries in frontal crashes. Figure 4 provides more detail on the injuries to this body region. Particularly, the frequencies of the injuries for the pelvis, thigh, knee, lower leg, and ankle/foot are charted for both the moderate injuries and the severe injuries. As shown here, the knee and lower leg injures are the more frequently occurring moderate injuries. Also, the thigh and lower leg are the more frequently occurring severe injuries.

Selection of Test Conditions Based on Accident Conditions - An additional test procedure for increased frontal protection should simulate those crash modes in the accident environment which result in highest frequency and risk of injury/fatality. Since FMVSS No. 208 sets performance requirements for full frontal impacts, the initial analysis focused on "offset", frontal impacts as candidate accident modes for simulation. The accident analysis has been coupled with offset crash testing to determine which impact configurations produce the highest likelihood and frequency for injury/fatality. As discussed in the accompanying report on the crash testing program, offset car-to-car crash tests have been conducted with collinear and oblique impact directions at overlaps between 40 and about 80 percent and velocities between 60 and 66 kmph (each car for both cars moving.) Based on the testing to date it appears that the 30 degree oblique impact at about 65 percent overlap produces the highest injury measures on the 50th percentile Hybrid III test dummy.

Drivers of air bag vehicles were grouped by their general area of damage (GAD) and principal direction of force (DOF1) into a frontal impact population. Those drivers were considered to be in frontal impacts if their vehicle sustained DOF1 between 11 and 1 o'clock or DOF1 was 10 or 2 and GAD1 was front or side with damage forward of the A-pillar. In the 1988-1994 NASS-CDS there were 1099 frontal impacts with about 839 deployments. Only drivers were included in this analysis, since almost all occupants with air bags in NASS are drivers, thus counts of occupants and vehicles are the same. The frontal impact population is then separated into specific crash modes to identify potential impact configurations with high frequency and risk of injury to be simulated by crash test procedures. The frontal population was separated by direction of force (DOF) into collinear or oblique (left or right), by damage distribution into offset (left or right) or distributed, and by object contacted into another vehicle or fixed object. Counts in the text or figures distributed ("D"=0 or SHL1=D), left ("D"<0 or SHL1=Y or L) and right ("D">0 or SHL1=Z or R) offset impacts. For those impacts with left or right damage the GAD1 must include the front corner of the vehicle (SHL1=F,Y or D) and is entered as left or right 1/3 of the vehicle's front (equivalent to SHL1=L or R for GAD1=F)

Figure 5 shows frontal damaged vehicles by DOF (oblique and collinear) and object contacted (other vehicle or fixed object). The largest portion of occupants are in collinear, vehicle-to-vehicle impacts (36 percent.) About 80 percent of are weighted unless noted as "raw" counts. DOF is used to delineate collinear (12 o'clock), left (10 &11 o'clock) and right (1 & 2 o'clock) oblique impacts. Object contacted is separated into other vehicle and fixed object. For frontal damage (GAD1=F), overlap (or offset) is defined by the crash "D" variable when known and after 1989; otherwise, the primary specific horizontal location (SHL1) is used, and is separated into drivers with air bags in frontal damage accidents were in vehicle-to-vehicle collisions and the impact direction was almost equal between collinear (50 percent) and oblique (48 percent.)

Figure 6 further categorizes cars (or drivers) with air bags into DOF and damage distribution, combining vehicle-to-vehicle and vehicle-to-fixed object impacts. No single configuration of impact direction/damage distribution dominates with any single configuration contributing more than 20 percent.

Grouping Into Most Appropriate Test Procedure - The exposure population for frontal impacts, i.e., number of collisions, is based on 1994 NASS to estimate the exposure

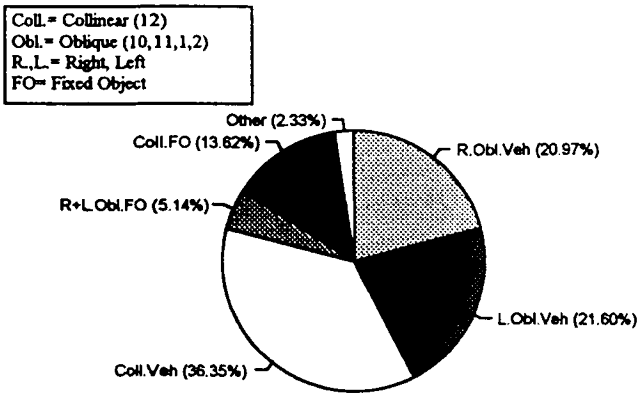

Figure 5. Frontals by DOF and Impact Partner

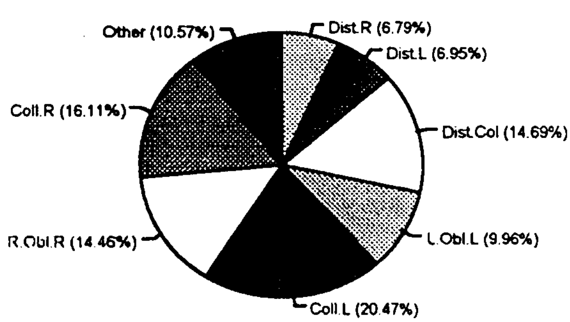

Figure 6. DOF and Damage Distribution

for a single year of an all air bag fleet. Figure 7 groups the various categories of occupants in "frontal" collisions (by DOF, damage distribution and crash partner) into which is deemed as the most appropriate test conditions of those current or envisioned for the future. The current frontal test is a full frontal impact into a fixed rigid barrier with impact angles on the car from -30 to +30 degrees. As shown in the figure, all collinear, distributed damage impacts and oblique, distributed damage, fixed object impacts with distributed damage are assumed to be best simulated by this test condition.

The left oblique, left offset configuration is the test condition being emphasized in the Improved Frontal Protection program. It is assumed that all left offset impacts, either collinear or oblique, are best represented by this test condition. Also, it is assumed that the left oblique, vehicle-to-vehicle impact, with distributed damage is better simulated by the left oblique, left offset test than by the barrier test. Not only may less than full overlaps often produce distributed damage, but the interaction of the vehicle and the propensity for higher intrusion is well simulated by this test even though the overlap may be less than full.

A right oblique, right offset configuration would include right side impacts in the same way as left side impacts are included in the left oblique, left offset test. About 9 percent of cars have offset, frontal damage which is opposite to the clock direction, i.e., left and right oblique impacts with right and left offset damage, respectively. Note that this would be the impact configuration for the "bullet" vehicle in a left or right oblique impact to the "target" vehicle, as shown in Figure 7.

Based on the assumed groupings of vehicle impact conditions from above, the left oblique, left offset test would represent about 36 percent of cars with air bags in "frontal" crashes, with right oblique, right offset also making up about 36 percent and full barrier about 15 percent.

Injury Risk by Test Configuration - Comparing injury risk shows that for MAIS$\geq$ 3, the "full barrier" groupings has the highest injury rate (8.6 percent.) Left oblique, left offset and right oblique, right offset groups both have serious injury rates of about 2.9 percent (Figure 8.)

Within the test groupings for left oblique, left overlap and right oblique, right offset the effect of overlap on injury rate was assessed. As a rough approximation of overlap percent, an average car width of 66 inches is assumed for "L" in the offset formula: Overlap = 1-(2*D/L). Overlap is then separated into 1/3 or less of the car width, over 1/3 to 2/3 of the width and over 2/3 of the width. As discussed above, left and right damaged vehicles with damage to the front corner were grouped into the 1/3 overlap category. By using these damage width groupings, the SHL1 parameter may be used when "D" is not known, which is separated into damage width increments of one-third of the vehicle width. The relatively low injury risk for configurations grouped under a left oblique, left offset test appears to be due to low occurrence of MAIS$\geq$3 injuries in narrow overlap impacts. For left oblique impacts the rate of MAIS$\geq$3 injuries is about 1.5% for 1/3 or less overlap and about 2 percent for right oblique impacts (Figure 9.) At overlaps over 2/3, left oblique impacts produce higher AIS$\geq$3 injury rates, increasing to about 5.5 percent for over 2/3 overlap

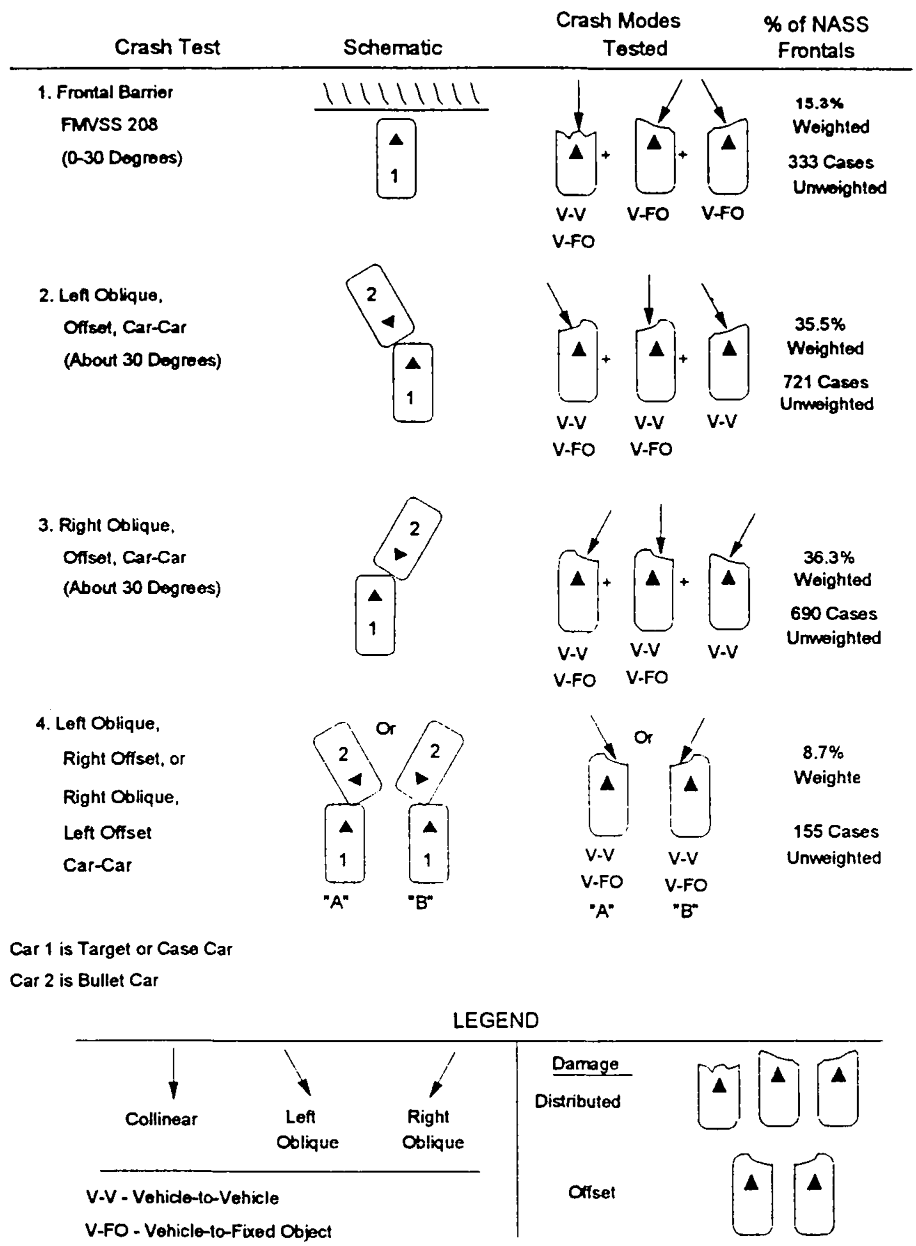

Figure 7 - Possible Tests and Crash Modes Simulated

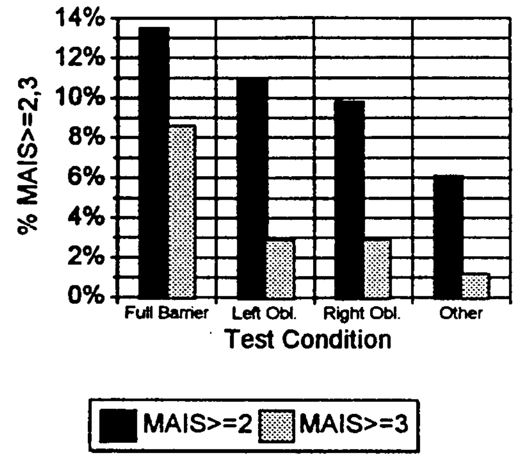

Figure 8. Injury Risk by Test Condition

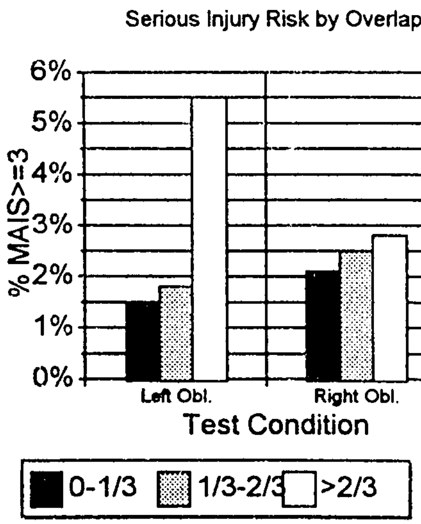

Figure 9. Injury Risk by Overlap

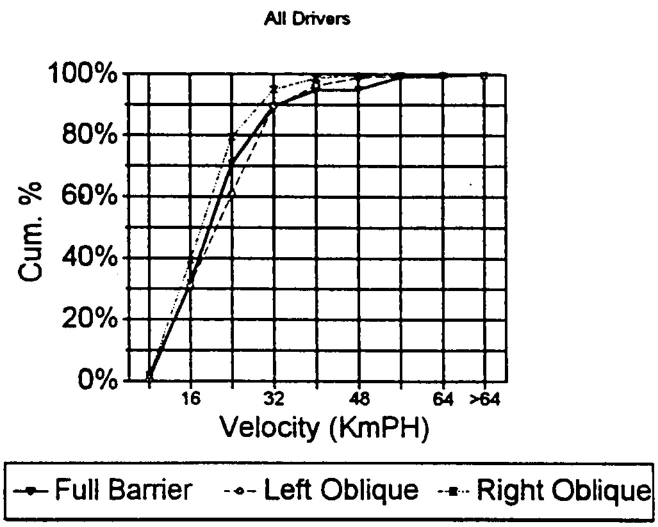

Figure 10. Crash Exposure by Cumulative Δv and Test Condition

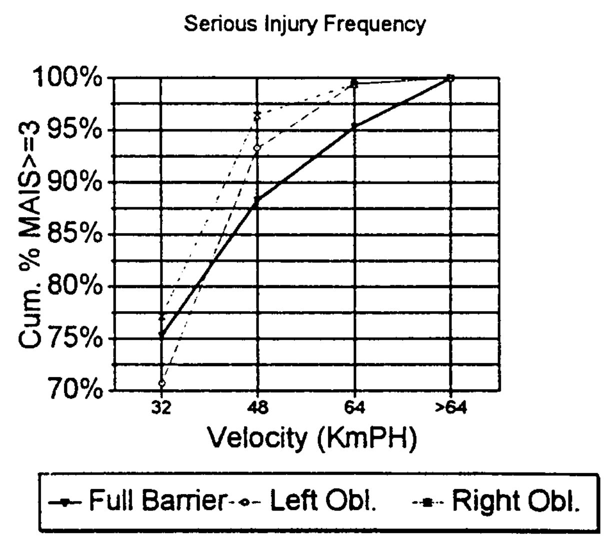

Figure 11. Crash Exposure by Cumulative Δv and Test Condition

(compared to less than 3 percent for right oblique.) The left oblique impact at over 2/3 overlap produces the highest AIS$\geq$3 injury rate of all offset impact modes considered. The higher MAIS$\geq$3 injuries for right oblique impacts with narrow overlaps, may be due to the method of quantifying overlap for non-frontal damage. As mentioned above, for side damage impacts with frontals DOF's, "frontal" overlap is not applicable and thus the overlap is categorized as 1/3 for purposes of quantification. These drivers of vehicles with right side damage are predominantly oblique impacts producing driver kinematics which may cause the driver to "slide off" the bag and impact hard surfaces when a passenger air bag is not present.

Crash Severity (Δv) Comparisons by Test Condition - As an indicator of crash severity to compare exposure for vehicles grouped by appropriate test condition, the total velocity change of the vehicle (Δv) is used. This is not a representative indicator of the overall crash severity exposure since Δv is known in less than 50 percent of the vehicles with driver air bags in the frontal impacts configurations which are being considered as potential tests. As shown in Figure 10 the Δv distributions are similar for all impacts regardless of test configuration grouping when considering all crash exposures. For MAIS$\geq$3 the oblique impact configurations have a larger percentage of injured drivers at lower Δv's than for the full frontal (Figure 11.) About 12 percent of serious injuries occur above 48 kmph Δv for full barrier type impacts, whereas for right and left oblique type impacts there are only about 4 to 7 percent, respectively, occurring above this. There is reason to believe that the magnitude of the Δv's computed in NASS tend to be low in less than full overlap frontal impacts. A limited study by NHTSA conducted a NASS type investigation of several staged car-to-car crash tests to compare the NASS investigators reconstruction of the accident to known impact conditions and crash severity. The Δv's computed by the NASS investigators were consistently lower than the measured Δv's in the staged crash tests.

Test Procedure - While the results of the aforementioned analyses are preliminary, several findings led to the recommendation that a test procedure be developed which simulates a moving car-to-car frontal offset crash [4]. First, the analyses indicated that impact conditions represented by offset/oblique tests, although producing lower risk of serious injury, are much more frequent and produce higher incidence of these injuries. Secondly, it was estimated that the target population of moderate to severe injuries affected by an offset requirement is much higher than that of a higher speed full barrier requirement. Finally, for occupants on the damaged side of the vehicle, intrusion is more likely producing a higher susceptibility for lower leg injuries in offset crashes than in full barrier crashes and, as a result, would receive more attention from an offset test requirement.

The test procedure was also addressed to higher percentage overlaps which showed the highest injury risk in the offset accidents. The higher overlap impacts tend to experience higher velocity change in the NASS accident data, possibly, because of a longer period of structural engagement and less rotation.

Fatal Accident Case Review -In addition to the agency's review of the NASS data, the agency's Fatal Accident Reporting System (FARS) file was examined for fatal crashes involving air bag equipped vehicles. This revealed 279 driver fatality cases in which photographs were available. In addition, another 22 cases (some with photographs) were identified in which the fatality was to a right front seated passenger of ages 15 years and under. To date, 21 of the cases involving the right front seat passengers have been reviewed. Particularly, the police accident report and photographs (if available) for each case were reviewed, and a determination was made by a team of experts regarding the probability that the air bag contributed to the fatality. Of these cases, 8 were deemed as probable, 6 as possible, 7 as none, and 2 as unknown (due to lack of necessary information). For the 8 cases in which the contribution was determined to be probable, 6 cases involved crashes of minor impact severity, and 2 cases involved crashes of moderate impact severity.

TEST SURROGATE

The Hybrid III 50th percentile male dummy, with extensive instrumentation and recognized methods of injury assessment, was selected for use for the bulk of the testing [5]. Additionally, the Hybrid III 5th percentile female and 95th percentile male dummies were selected for some tests to evaluate the different sized occupant kinematics and responses. During the testing, the body regions which have been instrumented (or will be instrumented in selected tests) for predicting injury include the head--linear and angular acceleration, the face--element for laceration potential, neck--upper and lower load cells, thorax--accelerations and deformation, abdomen--foam insert for deformation, pelvis--acceleration, femur--load cell, knee--sliding knee for shear, lower leg--shear and moment, and ankle--foot motion and load cell.

Finally, the Biomechanics Division has a research effort underway for developing an advanced thoracic element for the 50th percentile test dummy. A prototype thorax is available which incorporates a more realistic and human-like geometry and more comprehensive injury assessment instrumentation. In addition, we are developing improved lower extremities and an improved neck for the 50th percentile dummy [6]. As prototypes of these new components become available, these will be used in the frontal crash testing program.

CRASH TESTING

In order to evaluate higher severity crashes, the agency has initiated a frontal test program. These tests are being used to examine the effects that such parameters as collision direction and speed, vehicle size, structural aggressiveness, and geometric compatibility have on the occupant compartment intrusion and on occupant injury. Additionally, testing has been initiated to examine protection afforded to occupants of different sizes. These tests include car-to-car, moving barrier-to-car, car-to-pole, and car-to-stationary barrier crash tests. The review to date of the accident data indicates that a test procedure simulating car-to-car offset crashes provides the largest target population for addressing the injuries occurring in frontal crashes.

Offset frontal testing has been completed in which a lap and shoulder belt equipped Honda Accord was crashed into a series of driver air bag equipped cars including an Isuzu Stylus, a Volvo 740, a Geo Metro convertible, a Ford Taurus, a Dodge Dynasty, a Chevrolet Corsica, a Honda Accord station wagon, and a Saab 9000. Each of these tests was conducted with both cars moving at a nominally 116 kmph (72 mph) closing speed and with an overlap of 60 percent on the subject vehicle. In the tests, a Hybrid III 50th percentile male dummy was seated in the driver's position for both cars. All available restraints were utilized, i.e., both the air bag and safety belts were used. The maximum femur loading and maximum tibia moment are shown in Figures 13 and 14.

As shown in Figures 12 and 13, the Geo Metro is the only vehicle to exceed the FMVSS No. 208 injury criteria (i.e., the Head Injury Criterion was 1,699 and the maximum femur loading was 10,270 Newtons). However, it should be noted that during these tests the toe board intrusion often caused the femur to be oriented in a vertical direction, thereby resulting in the loads to be transmitted in a non-axial direction and not fully measured by the femoral load cells. This points out the need for improved instrumentation in this body area along with improved injury assessment capability. All of the vehicles exceeded the tibia moment that is considered to be injurious (225 Newton-meters [5]). The results are shown in Figure 14. Additional tests were conducted to determine the effects of changes in vehicle overlap and/or vehicle trajectory. For the new tests, the Chevrolet Corsica was selected as the subject vehicle and the Honda Accord remained as the bullet vehicle. Three tests have been conducted. Two tests were conducted at the nominally 116 kmph closing speed and with overlap of 50

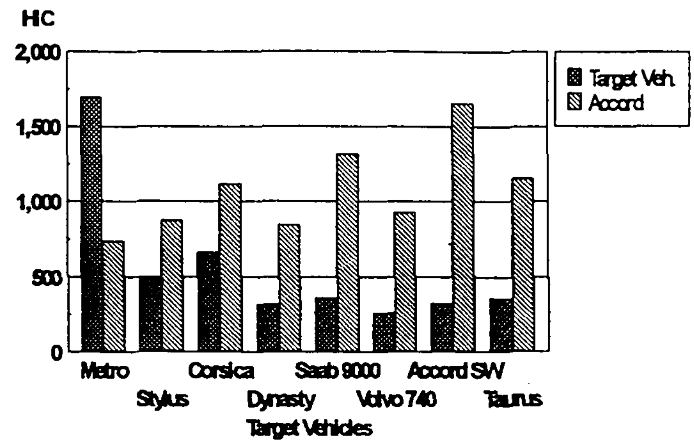

Figure 12. Offset Testing - Inline 60% Overlap

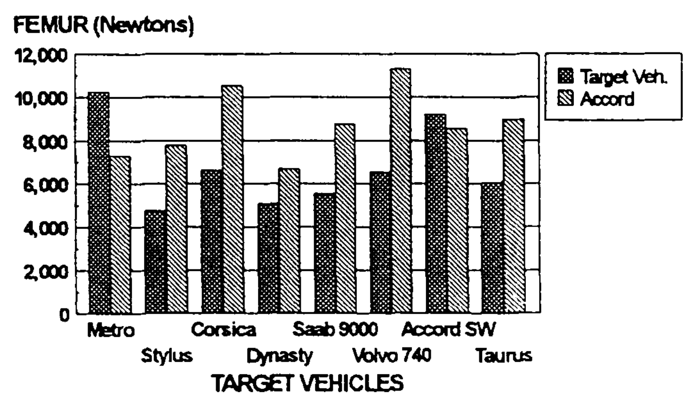

Figure 13. Offset Testing - Inline 60% Overlap

and 70 percent on the subject vehicle. An additional test was conducted to evaluate the effect of an oblique impact on the Chevrolet Corsica. In this test, the Honda Accord bullet vehicle was oriented at a 30 degree angle rather than the in-line orientation used for all of the previous testing. The test was set up so that there was 50 percent overlap of the subject vehicle during engagement

Again, the Hybrid III 50th percentile male dummy and all available restraints were used. The femur loads and tibia moments for these tests along with the corresponding 60 percent overlap, inline configuration test conducted for the aforementioned test series are shown in Figures 15 and 16.

As shown in Figure 15, the Chevrolet Corsica exceeded the FMVSS No. 208 femur injury criteria in only the oblique test. As can be seen, in this test the femur loading exceeded the requirement of 10,000 Newtons. However, as

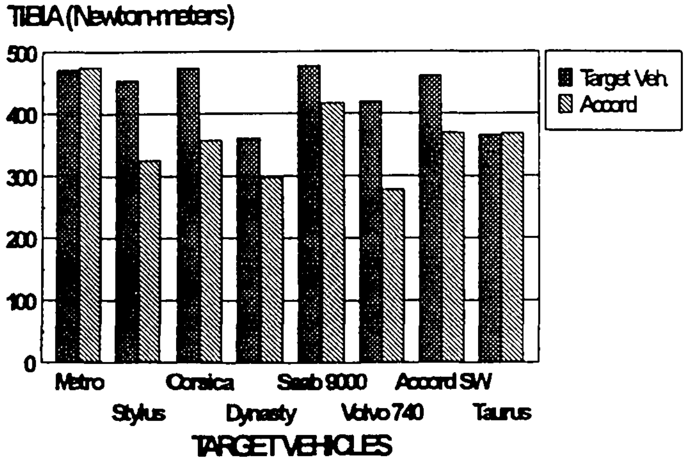

Figure 14. Offset Testing - Inline 60% Overlap

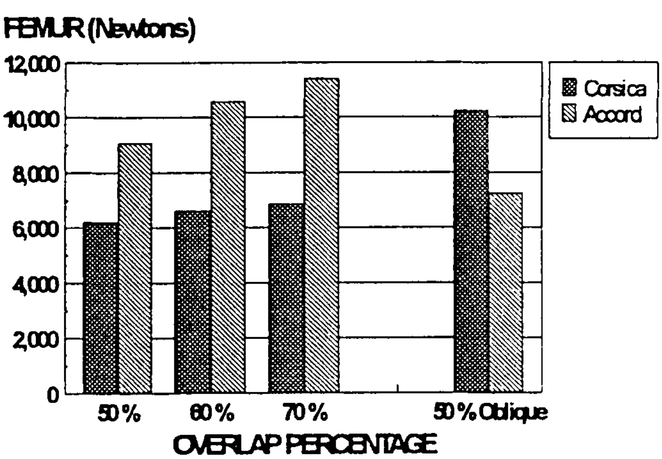

Figure 15. Offset Testing - Variation of Overlap Percentage and Impact Angle

observed in the 60 percent overlap test series, the tibia moments exceeded the 225 Newton-meter threshold in each of the tests (see Figure 16). Several trends are observed. The first is that the femur loads increase as overlap percentage increases. This dependence is due to the increased peak vehicle acceleration and shorter crash pulse duration as more of the vehicle structure is engaged with increasing overlap percentage. A second trend is that the tibia moment increases as overlap percentage decreases. This dependence is due to the increasing toe board intrusion as the overlap percentage is decreased. In addition to the frontal offset crash testing, full barrier testing also has been conducted.

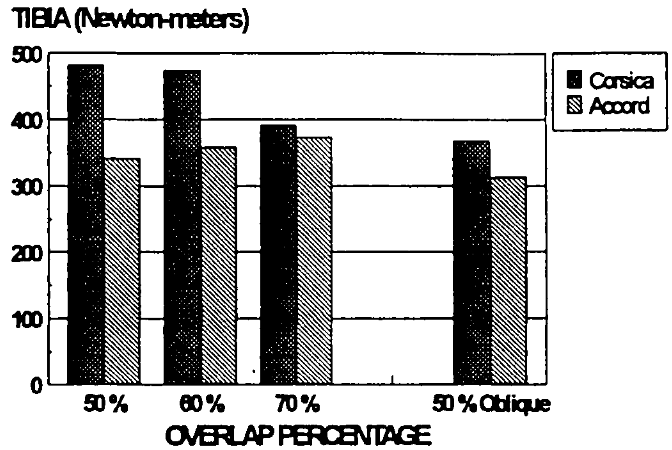

Figure 16. Offset Testing - Variation of Overlap Percentage and Impact Angle

Additional testing again used the Hybrid III 50th percentile male dummy and all available restraints for developing a crash test, based on accident data, that indicates the importance of oblique and offset crashes. For this effort, a finite element model was generated for the Ford Taurus. The Ford Taurus was then crash tested in a variety of configurations to validate the model and to assist in developing a test procedure which more closely mimics offset crashes as seen in NASS accident data. First the Taurus was tested in an inline configuration with a Taurus bullet vehicle. Each vehicle was moving at approximately 56 kmph. The driver dummy results are shown in Table 2. The driver dummy injury measures were within the FMVSS No. 208 requirements by a comfortable margin. Post test measurements also revealed minimal intrusion of approximately 114 mm at the left toepan. The next two Taurus tests were conducted for the purpose of comparing moving deformable barrier (MDB) results with the Taurus as the bullet vehicle in an oblique/offset configuration. The moving deformable barrier used a FMVSS No. 214 honeycomb deformable element. As seen from the dummy response, the results were very similar. Post-test intrusion measurements at the floorpan were almost identical at 298 mm and 305 mm, respectively, for the Taurus and the MDB tests. The next Taurus test was used to determine if a 45 degree configuration would be more or less severe than the 30 degree oblique test. This test was run to match the travel speeds in the 30 degree test by crabbing the moving barrier at 20 degrees to the track centerline and positioning the stationary car at 25 degrees to the track centerline. The barrier was then towed to 108.6 kmph. This configuration is equivalent to each vehicle moving at 58.8 kmph. As shown in Table 2, the dummy responses are significantly lower than previous results in the 30 degree configuration.

The last Taurus test was conducted according to the European WG11 test protocol except for the test speed which was 65 kmph. These results indicate that the test is very benign for both the dummy and the structure. For direct comparison to the above tests, the NCAP results are shown in the table. The dummy results from NCAP are very similar to the moving deformable barrier.

Since the deformable moving barrier oblique test appeared to present a fairly hostile test of the vehicle's structure and its occupant, this configuration was chosen to test previously tested car models. The basis of selection was the extent of collapse of the occupant compartment in this test series. The cars chosen were the Chevrolet Corsica and the Isuzu Stylus. The desired test speed was 61 kmph with a 1360 KG (nominal) moving deformable barrier. The actual test speeds were in excess of the target speed and are shown in Table 3. The results for the restrained drivers in these tests are also summarized in Table 3. As can be seen from the table, both restrained driver dummies exceeded the FMVSS No. 208 injury criteria. The Stylus driver exceeded the injury criteria in all three areas by a large margin. It was also observed from-post test observation of these crash test vehicles that intrusion of the occupant compartment was quite severe. The dummy was entrapped by the severely intruded floorpan and steering wheel. Extrication of the dummy required power tools. Another interesting phenomenon occurred during the test; the dummy face skin was peeled off of the driver dummy's head. It is unclear whether this occurred on rebound or during initial contact with the intruding "A" pillar. Given the excessive speed of these tests, it is uncertain whether or not a test in the 60 kmph range will produce FMVSS No. 208 failures, but given the severe nature of the intrusion and the excessive dummy readings it appears very unlikely that these cars could pass all injury criteria.

For comparison of tibia data, several New Car Assessment Program (NCAP) tests were conducted with lower leg instrumentation. The foot was rotated to allow measurement of lower leg moment about the y axis. In the other tests presented, the moment about the x axis was measured in the lower tibia. All future testing will be conducted with the newer orientation, that is measuring y-moments. As shown in Table 4, all dummies exceeded 225 newton-meters by a small margin. The average tibia moment was 273 newton-meters, whereas the offset tests had an average tibia moment of 436 newton-meters.

As mentioned above, in comparison to the offset car-to-car tests, the WG 11 type test with a fixed deformable barrier at similar offsets is a more benign test in terms of dummy injury measures and structural responses. Table 5 compares crash measures on the Ford Taurus from: 1) the NHTSA car-to-car collinear, 50 percent overlap test at 110 kmph closing velocity; 2) the NHTSA car-to-car, 30 degree oblique, 59 percent overlap test at 110 kmph closing velocity; 3) the NHTSA WG 11 type test with 50 percent overlap at 64 kmph; and 4) the IIHS WG 11 type test with 40 percent overlap at 64 kmph. Even though the subject car's crash energy is about 20 to 35 percent higher in the WG 11 type tests, all injury measures are lower in the WG 11 type tests, except for femur loads in the NHTSA test. The peak compartment G's are much higher in the oblique, car-to-car test and occur much earlier in the event for both collinear and oblique car-to-car tests (47 and 58 ms compared to 93 and 95 ms in the WG 11 type tests.) Although an exact analysis of intrusion has not been completed on these tests;.the initial assessment shows similar amounts of toe-board intrusion in both type tests. However, much more intrusion is observed in A-pillar and other higher surfaces in the car-to-car tests.

AIR BAG AGGRESSIVENESS

In monitoring the fleet performance of current air bag systems, NHTSA has identified aggressive air bag deployment as a potential cause of injuries and fatalities to occupants in various speed crashes. However, most troublesome are those injuries and fatalities in low speed impacts where there may not have been any injury were it not for the deploying air bag. On November 9, 1995, the agency published a request for comments in the Federal Register to summarize what NHTSA knows about these side effects of air bags and how it plans to minimize them in the future. In the notice, the agency identified several groups which may be at greater risk from the air bag deployment. These include unrestrained small statured and/or older people and persons with disabilities seated in the driver position, infants in rear-facing child restraints and unrestrained children in the right front seating position, and out-of-position occupants in both the driver and right front seating positions. The notice requested manufacturers, insurers, members of the medical community, and other interested members of the general public to share information about air bag designs or experience. For its part, the agency has added new activities to its frontal crash protection research program to address this issue. These include a number of tasks that provide for a general analysis of injuries\fatalities with air bags, including the analysis of fatalities to children under 15 with air bags and including the analysis of driver fatalities. Also, research has initiated laboratory testing using International Standards Organization test procedures ISO/DTR 10982 and ISO/DTR 14645 to evaluate aggressiveness of current air bags. In these tests, both production and "down-powered" inflator modules will be evaluated to determine the effects of inflation on injury measures. Also, research has been initiated to develop component level test procedures for assessing air bag aggressiveness. This activity is focused on evaluating injuries to the upper extremities.

DISCUSSION

While the analysis of the NASS data is still ongoing and the case reviews of FARS data are being initiated, the findings to date have led to the preliminary recommendation that a test procedure be developed which simulates a moving car-to-car frontal offset crash. Additional agency analysis needs to be completed before a final determination can be made. This condition differs substantially from the test procedures being developed in Europe or currently being used by the Insurance Institute for Highway Safety. In the latter cases, the test procedure utilizes a fixed deformable barrier. A limitation of their approach is that the test does not account for the effects of vehicle mass in a car-to-car crash. That is, the vehicle's change in velocity is similar for all vehicle sizes in the car-to-fixed deformable barrier crashes, whereas it is greater for subcompact and many compact vehicles when compared to the heavier vehicles in car-to-car crashes. Secondly, we observed that the time to peak acceleration was substantially longer in duration than that observed in the car-to-car tests. This is indicative of a "softer" crash pulse than that which would be experienced in a car-to-car crash.

From the agency's testing conducted to date, it is observed that occupants using a belt with an air bag are unlikely to get serious head or chest injuries without severe intrusion. However, lower extremity injuries often occur in offset crashes. This has been observed in both the accident data studies and in the crash test results. In our studies, the limitation in the number of air bag accident investigations have precluded a full quantitative description of the serious injury and fatality problems which will remain even after full implementation of air bags. Additional air bag accident data analysis and a full range of more severe frontal crash tests are underway.

At the conclusion of this research, it is expected that an additional new frontal crash test, test dummies, and injury criteria will be available for use in developing and evaluating improved restraint and structural concepts for improved frontal crash protection in air bag equipped vehicles.

ACKNOWLEDGMENT

The authors wish to acknowledge Ms. Marie C. Walz of the National Center for Statistics and Analysis for contributing information regarding the National Accident Sampling System and regarding projections of air bag implementation. The authors also wish to acknowledge Mr. Osvaldo Fessahaie of ISSI, Inc., for his technical support in analyzing the NASS data.

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TABLE 1 - Injury Risk by Restraint Condition, Frontal's, 1988-1995 (¾ Year File) NASS

RESTRAINT		MAIS 2-6[1]	MAIS 3-6	MAIS Fatal	Total
All Air Bag (Belted+Unbelted)	#	43774	14238	968	529201
	Row%	8.3%	2.7%	0.2%	
	Raw#	339	145	27	968
Non-Air Bag (Belted+Unbelted)	#	604909	175349	27440	6230850
	Row%	9.7%	2.8%	0.4%	
	Raw#	4457	1894	555	15676
Air Bag and Belts	#	28600	7511	515	445096
	Row%	6.4%	1.7%	0.12%	
	Raw#	197	77	12	1057
Belts	#	275612	75568	6636	4295759
	Row%	6.4%	1.8%	0.15%	
	Raw#	1717	636	126	8895
Air Bags and No Belts	#	15174	6727	454	051
	Row%	18.0%	8.0%	0.5%	
	Raw#	142	68	15	371
No Restraint	#	329297	99781	20804	1935091
	Row%	17.0%	5.2%	1.1%	
	Raw#	2740	1258	429	6781

[1] Includes MAIS≥2 Injured Drivers

TABLE 2. Offset Crash Test Results - Variation of Crash Conditions for FEM and Test Procedure Development

Taurus Test Condition	HIC	Chest G's	Femur (Newtons)	Tibia (Newton-meters)
Taurus-to-Taurus, Inline, 50 % overlap, 56 kmph	530	45.4	5654	184
Taurus-to-Taurus, 30 degree, 55 % overlap, 62 kmph	411	51	5824	242
MDB-to-Taurus, 30 degree, 53 % overlap, 57 kmph (each)	461	54.8	6708	493
MDB-to-Taurus, 45 degree crabbed, 65 % overlap, 105 kmph (MDB)	363	44.9	7223	338
Taurus-to-EEVC Fixed Deformable Barrier, 50 % overlap, 64.2 kmph	178	38.5	6154	142
Taurus, NCAP	524	53	7313	Not Collected

TABLE 3. Offset Crash Test Results - High Severity Crash Conditions for Test Procedure Development

Test Condition	HIC	Chest (G's)	Femur (Newtons)	Tibia (Newton-meters)
MDB-to-Corsica 60 % overlap, 30 degree, 66 kmph	1578	54.2	16277	474
MDB-to-Stylus 60 % overlap, 30 degree, 64 kmph	1243	100.1	23374	492

TABLE 4. Full Frontal NCAP Tibia Test Results (56 KMPH) - Driver Seating Position

Model	Test Weight (Kilograms)	Moment (Newton-Meters)
Altima	1493	252
Maxima	1561	256
Neon	1288	298
Millenia	1620	286

TABLE 5. Comparison of Crash Responses for Different Test Conditions

Conductor Test Type Closing Speed Angle Overlap	NHTSA Car-To-Car 112 kmph 0 Degree 50 Percent	NHTSA Car-To-Car 118 kmph 30 Degree Left 59 Percent	NHTSA WG 11 Type 64.7 kmph 0 Degree 50 Percent	IIHS WG 11 Type 64.4 kmph 0 Degree 40 Percent
Velocity Change	56 kmph	59 kmph	65 kmph	64 kmph
HIC (1000*)	530	411	178	305
Chest G's (60*)	45	51	39	34
Chest Defl. (76 mm*)	NA	25	NA	23
Max. Femur (10,000N*)	5,654 N	5,824 N	6,154 N	3,800 N
Max. Tibia (225N-M*)	184 N-M	242 N-M	142 N-M	72
Peak Compartment G's	30	47	34	36
Time to Peak G's	58 ms	47 ms	93 ms	95 ms

AUSTRALIAN NCAP PROGRAM REVIEWED – A COMPARISON OF THE NCAP PERFORMANCE OF 1995 AUSTRALIAN & US VEHICLES

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Michael Griffiths
Roads & Traffic Authority of NSW
Australia
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Paper Number 96-S1-O-12

ABSTRACT

Both Australia and the United States of America conduct consumer crash testing programs to assess vehicle crash performance. A 1993 analysis showed that the performance of Australian vehicles was inferior to that of equivalent US vehicles.

This paper reviews the relative crash performance of 1995 model Australian and US vehicles. Full frontal crash data was compared for light, compact, medium weight vehicles, four wheel drive vehicles, and light utilities (pickups). Offset crash data was compared for medium and four wheel drive vehicles.

Analysis shows that Australian occupants are still offered inferior crash protection compared to equivalent US model vehicles. The safety equipment fitted, which may contribute to these results are compared.

INTRODUCTION

Both Australia and North America conduct consumer crash testing programs to assess vehicle crash performance. A 1993 analysis of full frontal crash test data[1] showed that the performance of Australian 1992 year model vehicles was "approximately equal to that of the US fleet in the late 1970's". This paper analyses the relative crash performance of 1995 year model Australian and US vehicles to assess how the relative performance has changed.

METHOD

Full frontal crash test data for all 1995 vehicle categories was obtained from National Highway Traffic Safety Administration (NHTSA)[2]. Offset data for 1995 medium and four wheel drive (midsize utility) vehicles was obtained from the Insurance Institute for Highway Safety (IIHS)[3]. Australian crash test data was available for all vehicle categories in full frontal and offset configurations.

Australian and IIHS vehicle data was coded into vehicle categories, according to the NHTSA weight guidelines, in order to allow direct comparisons between vehicle categories. US 'luxury' vehicles (eg Mercedes, SAAB, etc) were omitted from the analysis on the basis that these vehicles had not been tested in Australia as they form part of the 'luxury' market. Inclusion of these vehicles in the US data may have unfairly biased the analysis. Additionally, only US data for utility (pickup) vehicles within a similar weight range as those tested in Australia (1245 – 1351 kg) was analysed, rather than the entire weight range of US utilities (pickups), to ensure comparability of results.

Australian full frontal and offset tests and IIHS offset tests used Hybrid-III dummies, whilst NHTSA full frontal tests used H-II and H-III dummies. This may have had some small effect on the data analysis[4].

Australian and NHTSA full frontal tests were conducted at a test speed of 56 km/h. IIHS offset tests were conducted at 64 km/h. With the exception of all utility vehicles, and one compact and one medium size vehicle (which were conducted at 64 km/h), all Australian offset tests were conducted at 60 km/h. Offset tests conducted at 64 km/h have a 15% increase in crash energy over 60 km/h tests. For this reason, Australian vehicles tested in the offset configuration at 60 km/h would be expected to have better crash performance than an identical vehicle tested at 64 km/h. Therefore, any comparisons drawn between US and Australian offset crash tests conducted at different speeds are conservative and favour the Australian vehicles.

Vehicle crash performance was assessed in terms of probability of life threatening head injury, probability of life threatening chest injury, and the combined probability of life threatening head and chest injury for full frontal and also for offset crashes. An injury risk value was also calculated combining these probabilities from offset and full frontal crashes[5,6].

Results of individual vehicles, where the same model was sold in both Australia and the US, were compared.

RESULTS

Full Frontal Crashes

Table 1 shows the probability of sustaining various life threatening injuries for drivers and passengers in full frontal crashes. It can be seen from the data that in full

Table 1
Full Frontal Crashes
Probability of Life Threatening Injury

Vehicle Category	Driver						Passenger					
	Head		Chest		Head & Chest		Head		Chest		Head & Chest	
	Aust	US	Aust	US	Aust	US	Aust	US	Aust	US	Aust	US
Mini / Light (1500 - 2499 lbs)	0.44	0.11	0.19	0.13	0.55	0.23	0.28	0.08	0.11	0.12	0.35	0.18
Compact (2500 - 2999 lbs)	0.29	0.06	0.21	0.12	0.44	0.18	0.34	0.12	0.14	0.12	0.41	0.22
Medium (3000 - 3499 lbs)	0.27	0.05	0.2	0.1	0.42	0.15	0.41	0.06	0.12	0.1	0.5	0.15
4 Wheel Drives	0.55	0.17	0.21	0.14	0.67	0.28	0.44	0.14	0.15	0.13	0.5	0.25
Pick Ups	0.26	0.17	0.17	0.13	0.39	0.28	0.22	0.13	0.09	0.10	0.29	0.21

frontal crashes, Australian drivers and passengers are at substantially greater risk of life threatening head and/or chest injury than their US counterparts. For example, in the Medium vehicle category, Australian Drivers and passengers have a risk of head injury much greater than that of their US counterparts. The overall risk of life threatening head and chest injury is greatest for Australian and US drivers in mini/light vehicles and four wheel drives. The ute (pickup) category has the least disparity with the US results.

While in every vehicle category, head injury risk was significantly greater for Australian front seat occupants compared to US occupants, the increased risk of life threatening chest injury was not as great.

The use of Hybrid II dummies in the US full frontal tests may give an advantage to the US manufacturers, as manufacturers are generally believed to choose the dummy which gives them the most favourable results.

Offset Crashes

Data from comparisons of offset crashes is shown in Table 2. The analysis is more limited, due to availability of less US offset data. The US data in Table 2 is based on the results of 4 compact vehicles, 6 medium vehicles, and 5 four wheel drive (MPV) vehicles. The US group of compact vehicles for which offset data was available had a mean crash performance the same as that of the whole US group for the full frontal test, indicating that the small sample size would most likely not bias the data.

For the medium group, the subset group with available offset data had better full frontal driver crash performance that the whole US group (9% compared to 15% probability of life threatening head and chest injury). Therefore, the US offset medium category subset shown may have better than average crash performance than the whole US medium category. However, the US vehicles were tested at higher speed for the offset crash test, suggesting that results may be artificially higher than the Australian results. This indicates that care should be taken when comparing the results of the Medium category comparison shown in Table 2.

For the four wheel drive category, the smaller range of US vehicles for which data was available, had frontal crash performance directly comparable to that of the whole US four wheel drive group, indicating that the smaller sample size was unlikely to bias results.

As can be seen from the data, front seat occupants in US vehicles are at considerably less risk than those in Australian vehicles for the compact and four wheel drive vehicle categories. This is probably also the case for the medium vehicle category. The difference between the results become even greater, when it is considered that all the US offset tests were conducted at 64 km/h, and involved 15% more crash energy. Both Australian and US offset tests used Hybrid III dummies only.

Once again the disparity between Australian and US chest injury risk results is far less than for head injury risk.

Combined Full Frontal and Offset Crashes

Table 3 presents the results of combined full frontal and offset crash data for Australian and US vehicles. The analysis is limited only to vehicles for which offset results were available. Therefore sample sizes for the US data are: 4 compact vehicles, 6 medium vehicles, and 5 four wheel

Table 2
Offset Crashes
Driver Probability of Life Threatening Injury

Vehicle Category	Head		Chest		Head & Chest	
	Aust	US	Aust	US	Aust	US
Compact (2500 - 2999 lbs)	0.17	0.03	0.09	0.06	0.24	0.09
Medium (3000 - 3499 lbs)	0.07	0.02	0.08	0.05	0.15	0.07
4 Wheel Drives	0.11	0.08	0.11	0.08	0.21	0.15

drive (MPV) vehicles. The limitations of interpretation of the data are discussed in "Offset Crashes".

For the driver, combined risk value is weighted in favour of the full frontal crash, according to the formula:

$$Preal = 0.59 \times Pcomb\ (full) + 0.41 \times Pcomb\ (offset)$$ [6]

The data shows that drivers of Australian vehicles are at greater risk of life threatening head and chest injury in frontal crashes than their US counterparts in similar vehicles. While there are limitations of the data in the medium vehicle class, the results are clear in the compact and four wheel drive classes. It would appear that drivers of Australian 1995 model vehicles have in the order of twice the risk of life threatening injury compared to drivers of US 1995 model vehicles.

Table 3
Driver Combined Probability of Life Threatening Injury
Frontal & Offset Crashes

Vehicle Category	Aust	US
Compact (2500 - 2999 lbs)	0.36	0.13
Medium (3000 - 3499 lbs)	0.31	0.11
4 Wheel Drives	0.48	0.2

Airbags

Significant differences have been noted between the crash performance of Australian and US vehicles. A possible reason for this is the presence of airbags in the vehicles. A comparison of the rate of airbag fitment in 1995 NCAP tested vehicles is presented in Table 4. Claims are not made about the effectiveness or otherwise of these airbags. The presence of the airbags is merely noted.

As can be seen from the data in Table 4, the rate of airbag fitment in Australian vehicles is significantly lower than that of the US fleet for the 1995 model year. There are no exacting mathematical trends relating the percentage rates of fitment of airbags for vehicle categories, and the relative injury risk for front seat occupants in these vehicle categories. However, a general trend is that US vehicle categories, with significantly higher rates of airbag fitment than the Australian equivalent category, also have significantly lower risks of life threatening head injury. For the category which has the most closely aligned rate of airbag fitment between the two countries, utilities (pickups), the crash results are similar. This would appear to support the argument, that the increased presence of airbags, is one of the contributing factors to the improved crash performance of US vehicles.

Table 4
Percentage of Vehicles Fitted with Airbags

Vehicle Category	% with Driver Airbag		% with Passenger Airbag	
	Aust	US	Aust	US
Mini / Light (1500 - 2499 lbs)	11	80	0	80
Compact (2500 - 2999 lbs)	60	100	0	83
Medium (3000 - 3499 lbs)	0	100	0	93
4 Wheel Drives	0	58	0	17
Pick Ups	0	33	0	0

Table 5
Individual Full Frontal Crashes – Driver and Passenger Injury Data

				HIC		Chest g (3 ms clip)		Injury Risk	
Make	**Model**	**Airbags**	**curb weight (lbs)**	**Driver**	**Passenger**	**Driver**	**Passenger**	**Driver**	**Passenger**
Mini Vehicle 1	**Australia**	–	1916	1645	1190	50.2	51.8	72%	39%
	US	D & P	1986	467	186	52	55	15%	16%
Light Vehicle 1	**Australia**	D	2299	1160	1170	74	53	56%	38%
	US	D & P	2317	867	684	61	57	31%	22%
Light Vehicle 2	**Australia**	–	2319	1165	1296	60.9	44.9	43%	43%
	US	D & P	2553	384	433	54	49	16%	13%
Light Vehicle 3	**Australia**	D	2650	960	2030	78	66	55%	92%
	US	D	2605	575	1602	54	56	18%	70%
Compact Vehicle 1	**Australia**	D & P	2657	710	920	53	48	20%	23%
	US	D & P	2654	482	532	46	51	12%	15%
Compact Vehicle 2	**Australia**	D	2914	873	903	50.8	43.8	23%	20%
	US	D & P	3020	471	357	43	58	10%	20%
Medium Vehicle 1	**Australia**		3031	1363	significant head to knee strike	44	43	48%	ND
	US	D & P	3128	607	881	51	50	16%	22%

Table 6
Individual Offset Crashes – Driver Injury Data

Make	Model	Airbags	curb weight (lbs)	HIC	Chest g	Injury Risk
Compact Vehicle 1	Australia	D & P	2657	400	41.6	9%
	US	D & P	2654	352	36	7%
Compact Vehicle 2	Australia	D	2914	799.2	42	16%
	US	D & P	3020	409	34	7%
Medium Vehicle 1	Australia	-	3031	834.6	39.7	16%
	US	D & P	3128	427	36	7%

Table 7
Individual Frontal & Offset Crashes Combined – Driver Injury Data

Make	Model	Airbags	curb weight (lbs)	Injury Risk
Compact Vehicle 1	Australia	D & P	2657	15%
	US	D & P	2654	10%
Compact Vehicle 2	Australia	D	2914	20%
	US	D & P	3020	9%
Medium Vehicle 1	Australia	-	3031	35%
	US	D & P	3128	13%

Vehicle Crash Comparisons Model to Model

This section compares crash performance of specific vehicles marketed in both Australia and the US. Crash performance data from these vehicle models is shown in Table 5, 6 and 7.

It can be seen from the data that the individual Australian vehicles did not perform as well in the crash tests as their US counterparts. Differences in specification, such as the presence of airbags, webbing clamps, pretensioners, low elongation webbing and 5 mph bumpers may be a factor in these results.

For the full frontal test only, the use of Hybrid II dummies in some tests may have favoured the US vehicles where head contacts did not occur. Nevertheless, while the use of Hybrid II dummies could account for HIC's up to 50% higher[4], this is not sufficient to explain differences of up to four-fold seen in the data.

Compact Vehicle 1 is known to have similar specifications in both Australia and the US, with the exception that the Australian vehicle does not feature 5 mph bumpers, and does not claim compliance with the US side impact test.

The similar test results appear to concur with the similarity in vehicle specification. The suggestion may be that Australian vehicles have different safety features to their US counterparts.

Conclusions

A comparison of 1995 Australian and US vehicle crash test data indicates that front seat occupants of Australian vehicles are at considerably higher risk of life threatening injury than front seat occupants of equivalent US vehicles. Far fewer Australian vehicles tested were fitted with airbags, when compared to similar US vehicles. The findings of this report indicate that, while there has been a significant improvement in some vehicles injury risk outcomes, since the 1993 findings of James Hackney[1], occupants of Australian vehicles are still at significantly greater risk in frontal crashes.

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THE CORRELATION BETWEEN TEST AND REAL-LIFE ACCIDENTS FOR THE CAR-TO-CAR FRONTAL CRASH

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ABSTRACT

Test and rating methods copying as closely as possible the real accident process are required in order to get reliable information on the safety performance.

For frontal impact tests this means that today the preferred impact obstacle is the deformable barrier and not the rigid wall. Comparison tests show that the barrier causes more realistic deformation of the vehicle than the rigid wall. Reliable data on vehicle compatibility is however still not available and based on the results it is demonstrated how the barrier could be developed in order to also get information on this passive safety aspect. Crash test results are evaluated with a rating procedure in order to determine the injury risk for passengers.

To prove the validity of the methods used results are compared with real life accidents on a type-specific basis.

INTRODUCTION

The collision type "frontal car-to-car impact" is still the most common accident type causing serious injuries [1]. Although the whole heterogeneity of this collision type cannot be dealt with in just one laboratory test there is wide mutual consent that the axial impact with a part-overlap is the most important of all possible configurations.

With this so-called offset test in the future the "opposing" vehicle will no longer be simulated by the rigid wall but by the more "real life" deformable barrier. A first final version of this barrier has been defined by the EEVC Working Group 11 [2]. For a test institution conducting crash test above all with the aim to give the consumer the most realistic information on the safety features of the vehicles on the market, the question is, to which extent the new method can be used for current tests. Another issue to be considered is which rating method is the most appropriate for the conversion of the gained test results into safety assessments which can be understood by the consumer.
In order to answer these questions a series of tests and studies of real accident processes have been conducted which will be presented and described below.

TEST METHOD

Requirements

The test should simulate in the most realistic manner the impact of the test vehicle on a similarly heavy vehicle. The test speed to be selected should result in an accident severity of approximately 50 km/h. Experience from accident research shows that markedly higher test speeds will considerably impair the compatibility especially of bigger vehicles and therefore have a negative influence on the overall accident process [1].

Realistic means for instance that front structure and compartment deformation and intrusion as well as dummy loadings and vehicle decelaration should be largely identical with a respective car-to-car crash. The same applies to airbag performance and in particular to its start of firing.

As a compromise between highest accident rate and worst case the overlap ratio should match an offset of the two vehicles of 50%.

The test must furnish data on aggressiveness and compatibility. Many experts believe that only this objective would justify the higher costs of the deformable barrier as opposed to the rigid wall.

Finally, for the reliability of results an adequate reproducibility will be mandatory.

EEVC barrier

In order to find out to which extent the described requirements are met for tests with the now completely developed EEVC barrier [2 and 3] and to get information on how much the results differ compared to the previous rigid wall tests the following comparison tests have been carried out [4]:

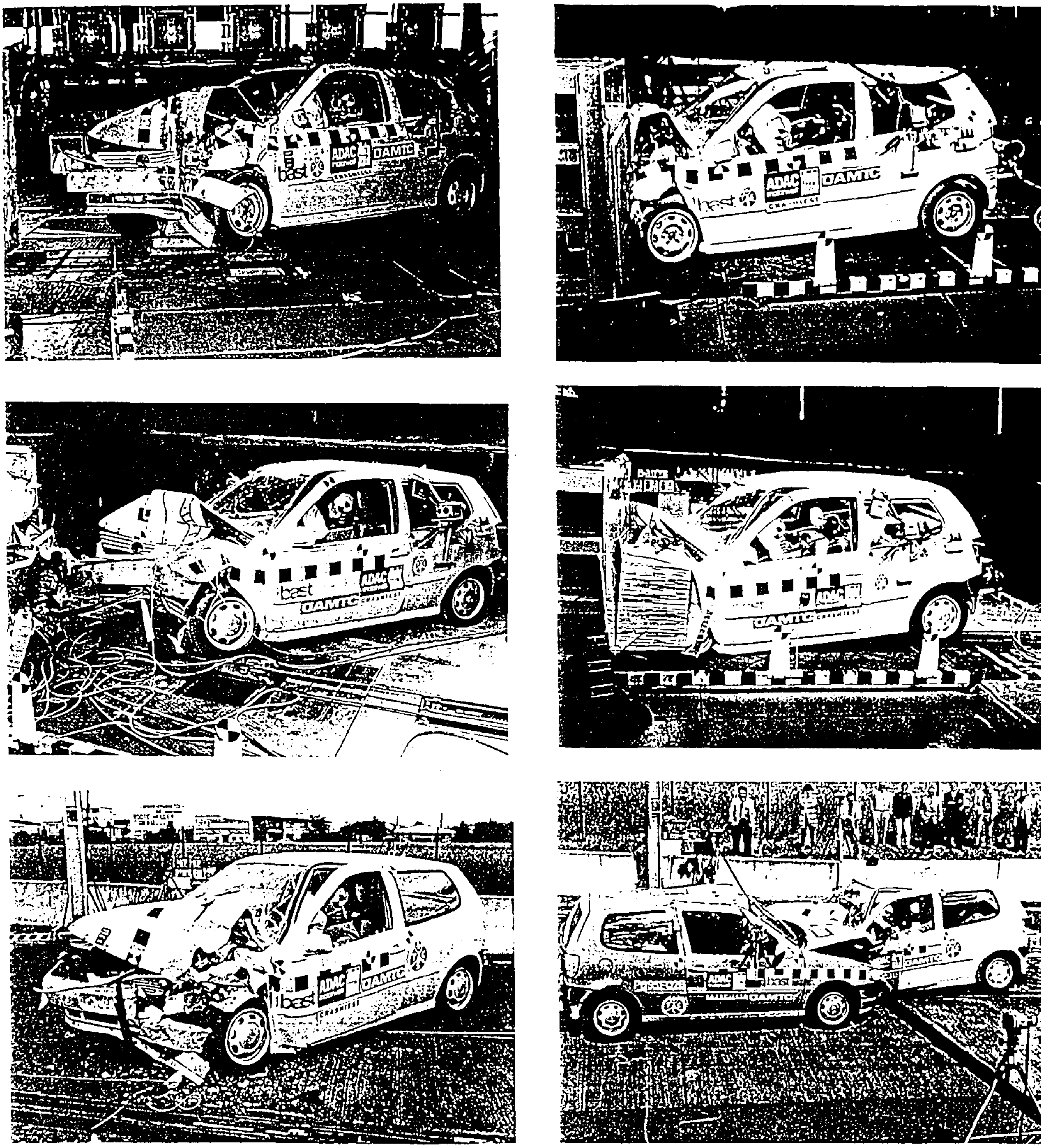

Figure 1. In car to car crash VW Polo shows significantly stronger and more heterogeneous deformation than in car to EEVC barrier and car to rigid wall test. But EEVC barrier does not show reproducebility problems

1. VW Polo against VW Polo, test speed both 50 km/h, approx. 50 % overlap and small height offset in such manner that the longitudinals of the two vehicles do not meet directly (reference test).

2. VW Polo against EEVC barrier, test speed 60 km/h, overlap 40%. This configuration was repeated in order to get data on the reproducibility of the tests.

3. VW Polo against the rigid wall, test speed 50 km/h, overlap 40%.

Vehicle deformations caused by the crash can be recognized in Fig. 1. In the car-to-car crash the front structure is torn, with a marked protrusion of the longitudinals. The side wall and roof area of the compartment is severely deformed and the footwell volume reduced considerably.

The barrier test deformation of the Polo compartment is already less signifant. The longitidinals cut and penetrate the barrier (bottoming out) and deform on the rigid wall behind the barrier. The footwell is also less deformed than with the car-to-car crash. The test repetition against the barrier produces practically identical results so that it can be said that this test series does not indicate bad reproducibility.

At the rigid wall front strukture deformation is uniformly distributed regardless of its rigidity. Compartment deformation and footwell reduction are even less significant than with the barrier test.

Fig. 2 shows the decelaration pulses of the test series. The differences between the car-to-car and the vehicle-to-wall crash are rather insignificant. The car/barrier pulse however is considerably lower over a long period, even fails presumably as a result of the cutting effect. It then rises again and is delayed by about 20%. It seems that the Polo experiences first a test against the barrier and then a test against the rigid wall. Another problem is that the airbag activation at the barrier is delayed by 10ms compared to the car-to-car crash. A trigger signal adapted to this barrier test could in real life cause airbag activation also in the case of minor accidents.

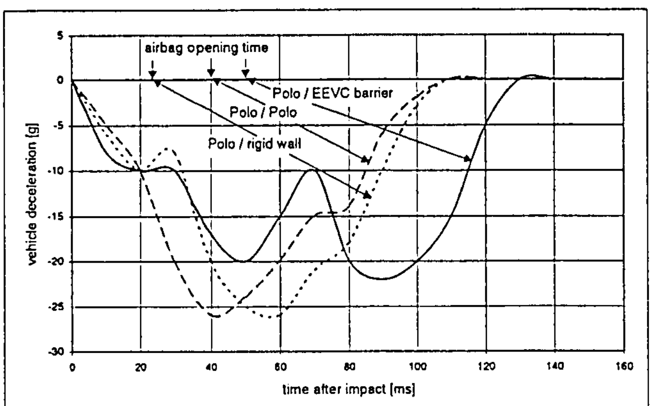

Figure 2. Car to car and car to rigid wall pulse show tolerable correspondence, car to EEVC barrier pulse is 15 % lower and 20 % longer

Fig. 3 confirms that the longitudinal of the Polo makes the barrier explode. This is obviously an indicator of increased aggressiveness. Due to the cutting a further quantification like an energy analysis is not possible. This means that also the question of whether 60 km/h is the energy-equivalent test speed compared to the 50 km/h car-to-car crash remains open.

Summary: The test series has in any case again confirmed that the deformable barrier system is the correct way for frontal structure tests and will provide reproducible results. The results shown and further findings by other institutions reveal, however, that specific improvements are required in order to ensure feasibility for all passenger car weight classes and designs. Essential requirements are:

1. Improvement of impact resistance and an increased barrier depth to prevent a bottoming out.

2. Increase of energy absorption capacity to prevent blocking out also for heavy vehicles.

3. To ensure a defined deformation in order to allow for a quantification of the absorbed energy and thus for the determination of the correct test speed as well as the aggressiveness and compatibility.

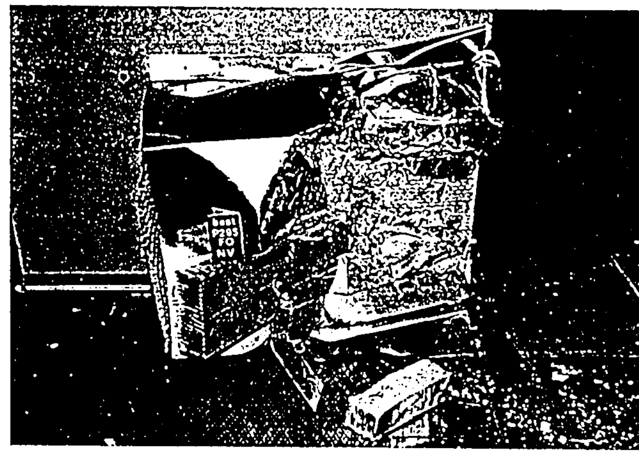

Figure 3. EECV barrier explodes during impact with VW Polo and needs better cutting resistance

ADAC barrier

The three-stage barrier displayed in Fig 4 was developed to meet the described requirements.
It is based to the largest extent possible on the EEVC barrier. The bumper has the same pressure rigidity and the same dimensions. The first stage has the same rigidity as the main body of the EEVC barrier, the second stage double and the third stage three times the rigidity. In order

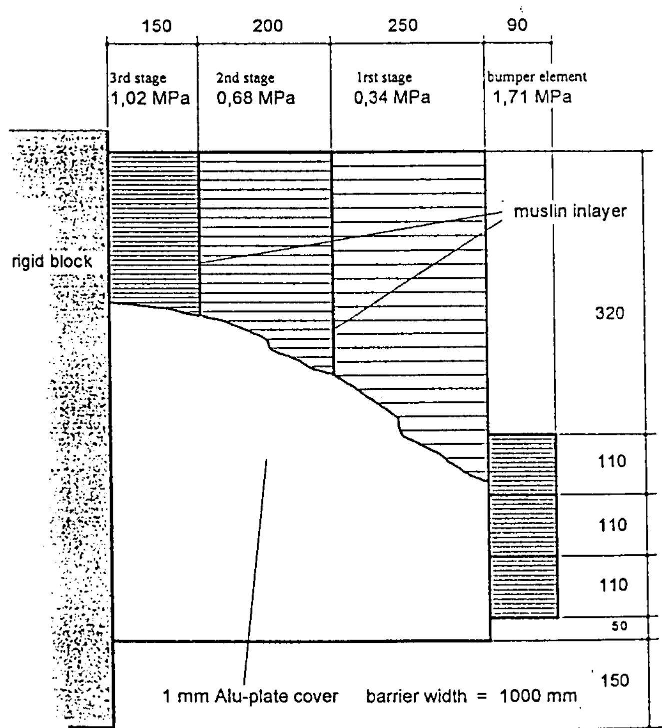

Figure 4. Three stages barrier with increased energy absorption capacity and better cutting resistance

to markedly reduce the cutting risk the bonding (adhesive) parts between the stages are reinforced with muslin. In

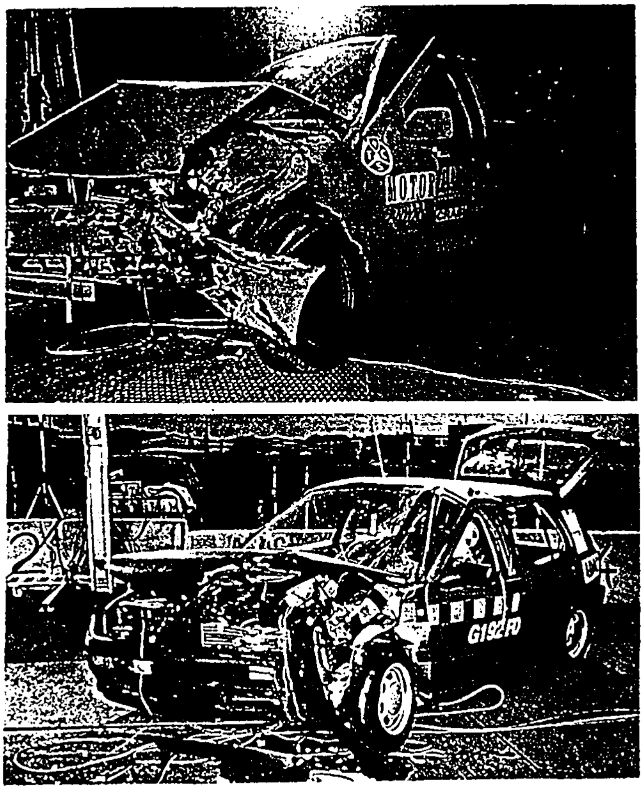

Figure 5. Golf to ADAC barrier (above) shows correspondence to real accident structure deformation (below)

addition to all other surfaces also the side checks are covered by aluminium sheet.

The barrier was tested in the framework of the current test project "Frontal crash performance of the VW Golf and its competitors" [5] with the models Nissan Almera, Renault, Megane, Fiat Brava, VW Golf.
The tests were again performed with a test speed of 60 km/h and 40% overlap. The car-to-car crash VW Golf against Opel Astra (50 km/h, overlap 50%) was available as reference test.

Fig 5 shows the Golf after the crash against the ADAC barrier and the Astra. Car deformations including the compartment and the footwell are on the whole far more consistent than was the case with the Polo test series. The different deformation performance of the engine hood is due to a design modification after the Golf/Astra test.

Also the decelaration pulses are far more similar (Fig 6) both with regard to level and duration. For the airbag activation time no comparitive figure is available from the car-to-car test. However, the activation time of 40 ms after impact is identical with the figure of the Polo against Polo test.

In order to get an energy balance also the rebound performance of the test vehicles was studied in detail on the basis of the high speed films (Fig 7 and 8). This also shows a good consistency between car-to-car and car/barrier.

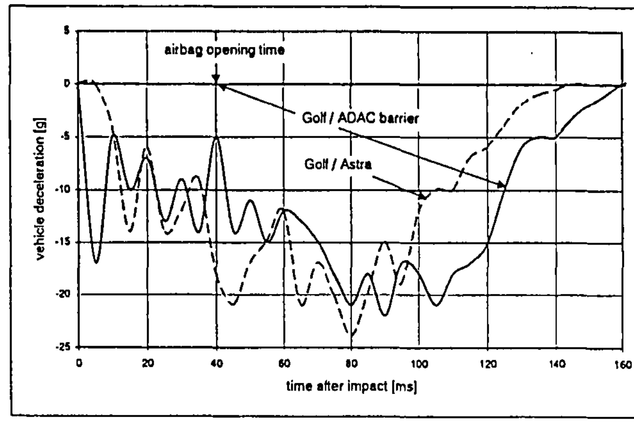

Figure 6. Car to Car and car to ADAC barrier pulse show better correspondence

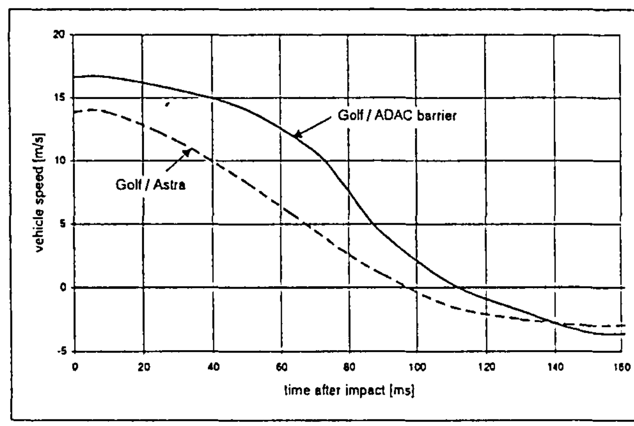

Figure 7 . Car to Car and car to ADAC barrier speed pattern show tolerable correspondence

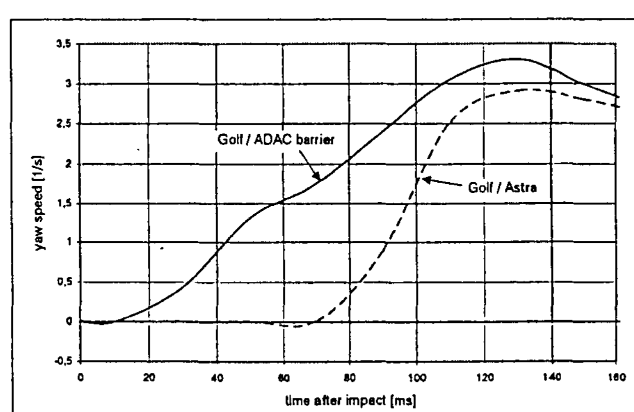

Figure 8. Car to Car and car to ADAC barrier yaw speed pattern show tolerable correspondence

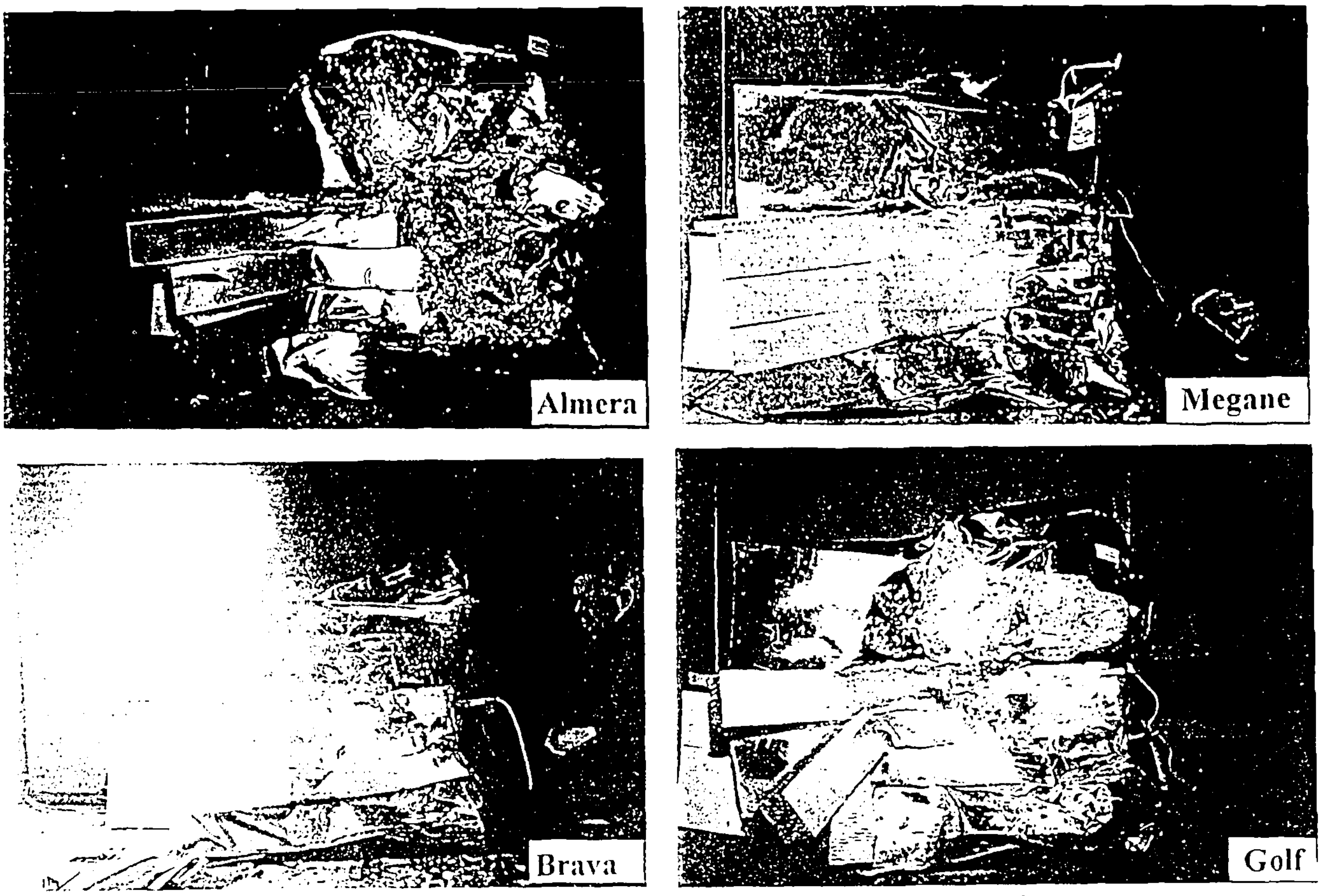

Figure 9. The 4 test vehicles produce quite different barrier deformation, but no bottoming out occurs

0	0	26	52	52	549	627	209	209
0	0	52	105	105	340	549	523	523
0	0	26	78	78	78	470	836	836

total absorbed energy bumper 6322 Nm

0	0	13	125	548	866	828	833	871
0	0	19	180	557	752	753	833	871
0	0	113	375	658	757	796	871	871
0	0	93	415	758	871	871	871	871
0	0	194	543	785	871	871	871	871
0	0	350	770	855	871	871	871	871
0	0	213	535	632	611	728	862	871
10	35	141	301	320	268	399	629	753

total absorbed energy 1rst stage 35013 Nm

0	0	0	0	287	391	157	291	865
0	0	0	0	108	108	351	785	1066
0	0	0	0	78	163	787	1405	1405
0	0	0	0	98	345	949	1405	1405
0	0	0	0	28	291	829	1214	1351
0	0	0	6	54	217	464	943	1351
0	0	0	6	46	108	149	478	861
0	0	0	0	0	0	0	46	159

total absorbed energy 2nd stage 21049 Nm

0	0	0	0	0	0	0	0	0
0	0	0	0	0	0	15	48	178
0	0	0	0	0	0	280	593	783
0	0	0	0	0	0	385	819	978
0	0	0	0	0	0	120	274	563
0	0	0	0	0	0	0	0	223
0	0	0	0	0	0	0	0	33
0	0	0	0	0	0	0	0	0

total absorbed energy 3 rd stage 5292 Nm

0	0	13	125	835	1257	984	1124	1736
0	0	19	180	665	860	1119	1666	2114
0	0	113	375	737	919	1862	2869	3059
0	0	119	468	908	1765	2833	3304	3463
0	0	246	647	917	1502	2369	2882	3307
0	0	376	854	988	1166	1805	2650	3281
0	0	213	541	678	720	876	1340	1765
10	35	141	301	320	268	399	675	911

total absorbed energy total barrier 67677 Nm

Figure 10. Golf to ADAC barrier: Only the first two barrier stages absorb greater amounts of energy, but the 3rd stage is needed to avoid bottoming out

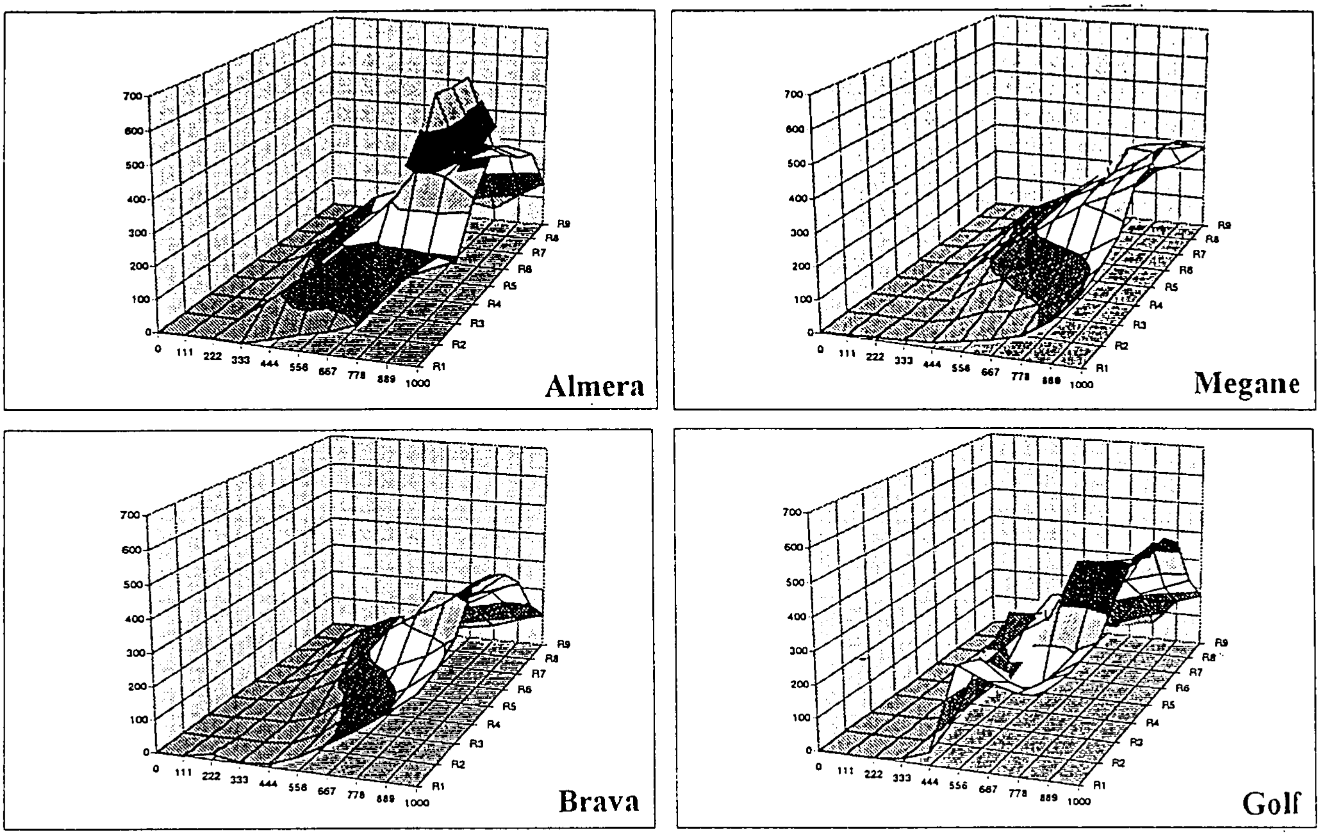

Figure 11. Barrier deformation: Brava produces the lowest and most equalized barrier deformation (low aggressivinces), Almera the highest and less equalized values (high aggressiveness)

Fig 9 shows that the ADAC barrier performance is excellent with the Brava and Megane, good with the Golf and even sufficient with the Almera with its extremely rigid longitudinals. Although with the latter the last stage had been considerably affected there was no bottoming out.

Summary: The ADAC barrier is a considerable improvement compared to the EEVC barrier. But further tests will have to prove whether all requirements can be met for all other vehicles.

Aggressiveness and compatibility

For the estimation of the energy absorbed in the barrier the horizontal barrier deformation was measured over the total surface. From the deformations Δx of the individual stages the energy absorbed by the barrier $[J/m^2]$ can be determined with the formula

$$e = (\Delta x * p)_{bumper} + (\Delta x * p)_{1.St} + (\Delta x * p)_{2.St} + \Delta x * p)_{3.St.}$$

Fig 10 shows the dissipation of energy determined by this formula for the crash Golf/ADAC barrier for the bumper and the individual stages. It can be seen that the energy is mainly absorbed by the first and second stage. The third stage is necessary to prevent bottoming out.

Fig 11 shows a comparison of the barrier deformation of the vehicles. Figures for the Brava are the lowest and most stable and the Brava is therefore the least aggressive vehicle. The extreme opposite is the Nissan with the deepest barrier deformation by far. Due to the extremely varied rigidity distribution of the front end it is certainly the most aggressive of the four vehicles tested. One question remains: is the Brava too soft, since it takes very little energy into the barrier? When more information is available it should be considered whether limits for barrier energy absorbtion depending on vehicle weight class should be defined as design targets.

Summary: With ADAC barrier there is the possibility to gain information on aggressiveness and compatibility through barrier deformation.

Crash speed

In order to get information on correct crash speed the energy absorbed by the vehicle has to be determined based on the formula

$$E_{absorbed\ by\ car} = E_{kinetic\ before\ impact} - E_{total\ abs.\ by\ barrier} - E_{rebound\ yaw} - E_{rebound\ translat}$$

The speed equivalent to this energy is

$$EES_{car} = (2 * E_{absorbed\ by\ car} / m)^{0,5}$$

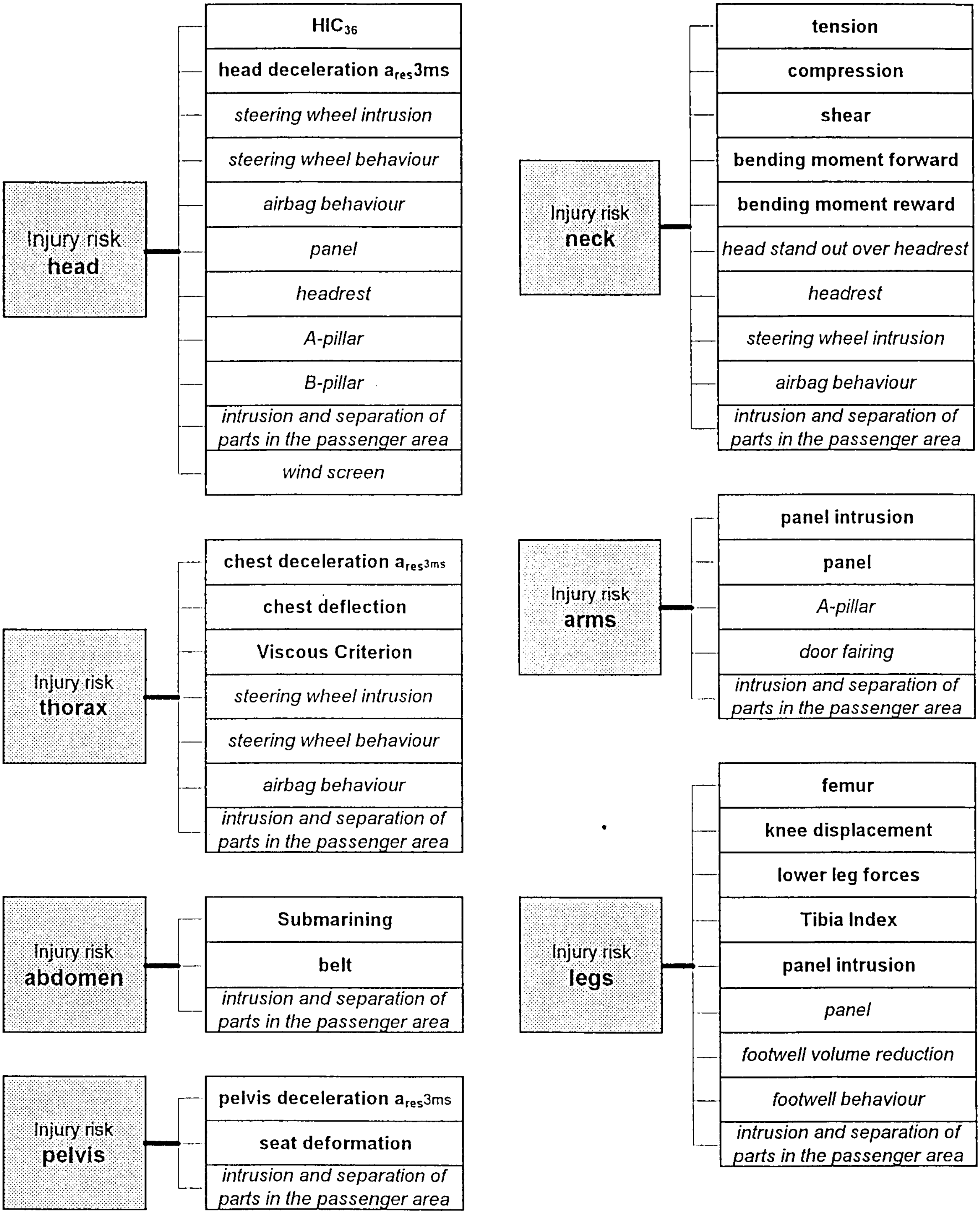

Figure 12. Criteria structure for driver specific injury risk rating

	car to ADAC barrier				car to car
	Almera	Megane	Brava	Golf	Golf /Astra
car mass (kg)	1545	1469	1473	1394	1245
car impact speed (km/h)	60	60	60	60	50
rebound yaw speed (1/s)	3	3,5	2,9	3,3	2,9
rebound translat. speed (m/s)	3,5	3,7	2,7	3,5	3
energy bumper (Nm)	5565	3981	8177	6322	0
energy 1rst stage	34401	30500	28466	35013	0
energy 2nd stage	15950	12191	7111	21049	0
energy 3rd stage	6184	114	12	5292	0
total energy barrier	62099	46787	43766	67677	0
rebound yaw energy	9472	11361	8067	9152	6313
rebound transl. energy	9463	10055	5369	8538	5603
total kinetic energy before impact	214583	204028	204583	193611	120081
energy absorbed by car	133549	135825	147381	108243	108166
EES car (km/h), (abs.by car)	**47,3**	**49,0**	**50,9**	**44,9**	**47,5**
EES car - EES car (car/car)	**-0,1**	**1,5**	**3,5**	**-2,6**	**0,0**

Figure 13. Energy calculation shows: 60 km/h against ADAC barrier corresponds to a 50 km/h car to car crash within a tolerance of about +/- 3 km/h

The results are presented in Fig 13.

Consequently the speed equivalent to the energy absorbtion in the vehicle for the car-to-car crash with 50 km/h is 47.5 km/h. The Brava with its extremely soft structure absorbs more deformation energy (corresponding to 3.5 km/h) and the Golf with its comparably rigid body less (corresponding to 2.6 km/h). .

Summary: The test speed of 60 km/h against the ADAC barrier generates the same accident severity as the 50 km/h car-to-car crash with a tolerance of about +/- 3 km/h.

RATING AND VALIDATION

The aim is to obtain the best possible practical information on passive safety from crash test results. Since the employed test method is a structure test and not a restraint system test not only dummy loadings will have to be integrated but possibly all laboratory crash results and in particular vehicle-specific results like compartment deformation. performance of possible impact zones and the intrusions of vehicle parts [6].

It also would seem useful to evaluate the results passenger- and vehicle-related: The consumer is mainly interested in what happens to the passengers and the design engineer in how indidividual parts perform in a crash. As an example the criteria structure for the assessment of the injury risk of body parts of the driver developed on the basis of current accident statistics is presented in Fig 12. The results of specific injury risks are then summarised via a weighted average to produce the overall injury risk of the driver. Fig 14 shows the result of such rating for the above-mentioned test series Almera. Megane. Brava and Golf. With an overall rating for the Nissan Almera of 1.9 (1 = very high safety level. 5 = poor safety level) it provides the highest safety level for the driver.

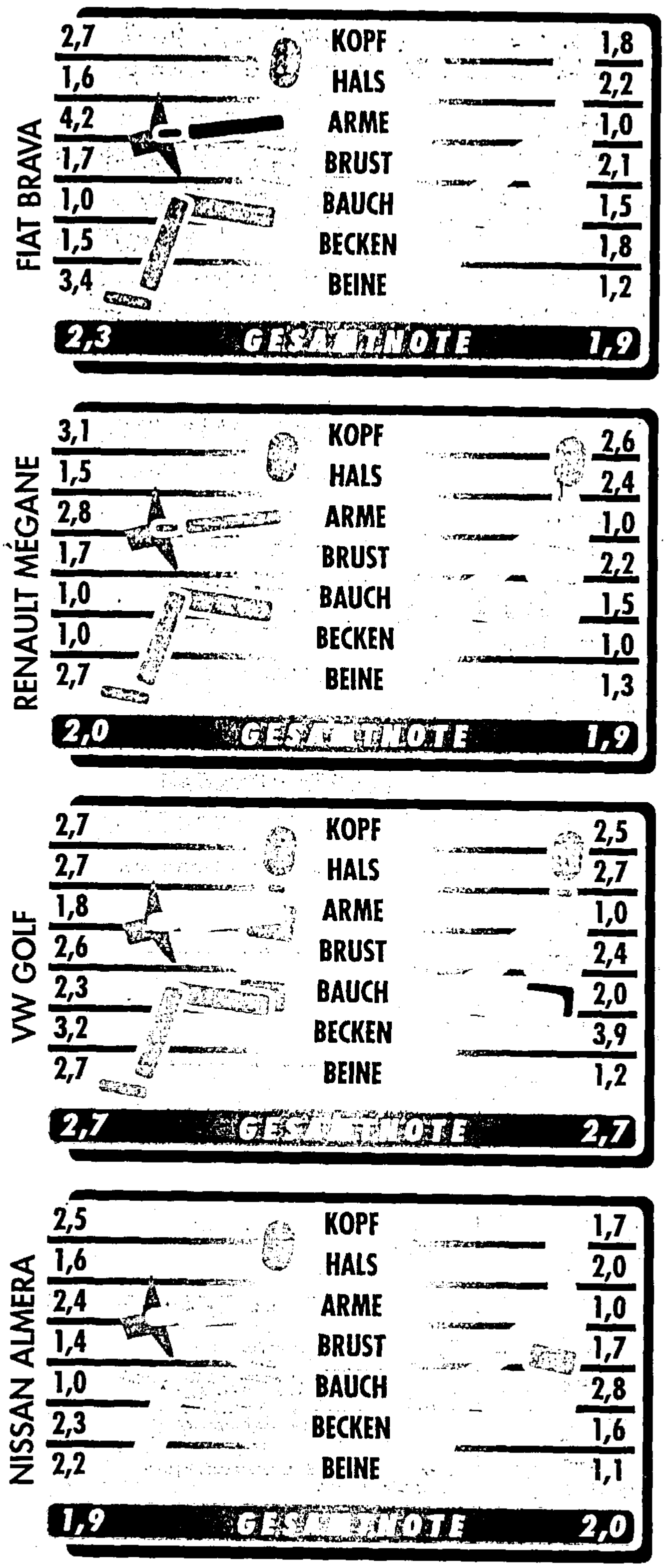

Figure 14. Injury risk rating figures for driver and rear seat passenger: Nissan Almera reaches the highest marks

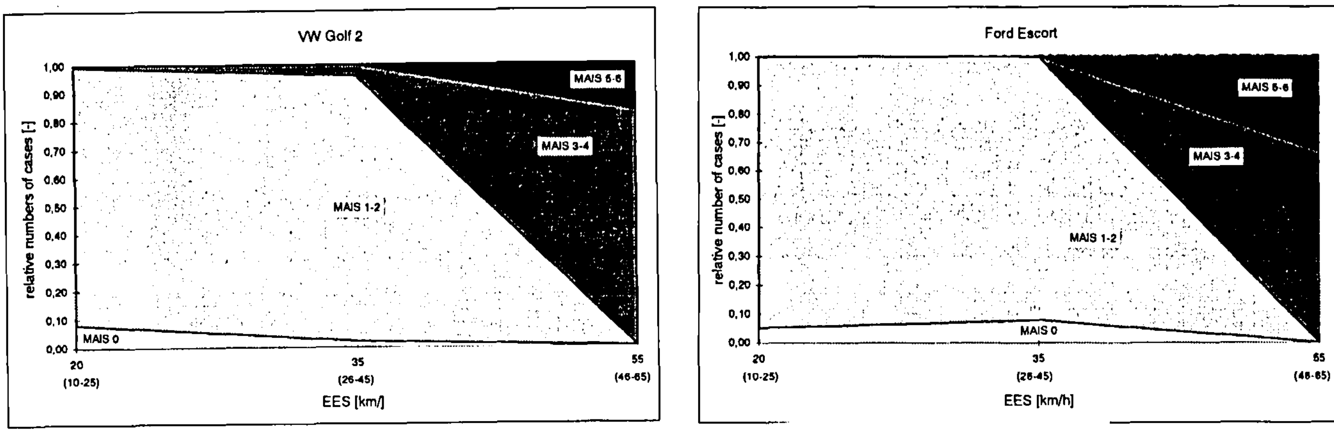

Figure 15. For higher accident severeties (EES): driver injury risk in Golf lower than in Escort

Validation means that it is established to which extent the results from crash test and rating are identical with the real accident process. It should be sufficient to establish exemplary proof on the basis of only a few vehicle models. A basic problem is that normally accident statistics do provide larger scale data for older vehicle models only. For this reason two 1989 vehicle models were selected for validation. i.e. the VW Golf 2 and the old Ford Escort.

The rated crash test results generate a rating of 3 (sufficient safety level) for the driver injury risk for the Golf and 4 (less sufficient safety level) for the Escort [7]. It should be noted that Escort dummy loadings are on the whole lower than those of the Golf. The Escort has a much softer compartment and therefore a lower crash test deceleration level. The higher injury risk figure is exclusively the result of severe compartment deformations and component intrusions. A rating method based only on dummy loadings would generate a different result.

From a representative sample of some 15.000 car-to-car accidents (i.e. 15% of all car-to-car accidents with personal injuries reported to the insurance companies in Germany in 1990) the respective Golf and Escort frontal accidents were extracted [8]. of those 200 Golf and 100 Escort cases could be used for validation.

Fig 15 shows the evaluation of the data on the basis of the parameters severity of the accident (EES) and of driver injuries (MAIS). This demonstrates that with high EES the rate of serious injuries and therefore the injury risk for the Escort driver is higher than for the Golf driver. This proves that there is at least no qualitative discrepancy between rating results and accident statistics.

An interesting fact is that for low EES the result is reversed. i.e. accident data for the Golf show a relatively higher injury risk than for the Escort. The reason could be the soft Escort compartment and the resulting low deceleration level.

Summary: For the evaluation of structure tests rating can only produce practical results where in addition to dummy loadings generally all injury-relevant vehicle-specific results are integrated. A comparison between crash test data rated with the above mentioned system and real life accident data shows at least acceptable qualitative conformity.

CONCLUSIONS

Crash tests with the EEVC barrier showed reproducible results. Depending on vehicle design considerable deviations compared to real car-to-car crashes are however still possible. Conclusions on aggressiveness and compatibility as one of the essential objectives for a new structure test will in many cases not be possible.

First tests with a three-stage barrier. i.e. a development of the current EEVC barrier. produce much better results. At least for all vehicles up to lower middle-class the requirements will have been met by this barrier.

Deviations compared to the car-to-car crash are considerably less and there is the possibility to gain information on aggressiveness and compatibility through barrier deformation. The test speed of 60 km/h against this new barrier with a tolerance of +/- 3 km/h causes the same accident severity as the 50 km/h car-to-car impact speed. To which extent the barrier is suitable for all passenger cars and minibuses will have to be confirmed by further tests.

For the evaluation of structure test only such rating methods are suited which in addition to dummy-specific also integrate vehicle-specific data. A validation shows that rating results at least from the qualitative aspect are consistent with real accident processes. However. also with regard to this issue further activities will be required in order to confirm results.

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A COMPARISON OF AUSTRALIAN AUDIT CRASH TESTS AND REGULATORY REQUIREMENTS WITH INJURY PREDICTION

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Paper Number 96-S1-W-18

ABSTRACT

Audit crash tests of showroom vehicles by Australian NCAP, with subsequent media promotion and ready availability of the results has enabled the Australian vehicle user to be better informed of the safety of vehicle occupants in frontal collisions. The difference between the regulatory and audit crash test speeds can have a significant effect in the potential for head chest and leg injury. Data is presented to show the injury differences to front seat occupants in frontal crashes including the relationship of the speed of the crash to the injury severity of occupants in the crash.

INTRODUCTION

The role in improved regulatory requirements is acknowledged as not all vehicles in the market will be subject to audit crash testing. Regulatory programs ensure the minimum safety of all vehicles available in the Australian market. Both audit and regulatory requirements require that the test be representative of real world crashes where occupant survival is possible. Australia represents a very small percentage of world vehicle production, it is also important that regulatory and audit type crash test procedures are in harmony with world practice.

The Australian NCAP Full Frontal crash tests carried out to National Highway Traffic Safety Administration (NHTSA) 56 km/h test procedures with the Offset test very similar to the Insurance Institute for Highway Safety (IIHS) American crash test procedure at 64 km/h.

The Australian passenger car market consists of 28 manufacturers marketing 139 different models with almost 200,000 built in Australia and another 300,000 imports.

REGULATORY REQUIREMENTS

The Australian regulatory requirements for new vehicles requires reporting of test results to Australian Design Rule (ADR) 69 on occupant safety. Regulation ADR 69 is very similar to Federal Motor Vehicle Safety Standard (FMVSS) 208 except the occupant dummy is belted for the crash test. The function of ADR 69 is to....***specify vehicle crashworthiness requirements on front seat dummies in a frontal crash to minimise the likelihood of injury.***

Results of manufacturers tests are reviewed by the Federal Office of Road Safety (FORS) and the results remain within the office, and are not available to the public or researchers. Consumers are advised that the vehicle meets the requirements of the regulations for minimum legislated safety standards. All passenger vehicles manufactured for sale in Australia from January 1996 must meet the safety standards of ADR 69.

The existing regulatory system requires all passenger vehicles to meet the ADR 69 safety requirements, but the vehicle categorisation system excludes 4 Wheel Drives, vans and utility vehicles. This means that almost 28 percent of the new Australian vehicles have no similar occupant protection regulations (ADR 69). Plans exist to introduce the regulated ADR 69 requirements to 4 Wheel Drives, vans and Utilities within several years.

REGULATORY STANDARDS DEVELOPMENT

An evaluation program by FORS carried out in 1992 (OR 11) , published results from frontal crash tests on 7 high volume passenger vehicles. All but one vehicle had a HIC maximum value under 1000. All vehicles met the chest deceleration maximum value of 60 g and chest compression of less than 75mm. One vehicle exceeding the lower leg maximum allowable load of 10 kN. For vehicles not required to meet the ADR 69 standard until 1996 this was a satisfactory result and showed that the Australian vehicles were meeting the safety standards for occupant protection that the regulation would shortly require. However passenger head contacts with the dash were recorded in four of the vehicles which can cause neck injury.

INJURY LIMITS

The injury limits noted in table 1 above were suggested by Mertz and were adopted in FMVSS 208 and ADR 69. Values for HIC, chest g, chest deflection and femur loads below the injury threshold numbers determined by Mertz predict that a serious injury rated by Abbreviated Injury Scale (AIS) 3 would not occur.

TABLE 1
FEDERAL OFFICE OF ROAD SAFETY CRASH TEST RESULTS
PASSENGER CARS
Full frontal crash at 48 km/h (OR11)

Vehicle 1991 Models	Comm-odore Driver/Passenger	Falcon D/Pass.	Magna D/Pass	Camry D/Pass.	Pintara D/Pass.	Laser D/Pass.	Corolla D/Pass.	Limit ADR 69
HIC	622/872	848/699	779/NA	623/523	882/322	1012/860	820/NA	1000
Chest g	58/42	48/46	54/51	47/41	47/46	59/48	47/43	60 g
Chest compression	49/39	37/39	42/37	41/29	39/33	43/29	46/30	75mm
Femur Left kN	2.1/2.0	3.3/1.6	2.3/1.2	15.4/1.8	1.0/1.3	4.2/1.8	1.1/1.5	10kN
Femur Right	1.8/0.9	1.9/2.0	2.1/1.1	3.4/3.1	3.6/3.1	6.8/2.9	9.4/1.4	10 kN

Serious injury included reversible brain concussion and bone fractures. However if the injury levels were above AIS 3 rating then the injuries would be regarded as life threatening and would include brain damage, or thoracic and abdominal organ damage

The report concluded that ***none of the vehicles tested produced dummy response which were likely to be life threatening for the front seat occupants***.

The results of these crash tests show that in 1991 when these cars were built, where they represented 59 percent of the new passenger vehicles sold in the Australian market, all were close to meeting the proposed ADR 69 occupant protection standard.

4 WHEEL DRIVE, VANS & UTILITIES

Another report by FORS released in 1995 (OR 17) addressed the occupant protection safety levels of Four Wheel Drives and Light Commercial vehicles. Market analysis had shown that this group of vehicles accounted for 26 percent of the Australian new vehicle market. It was considered that this increasing segment of the market was likely to continue and some additional occupant protection measures needed to be added to the legislated requirements.

TABLE 2
FEDERAL OFFICE OF ROAD SAFETY CRASH TEST RESULTS
4 WHEEL DRIVE AND UTILITIES
Full frontal crash at 48 km/h (OR17)

Vehicle 1994 Models driver/passenger	Toyota Land-cruiser	Mitsubishi Pajero	Suzuki Vitara	Toyota Hilux	Mitsubishi Triton	Holden Rodeo	Limit ADR 69
HIC (36)	774/573	1167/628	945/1379	881/589	791/471	709/613	1000
Chest g	43/37	46/48	69/54	51/46	42/40	48/43	60
Chest Compression	44/34	54/44	57/46	43/33	39/38	42/40	75mm
Femur Left kN	3.0/2.3	0.8/0.7	5.2/0.6	0.7/1.4	0.8/1.1	0.2/0.7	10 kN
Femur Right kN	2.8/3.8	0.7/1.3	1.8/0.6	3.0/1.9	0.9/0.6	0.4/0.6	10 kN

Fatal and serious injury accident studies confirmed the trend that 4 Wheel Drives, vans and utilities were over represented in Head on accidents resulting in serious injury and therefore justification existed to improve the occupant safety level of this group of vehicles.

The report concluded that Head injuries were generally unlikely but could occur in two of the vehicles, with chest and thoracic injuries likely to occur in one vehicle.

As a result of the tests a recommendation has been made to shift the focus toward performance based testing of the vehicle safety system as a whole and would therefore bring the level of occupant protection equal to that of passenger cars. The result would be to include the group of vehicles in the regulated group required to meet ADR 69 occupant safety levels.

The above group of vehicles represented 56 percent of the 4 Wheel Drives and 46 percent of the utilities sold in the Australian market in 1994.

FUTURE TRENDS IN OCCUPANT PROTECTION

Improvements can be expected from meeting the requirements of ADR 69 in the passenger vehicles, four wheel drives , utilities and passenger vans if data from Australia is compared to historical US published material.

TABLE 3
COMPARISON OF OCCUPANT SAFETY
Results for full frontal crash tests at 48 km/h,

	HIC driver	HIC Passenger	Chest g Driver	Chest g Passenger
Australian Passenger Vehicles 1991	80	65	85	75
Aust. 4WD Utilities 1994	88	71	83	75
77-79 US passenger car	81	82	78	65
87-92 US passenger car	47	41	75	63

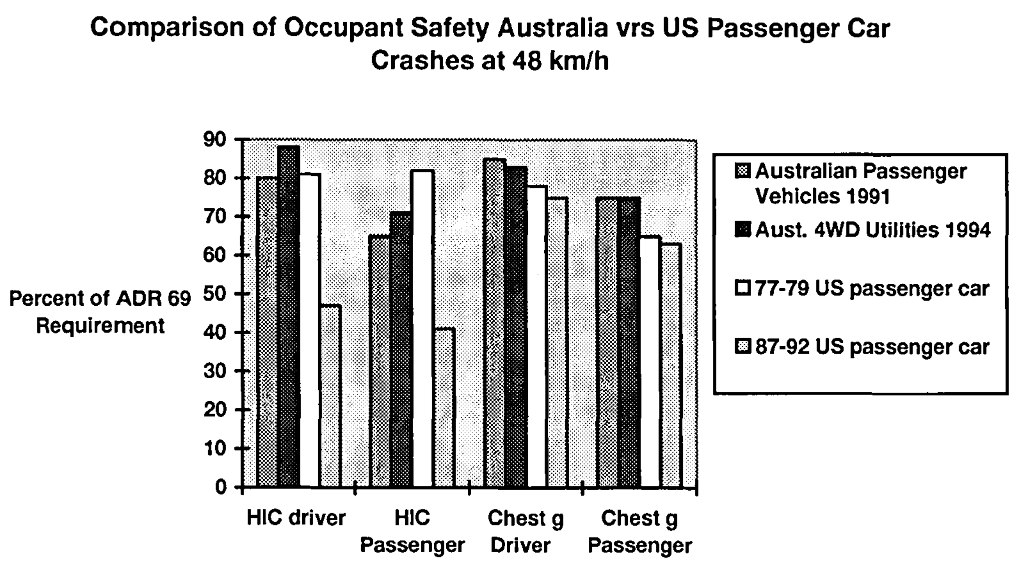

Figure 1 COMPARISON OF THE SAFETY PERFORMANCE OF PASSENGER CARS, 4WD AND UTILITIES IN AUSTRALIA AND U.S.

Figure 1 indicates the Australian fleet was meeting the intent of the ADR 69 occupant safety requirements in 1994. Some improvements would be expected as the vehicles that exceeded the requirements, especially in HIC, would make changes to improve the occupant safety and this would lower the average values. The level of occupant safety that could be expected from this measure alone may be small. However, other manufacturers may also update safety systems at the time of ADR certification which would lower the average from the 1992 data.

A study of the levels of HIC and chest 'g' from the NHTSA results that are published from FMVSS 208 certification for the 48 km/h full frontal crash test in the US may give guidance to the levels of HIC and chest 'g' that Australia could expect in the future.

In Fig 1 data from FORS passenger car tests carried out in 1992 were compared to the US passenger cars by Mr J Hackney of NHTSA. He said in the 48 km/h test the Australian FORS data is equivalent to that of 1977-79 US results. The US fleet results averaged over 1987-1992 passenger cars had dropped by a significant 50 % from the earlier 77-79 results.

Australia could therefore expect a similar drop in the 48 km/h FMVSS 208 data in the Australian passenger car fleet in the next few years.

The Australian ADR system that administers the ADR 69 crash tests do not presently make available ADR 69 data, so for all of the models that are not part of NCAP, consumer or occupant safety researchers will have no data on which to make an informed decision or to study the improvements in the level of occupant safety.

AUDIT NCAP TYPE CRASH TESTS

The consumer led vehicle occupant safety program NCAP which commenced publishing crash test results in 1993, has resulted in new vehicle buyers becoming more aware of differences in vehicle safety.

Vehicle safety promotional material from manufacturers has also contributed to the improved level of occupant safety with new car brochures and displays at car shows. Some Television Advertising of new vehicles shows vehicle occupant safety as an important aspect in the promotional material.

A national study undertaken in 1995 in Australia found that motorists thought that special car safety and crash tests to be an extremely important service that motoring organisations could provide.

It is impossible to confirm the contribution any specific promotional material is the reason that car safety has taken a higher profile, but it is certain that follow up marketing by the manufacturers has shown the benefit of promoting vehicle safety or it would not be continued.

NCAP has been crash testing and publishing the results since 1993. One group of vehicles have shown a significant reduction in the likelihood of the front seat occupant receiving a serious injury in a frontal crash.

LARGE/MEDIUM CARS NCAP CRASH RESULTS

The results of this initial NCAP testing showed the occupants of Australian vehicles would have increased risk of serious injury at the higher crash speed of 56 km/h compared to the results from the FORS 48km/h crash tests. Only one vehicle had a driver HIC below 1000, the injury threshold for serious injury, with one third of the vehicles with chest 'g' below the injury threshold .

Equivalent US star ratings have been added to the table to show the level of occupant protection, and the number of single stars is of concern. The additional energy of the crash which is 36 percent more than the 48 km/h crash test give a greater spread of HIC and chest values than in the 48 km/h test.

TABLE 4
NCAP CRASH TEST RESULTS, 1992 PASSENGER CARS

	HIC	Chest 'g'	Injury risk	Star Rating per NHTSA
Volvo 940 Driver	490	50	14	4 star
Passenger	600	46	14	4 star
Toyota Camry Driver	1090	63	41	2 star
Passenger	1240	52	42	2 star
Mitsubishi Magna TR Driver	1140	60	41	2 star
Passenger	1580	58	70	1star
Mazda 626 Driver	1160	60	42	2star
Passenger	930	55	27	3star
Subaru Liberty Driver	1360	58	54	1star
Passenger	1810	44	81	1star
Honda Accord	1500	58	64	1star
Passenger	1330	50	48	1star
Ford Falcon EB Driver	1340	74	65	1star
Passenger	780	58	25	3star
Nissan Pintara Driver	1750	64	82	1star
Passenger	890	67	38	2star
Holden Commodore VP Driver	1690	82	86	1star
Passenger	2410	55	97	1star

AUSTRALIAN NCAP RESULTS PASSENGER CARS 1993

COMPARISON OF AUSTRALIAN WITH US NCAP DATA

Comparisons between 1993 Australian large/medium passenger cars NCAP and US NCAP results for model years 1987-92 showed that the Australian fleet lagged behind in occupant safety levels. Research available to improve occupant protection was expected to be applied to these vehicles which was substantiated with the better 1995-6 NCAP results.

LARGE MEDIUM PASSENGER CARS

In more recent data published by Australian NCAP in 1995-96, the results for the large/medium passenger car fleet had improved significantly. The average injury risk which is the percent chance of the occupant receiving a life threatening injury (or AIS greater than 3), was now 26, down from 54, a very worthwhile and significant improvement in occupant protection. This reflects the improvement of some of the poorer performers from the earlier test and their significant market share.

	HIC driver	HIC Passenger	Chest g Driver	Chest g Passenger	Injury Risk Driver
Australian Passenger Vehicles 1993	128	128	105	90	54
Australian Passenger Vehicles 95/96	101	104	92	79	26
79-80 US passenger car	128	130	95	84	NA
87-92 US passenger car	85	73	81	72	NA

COMPARISON OF OCCUPANT SAFETY, AUSTRALIA vrs US PASSENGER CARS,56KM/H CRASH

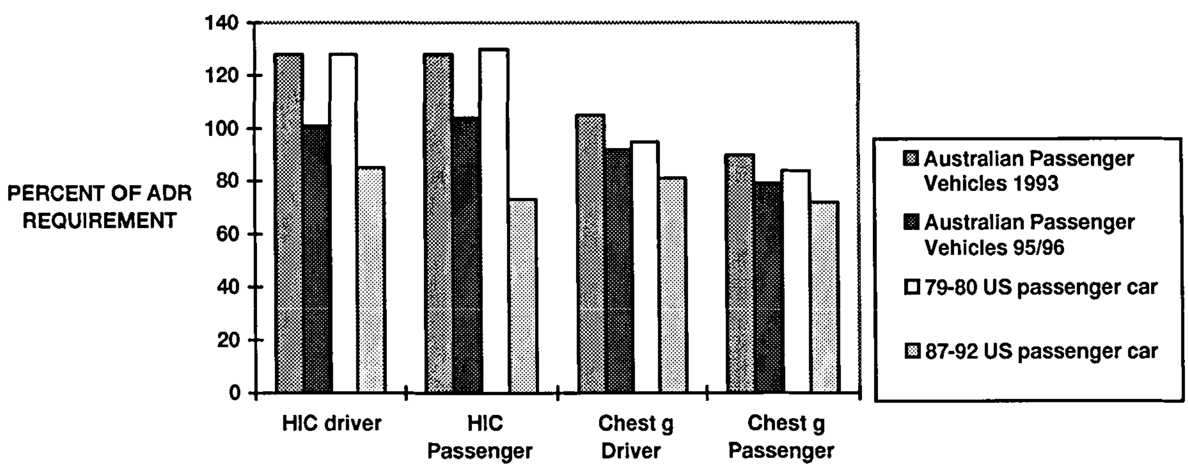

FIGURE 2 Comparison of the safety performance of large/medium passenger cars in the Australian and U.S. markets from 56 km/h Frontal crash test

SMALL PASSENGER CARS

Australian small car results in 1994 had a driver and passenger injury thresholds well above the life threatening level and these have been maintained with later crash tests in 1995/6. Injury risk for drivers averaged 53 in 1994 tests dropping to 49 in early 1996. Many popular models with the largest market share have this level of occupant protection.

Equivalent passenger cars in the US in 1995 would now result in occupant safety in the three and four star ratings which is equivalent to injury risk numbers from 10 to 35. In the US 95 model year 85 percent of new vehicles were fitted with driver and passenger airbags. None of the models tested in Australia in 1995 had air bags. This could explain the difference in occupant protection levels measured by the NCAP crash tests.

TABLE 5
SMALL CAR HIC PASSENGER CARS, PERCENT OF ADR 69 REQUIREMENT

	HIC driver	HIC Passenger
Australian Small Cars 1993	131	128
Australian Small Cars 1995/6	149	119
US 1993 Small Cars	68	66

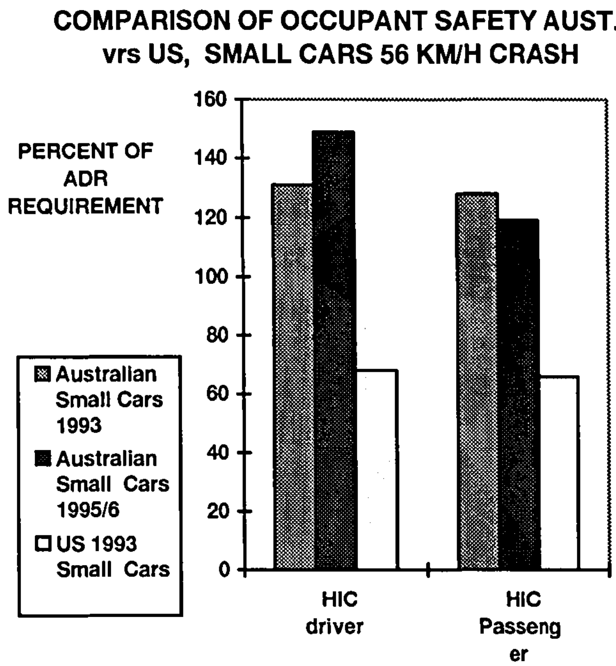

FIGURE 3 Comparison of the Safety Performance of small cars in Australian and US, 1993 for 56km/h crash

PASSENGER VEHICLE OCCUPANT SAFETY EQUIPMENT

One difference observed in the Australian NCAP testing was the number of vehicles tested that did not have the same level of safety equipment that equivalent model for the US market had as standard equipment. Cars manufactured in Australia for export to America and Europe in the years 1992-95 were equipped to higher safety specification levels than models sold in Australia. Passenger airbags are only starting to be offered as standard equipment in the smaller car segment of the Australian market in the 1996 model year models. Reviewing the European passenger vehicle specifications shows the European model is more likely to have a driver airbag as standard equipment than the equivalent model marketed in Australia in 1995/6.

In the small car segment of the market, price is very important. The typical additional cost of the driver airbag of A$ 950.00 results in very few owners taking up the airbag option. It is estimated the optional airbag installation rate in 1995 was only 5 percent.

CONCLUSION

There is a role for both legislated and consumer crash testing of vehicles to improve the level of occupant safety in Australian vehicles. Data presented shows that cars made for Australians in 1995/6 are still not at the level of occupant protection that US models were at in US model years 1987-92. With the introduction of new models in Australia there is evidence that the occupant protection levels are improving. However only some manufacturers are responding positively. Some differences exist that inhibit the wider spread adoption of airbags in driver and passenger positions, and there is no doubt that the regulated requirement for airbags in the US has accelerated the fitting of these devices. No similar regulation exists in Australia. Market pressure by consumers increasing interest in vehicle safety should accelerate provision of airbags, but the consumer in the very price competitive segment of the market is presently not prepared to pay for the additional occupant protection airbags would provide. It is imperative that we legislate to make the Australian cars as safe as possible meeting international best practice.

ACKNOWLEDGMENTS

The author would like to acknowledge the assistance of Australian NCAP groups and the South Australian Department of Transport in supporting the publication of this paper. The views expressed in the paper are those of the author alone and do not necessarily represent the views of the SA Department of Transport.

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LIGHT TRUCK SAFETY CONCEPT MODELS AND THEIR APPLICATIONS

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Paper Number 96-S1-W-19

ABSTRACT

In today's fast changing world, the key to success in vehicle design is to bring the vehicle to the customers at the earliest possible date. Also vehicles should possess the highest quality and safety standards, as both are important features to the customers. To achieve this, safety CAE should be used to set targets and guide design in the up-front process instead of trial-and-error approach. To make safe vehicle up-front stage is very crucial in determining vehicle architecture, performing trade-off studies and setting design targets. All of this can be achieved by using safety CAE Concept/ Hybrid Models in the early design stages, where trend prediction is more important than exact prediction.

This paper illustrates development of one such safety CAE tool. The MADYMO full vehicle concept model of a body-on-frame utility vehicle for frontal impact was developed to perform up-front safety analysis. This full vehicle model is capable of predicting both structural and occupant response. To gain confidence, the model was correlated with the test for both structural and occupant response. A problem of higher chest G's was identified. Two approaches were taken to look at the problem:

- Direct optimization was used with some parameters to minimize the chest G's.
- DOE (Design of Experiments)approach was used to study the sensitivity analysis and understand the interaction of various components on the response of interest (chest G's).

The results and the direction provided by both the approaches are compared later. This paper demonstrates the approach to solve such problems. A methodology was also developed to generate this type of MADYMO concept models for all other similar type of configuration. This new tool needs geometry, weight, stiffness characteristics etc. of key components as an input. Based on this information it generates the full vehicle MADYMO model which can be used to predict occupant and structural response. This method has been implemented for body-on-frame pick-up trucks and utility vehicles. Later it will be extended for unibody and subframe type vehicles. Once implemented this new method will help drive a more efficient and effective design for safety.

INTRODUCTION

Recently there is a tremendous push from the management to reduce the cycle plan for the vehicle development and prototypes drastically. This in turn puts more emphasis on quick up-front analysis. Typical current safety CAE process for the system level analysis is explained in Figure 1 (days indicated are for illustration purpose only). Two major time consuming steps in the cycle are building the base model and performing trend studies. If we assume total analysis cycle time of 45 days, 100% improvement in the two major time consuming steps will result into almost 60% improvement in the cycle time and efficiency. With this in mind a methodology was developed for generating standard concept model and linking them to DOE (Design of Experiment) for parametric studies which can drastically reduce the time step and improve the efficiency.

MODEL

The full vehicle MADYMO frontal concept model for body-on-frame type of vehicles for 31 m.p.h. unbelted condition was developed (Figure 2). It can be divided into interior , Figure 2a & chassis Figure 2b.

The chassis has most of the structural energy absorbing members (shotgun, frame, rediator, engine etc.). The frame geometry is represented by ellipsoids and the buckling/bending of the frame are represented by Cardan and point restraints. Both frames are connected to each other by cross members (only first two cross members are modeled). The bumper characteristics are modeled into the frame, as the bumper is insignificant for truck frontal impact. Radiator and engine are modeled using planes and ellipsoids.

The interior has most of the cab data. Dash, toeboard, floorpan, roof, windshield, seats are modelled by planes. Steering system is modelled with ellipsoid. The steering system is capable of collapse and rotation. The model includes a partially folded flat airbag. Hybrid-III dummy is positioned at the mid-seat position for the analysis.

Barrier and ground are modelled with planes in the inertial co-ordinate system.

RESULTS

The full vehicle model is simulated for 31mph unbelted, frontal impact condition. Figure 3 summarizes the results for both structural and occupant response.
Structural responses - rocker@B-pillar, frame@B-pillar & engine pulse, velocity & displacement trends are well captured & results are in the ball-park

Occupant responses - resultant chest & head acceleration and femur loads are also in the ball-park.

Note: Interior portion has not been validated at this point and is used to demonstrate the methodology.

PROBLEM

Occupant response shows the higher chest G's. It was necessary to lower the chest G's to meet FMVSS 208 (Federal Motor Vehicles Safety Standards) criteria. Two approaches were taken to look into this problem:

1. DIRECT OPTIMIZATION
2. DOE.

DIRECT OPTIMIZATION APPROACH

The optimization method uses Box Complex method modified for constraints [1,2]. The method is robust for optimizing non-smooth functions. The design variables to be optimized are identified and constraints are entered as upper and lower bounds on each design variables.

The OPTIMIZER program[3] works in the following sequence:

1. The user defines initial values and ranges of design variables.
2. With these values, a new MADYMO input file is created.
3. The Program submits the MADYMO run and waits for the job to finish.
4. The program then analyzes the results, calculates various peak values to determine the value of user determined objective function.
5. The optimizer checks for convergence and stops if converged.
6. If convergence is not reached, it determines a new set of design variables values and repeats the process from step 2.

The typical objective function for occupant analysis given below is minimized.

$$f = \left(\frac{HIC}{CL}\right)^{n1} + \left(\frac{ChestG}{CL}\right)^{n2} + \left(\frac{FemurLoad}{CL}\right)^{n3}$$

$n1,n2,n3$: Powers to penalize factor that exceeds corporate guidelines.
CL : Corresponding Corporate Limit

For this particular study the occupant's chest G's alone was minimized. Optimization was performed using following factors and response.

Factors:

- Fronthorn Stiffness
- Shotgun Stiffness
- Steering Column Stroke
- Body Mount Stiffness
- Engine Mount Stiffness

Response:

- Chest G's (3 ms)

Figure 4 shows how the individual factors are varied. The optimizer converged after many iterations giving the occupant and structural response as shown in the Figure 5. As seen the chest G's were reduced considerably as it was the only thing included in the objective function. Figure 6 shows the optimum settings of the parameters to achieve the minimum chest G's.

Direct OPTIMIZATION provided following directions to reduce the chest G's:

- Front Horn: Higher initial load and stiffness along with lower level of load in the later part.
- Steering Column: Lower steering column load.
- Shotgun: Lower shotgun stiffness.
- Body & Engine Mounts Stiffness: Higher stiffness.

DOE APPROACH

DOE is an advanced statistical method used for systematic analysis of parameter effects on the response of interest [4]. RSM (Response Surface Methodology) approach of classical DOE was used to study the effect of design variables and interaction on the response of interest. From RSM analysis, empirical equations are generated (linear or non-linear) representing the system behavior as a function of design variables. These empirical equations are used to examine the entire design space. This is a robust method for continuous smooth functions.

DOE process works in the following sequence:

1. Identify the response to optimize.
2. Determine the factors that influence the response
3. Screening DOE to identify factors that have the largest effect on the response and eliminate the rest
4. Define experiment with appropriate levels of each factors.
5. Run the experiment.
6. Generate the equation.
7. Confirm results (fit).

For this particular study an experimental matrix was setup using the factors and responses used with OPTIMIZER (the range and variation for each factor was different however).

Typical main effect and interaction plots for the setup for Chest G's are shown in Figure 7 and Figure 8 respectively.

Main Effects:

Front Horn: Higher Front Horn initial load (F2) and lower load in the later part (F3) will reduce the chest G's. It also quantifies the reduction eg. as we increase the initial load from 80 to 120%, keeping the other things at nominal, the chest G's decrease by 5.96 G's.

Steering Column: As the steering column load is reduced from 120 to 80% the chest G's are lowered by 8.72 G's.

Shotgun: As the shotgun stiffness is dropped from 120 to 80% the chest G's are reduced by only 3.4 G's.

Engine and Body Mount Stiffness: Has minimum effect on the chest G's.

As seen DOE study not only supports the conclusions drawn earlier by the OPTIMIZATION study but it also provided the direction (that is by lowering/increasing the stiffness whether the response decreases/increases).

Interactions:

Figure 8 shows one of the interaction of FrontHorn load (later part) and column stroke on chest G's. As seen to reduce chest G's front horn load in the later part should be low (80% of the base) and the stroke should be slightly lower. Higher FrontHorn Load in the later part and lower steering column load will result into higher chest G's.

Empirical equation obtained from DOE has following form:

$$Y = \beta_\circ + \sum_{i=1}^{k} \beta_i x_i + \sum_{i=1}^{k} \beta_{ii} x_i^2 + \sum_{i<j}^{k} \sum \beta_{ij} x_i x_j + \varepsilon$$

Where

Y : response

x : factor

$\beta_\circ$: constant term

β_i : coeff. for main effect term

β_{ii} : coeff. for square term

β_{ij} : coeff. for interaction term

k : levels

ε : error term

METHOD

As explained earlier and in Figure 1, building the full vehicle MADYMO model from line data is the most time consuming part. Reflecting the changes of the geometry in the model is also a difficult task. Particularly for a new analyst, this is one of the most time consuming process. Ideally, analysis should lead the design, however, due to the above mentioned fact, more often analysis tends to follow the design. Thus it's effectiveness is reduced considerably. A need for a method was felt to reduce this time. With this background a preprocessor was developed to build standard full vehicle MADYMO model. In this preprocessor the user specifies the data about vehicle geometry, interior and general details in three template files in tabular format. The program takes input from these templates and generates the full vehicle MADYMO model. If the user prefers to use OPTIMIZATION, he can do so at this point. If the user prefers to use DOE, using some predefined parameters (fronthorn crush, kick-down buckling, shotgun stiffness, steering column stroke etc) then AutoDOE [5] link is available with the preprocessor, which generates the MADYMO input data files, based on the selected factors and DOE setup. The next step is then to run DOE i.e. all data files generated are submitted on various workstations and results are postprocessed. This modified process and method reduces the total analysis cycle time substantially (50%).

SUMMARY

The objective of this work was to find out the solution to the vehicle design problems using OPTIMIZATION and DOE on the full vehicle MADYMO concept model.

OPTIMIZER gave the global minimum and corresponding settings of different parameters, with minimal efforts and expertise. The method is good even for non-smooth functions.

DOE provided the information about main/parametric effects and the interaction of parameters on the response of interest. In this case, the results by both methods complement each other for reducing chest G's. Optimization techniques can even be applied to DOE analysis which yields the same results as direct OPTIMIZATION [6].

Methodology was established for generating full vehicle MADYMO concept model from vehicle design, for up-front safety analysis. This method can be applied to support the vehicle programs from up-front stage to final stage.

Up-front stage applications:

- Set targets
- Study trends
- Size components
- Evaluate various packaging and architectural alternatives.
- Lead the design

Final Stage Applications:

- Understand the sensitivity and interactions of various parameters on the response of interest.
- Excellent complement to FEA
- Crisis Situation - to brain storm and pro- vide directions

The method builds the base model quickly, reducing the analysis cycle time considerably and provides the detailed information, such as main effects and interaction effects, of various parameters on the response of interest. This provides the in-depth knowledge of how the overall system is behaving in the given design space. Thus all the possibilities are exploited from the beginning of the program.

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4. Bois A.C. and Lupkar H. A. "Design of Experiment in Occupant Simulation", International Congress and Exposition, Detroit, MI, Feb. 25, 1991.

5. AutoDOE, Software link to perform DOE's on CAE models. A product of StatDesign Inc., 23440, Kean, Dearborn, MI 48120, USA

6. Rajiv Pant and James Cheng "Integration of Concept/ Hybrid Models into Crash up-front Process with Optimization", SAE-VSM, March 1995.

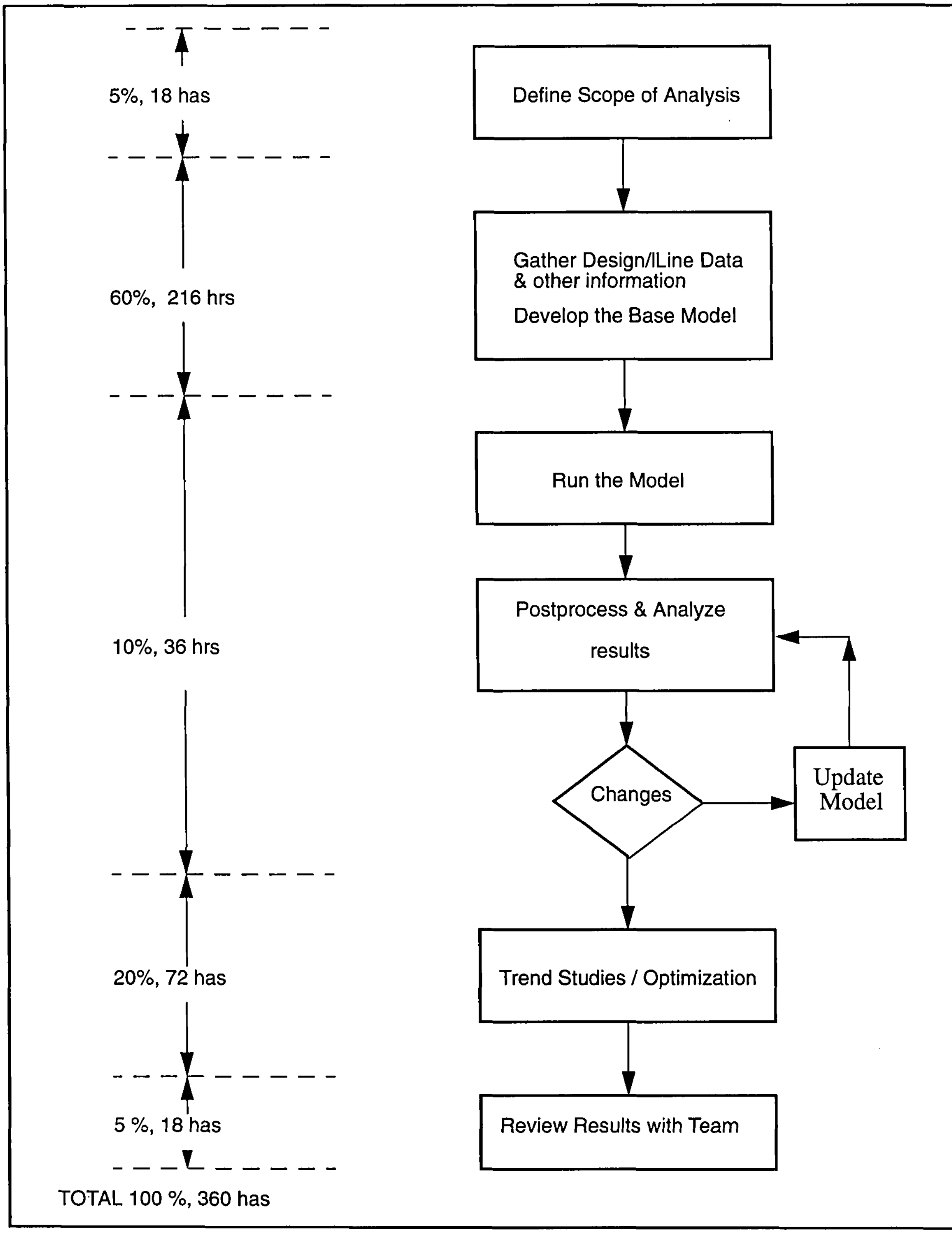


Figure 1. Typical Safety Analysis Cycle

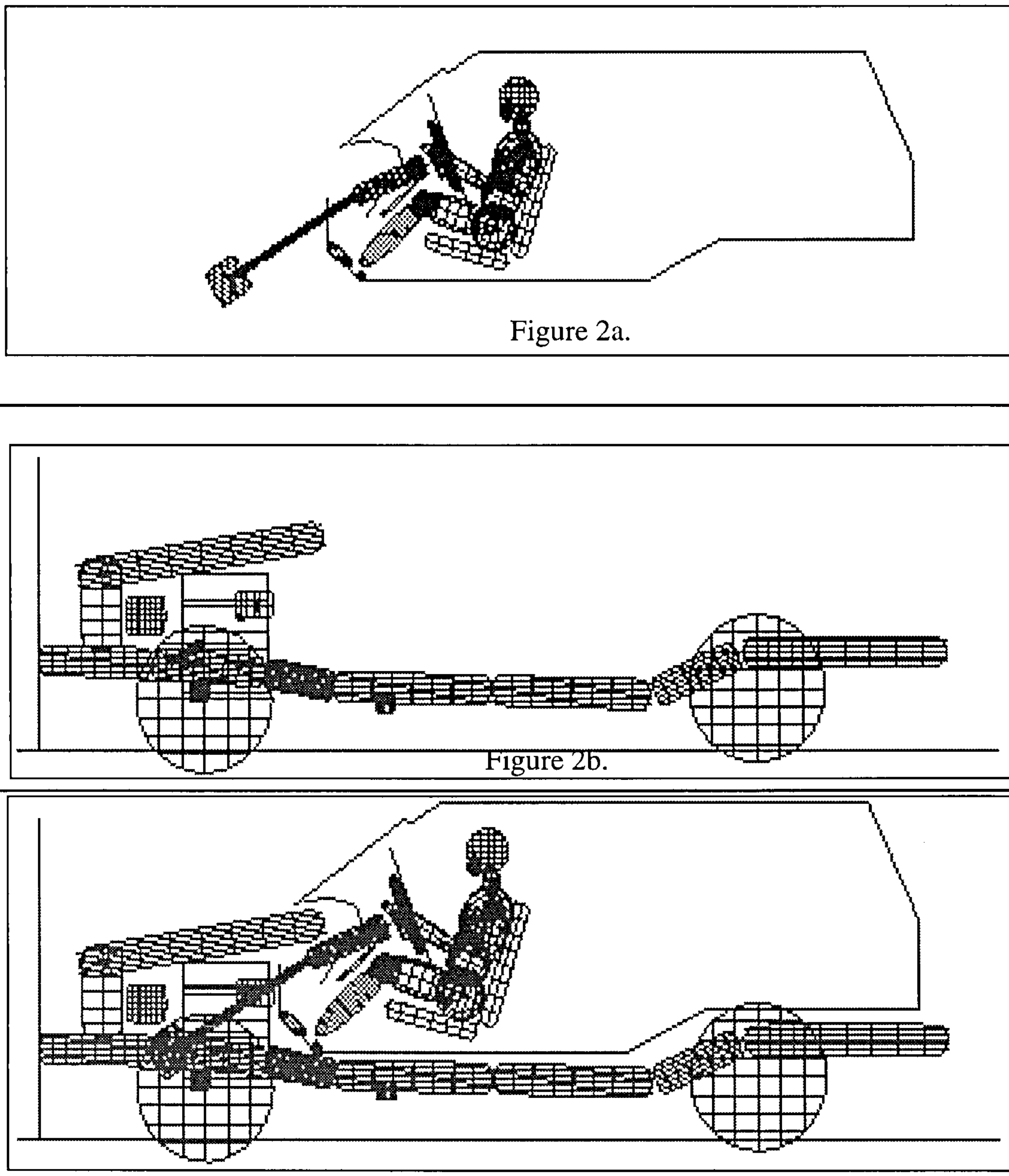
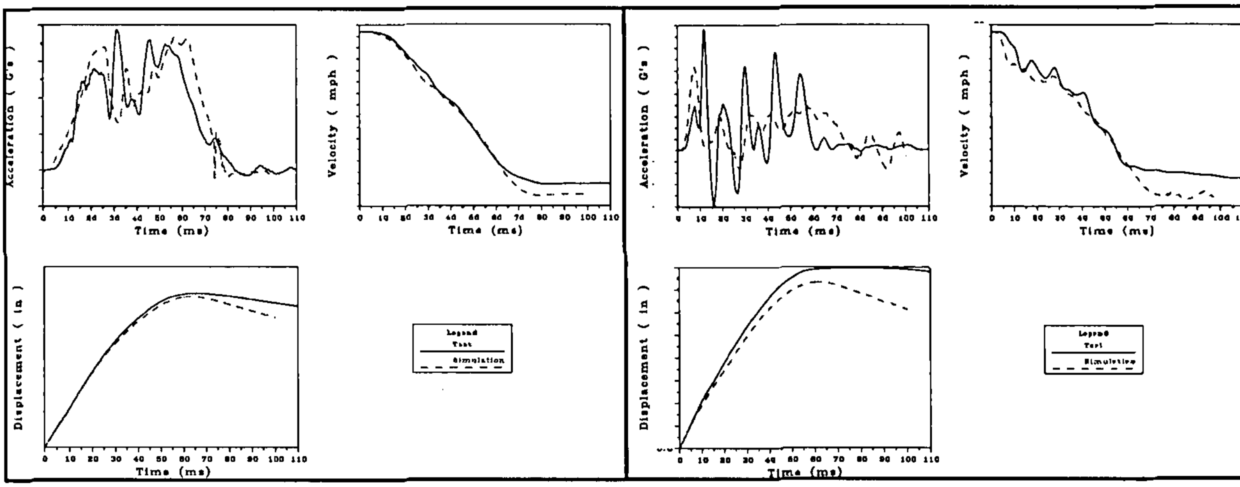


Figure 2a.

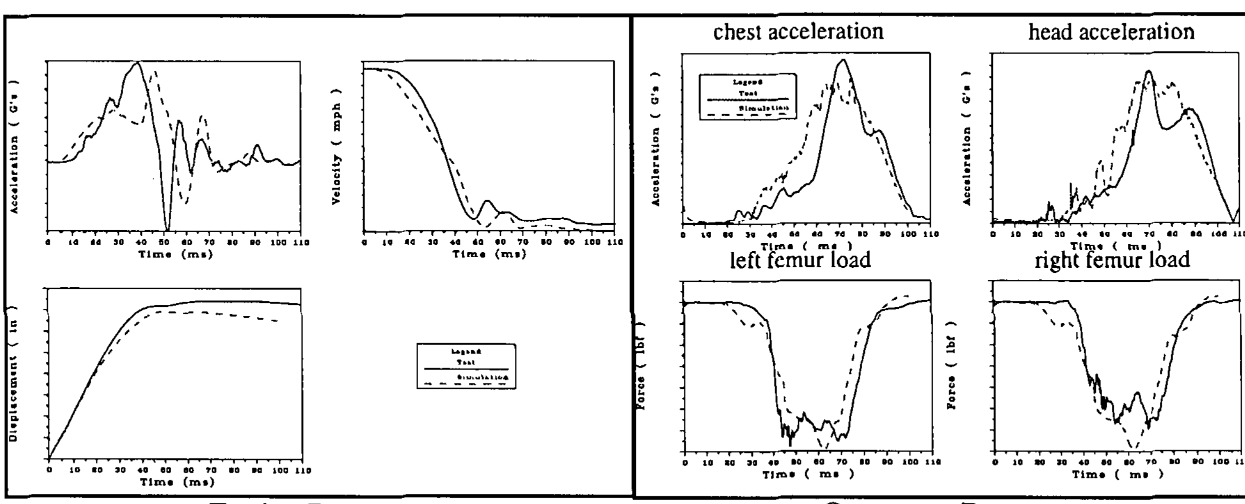
Figure 2b.

Figure 2. MADYMO model of body-on-frame type of vehicle



Rocker @ B-pillar

Frame @ B-pillar



Engine Response

Occupant Response

Figure 3. Occupant and Structural Response from Full Vehicle Model

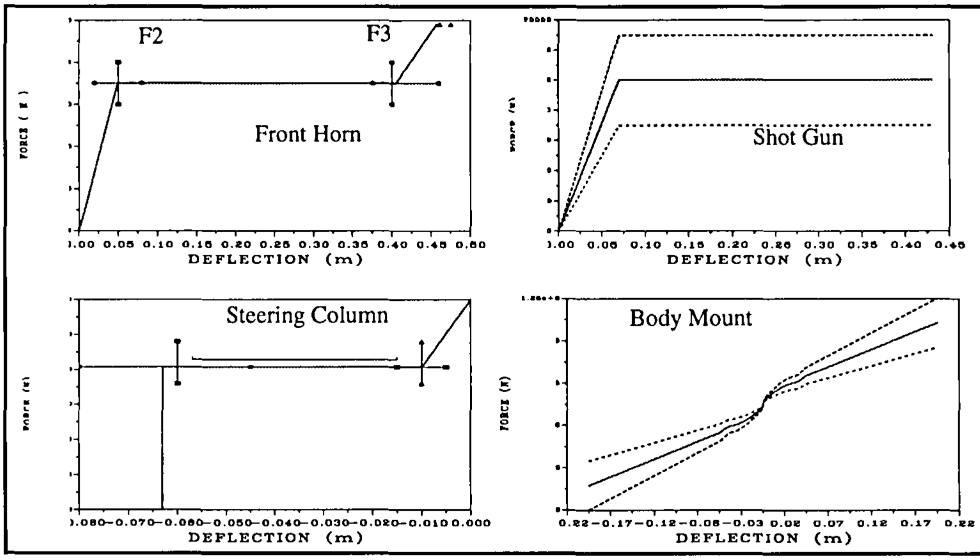


Figure 4. Settings of various factors for optimization

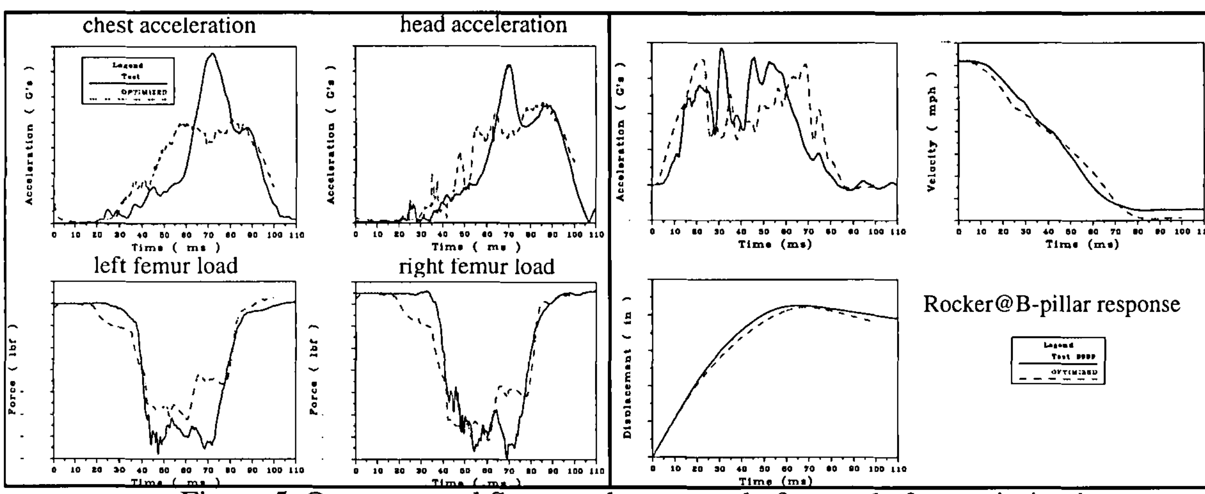


Figure 5. Occupant and Structural response before and after optimization

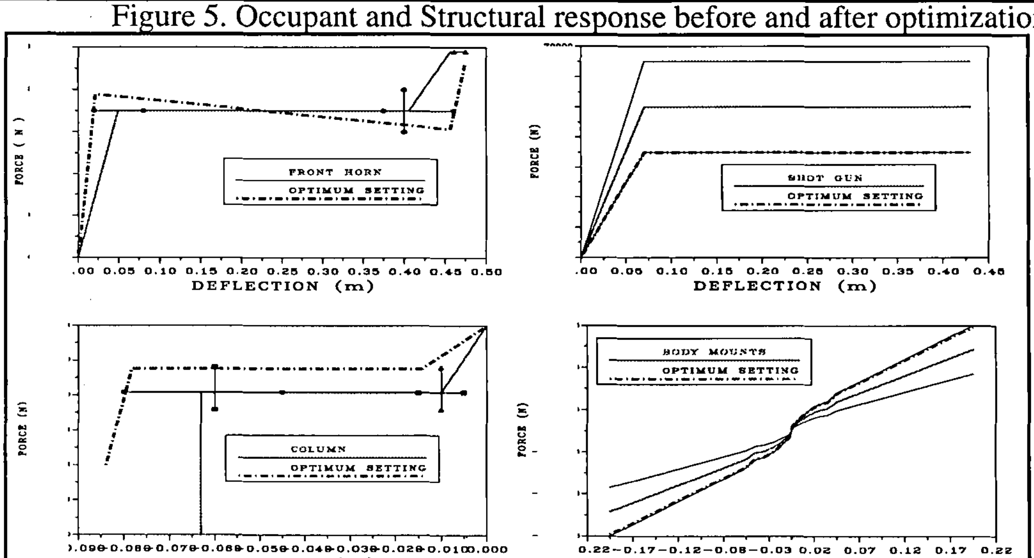


Figure 6. Optimum Settings for minimum chest G's

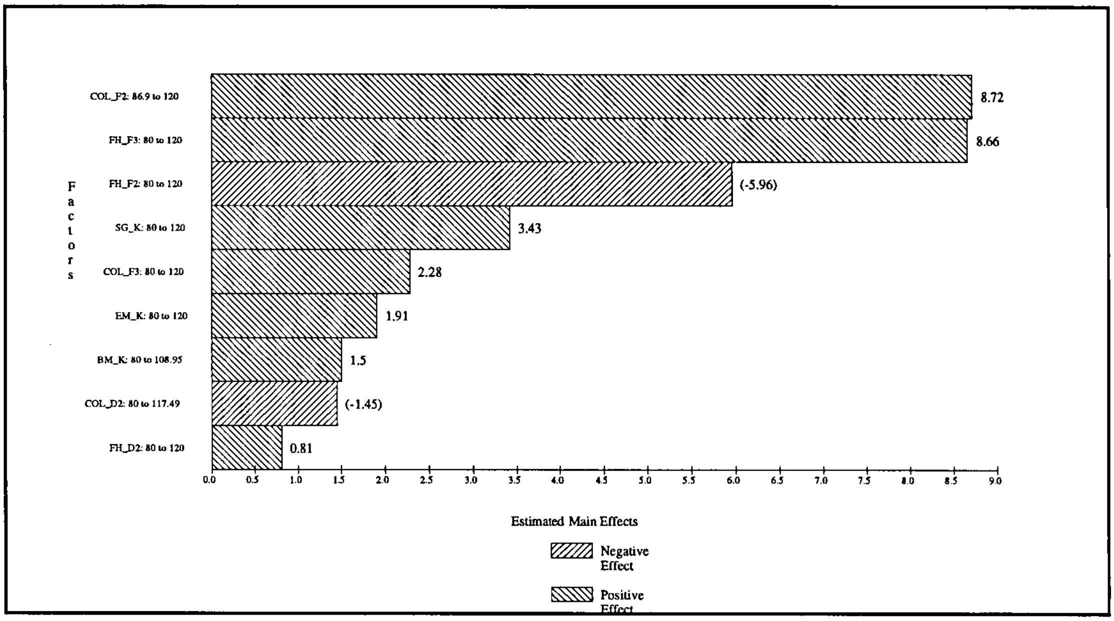


Figure 7. Main effect Plot

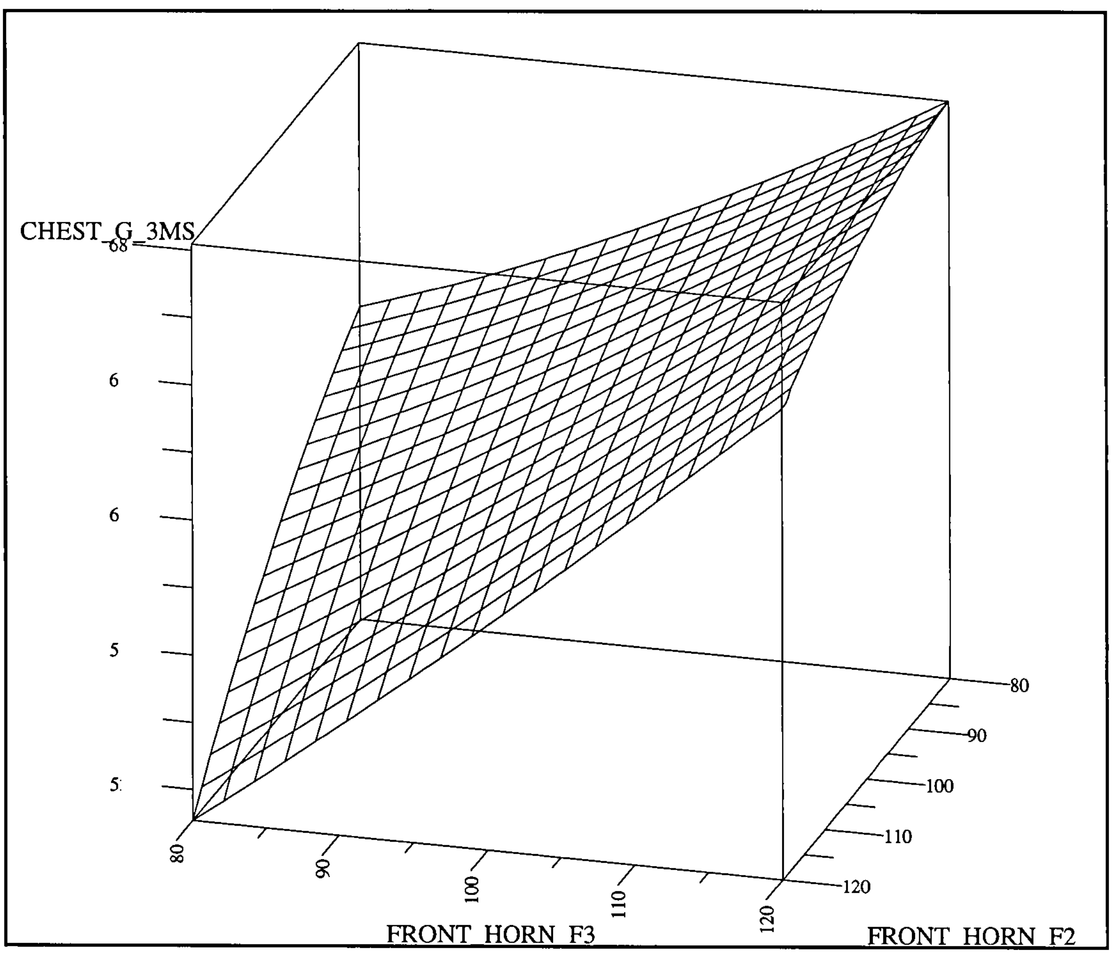


Figure 8. Interaction Plot

DEVELOPMENT OF DRIVER SIDE AIRBAG SIMULATION

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Chiharu Murase
Toyota System Research
Japan
Paper No. 96-S1-W-20

ABSTRACT

The chest of occupant who does not wear the seatbelt is not only restrained by the driver side airbag but also receives the reaction force applied by the steering wheel. This reaction force takes the form of more concentrated load than from the airbag. On the other hand, the larger chest deflection has some potential to cause the higher risk of the chest injury. Because the human rib cage is relatively compliant under a concentrated load, a concentrated load may have a tendency to cause a higher risk of the chest injury. It is estimated that the reaction force applied by the steering wheel to the test dummy has some increased potential to cause the chest deflection.

However, it is impossible to measure the reaction force that is applied by the steering wheel to the occupant's chest. For this reason, a driver side airbag simulation that uses a finite element model(FEM) to attempt to measure the reaction force applied by the steering wheel was developed.

The results of the developed simulation reveal that the contact force caused by interacting with the steering wheel and the occupant's chest is reduced and the risk of the chest injury is somewhat lower with deploying the airbag comparing to the case without deploying the airbag in certain circumstances.

INTRODUCTION

The chest of occupant who does not wear the seatbelt is restrained by the driver side airbag, it receives a reaction force that is applied by the airbag and the steering wheel. It is estimated that these reaction forces restrain the occupant's chest and bring it to a stop. On the contrary, if these reaction forces deflect the rib cage and cause the chest injury. Therefore, in order to develop a restraint system, it is important to understand the role of both the airbag and the steering wheel. The external force caused by interacting with the airbag and the rib cage is a distributed load. On the other hand, the external force caused by interacting with the lower part of the steering wheel and the rib cage is a more concentrated load than interacting with an airbag.

When compared to the case, in general a more distributed load is applied, the rigidity of the human rib cage is lower when a more concentrated load is applied [1]. Accordingly, the concentrated load has a tendency to cause a larger deflection of the rib cage. Also, larger rib cage deflection has a tendency to lead to the higher risk of chest injury. Because the external force caused by interacting with the lower part of the steering wheel and the rib cage is often a more concentrated load, the risk of chest injury has a tendency to be increased to some degree.

The chest of the HybridⅢ dummy contains a highly rigid sternum. Therefore, because the concentrated load applied to the sternum is distribute to the each ribs, it is highly likely that the tendency of the sternum to deflect would not be increased because of the concentrated load. Furthermore, the existing measuring devices cannot measure the concentrated load that is applied to the rib cage.

In contrast, a simulation using the finite element model and assumption can output a reaction force that is applied to the rib cage. Thus, the potential risk of chest injury caused by a concentrated load can be detected through the use of an FEM to measure both the distributed and the concentrated loads that are applied to the rib cage.

DEVELOPMENT OF FINITE ELEMENT MODEL

For the purpose of measuring the contact force caused by interacting with the steering wheel and the rib cage, the finite element model(FEM) to simulate the HYGE SLED tests that simulate a 30 mph frontal crash test was developed. The FEM comprises a Hybrid III dummy, a fully folded airbag, a steering wheel, a steering column, and interior parts. The Hybrid III dummy model contains a deformable rib cage, a neck and a lumbar spine.

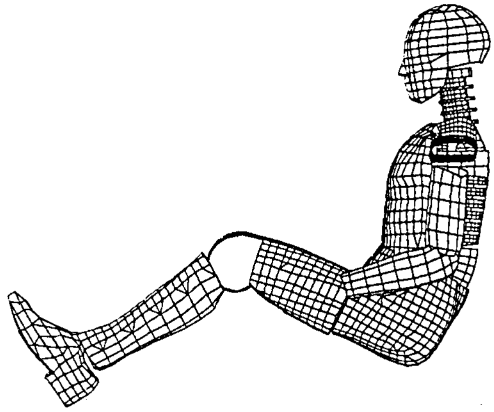


Figure1.HybridⅢ Dummy Model

HybridⅢ Dummy Model

In order to simulate the contact between the rib cage, the airbag and the steering wheel more realistically, 8-point brick and shell elements to model the rib cage, the neck, and the lumbar spine was used. Furthermore, in order to simulate the pelvic interfelence, the shape of the pelvis and the femurs was modeled. The remaining parts of the HybridⅢ dummy is modeled shell elements to simulate their shape and to represent the rigid elements. Figure 1 shows the finite element model of the Hybrid Ⅲ dummy.

Air Bag Model

To simulate the process of the initial stages of airbag deployment, the fully folded airbag was modeled as shown in Figure 2.

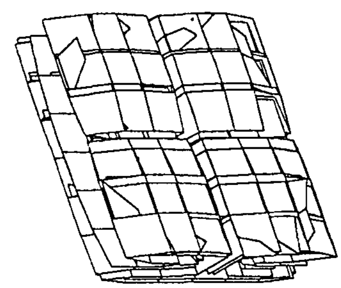


Figure2. Fully Folded Airbag Model

Steering Wheel Model

To simulate the large deformation of the steering wheel, the steering wheel was modeled as described below. Because the spoke portion is made of die-cast aluminum, it was modeled using 8-point brick elements. Because the steering wheel's rim portion is made of steel pipe, it was modeled using shell elements. The airbag case and the inflator was also modeled using shell elements. The steering wheel's exterior parts, which is made of urethane, was modeled using 8-point brick elements. Figure 3 shows the developed model of the steering wheel.

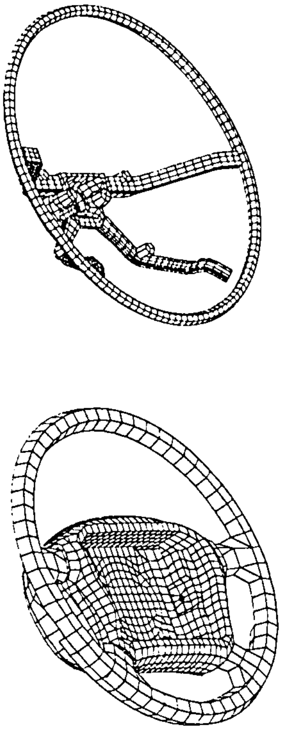


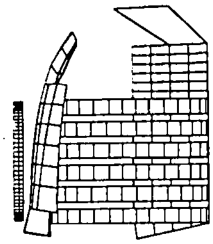
Figure3. Steering Wheel Model

ANALYSIS AND RESULTS

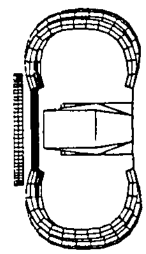
Chest Compression

To investigate the rigidity of the rib cage of the Hybrid III dummy under a distributed and a concentrated loads, two case of the chest compression analysis was conducted: one in which the rib cage of the Hybrid III dummy was compressed by a column measuring 152mm in diameter; another in which the rib cage was compressed by a horizontally laid column measuring 30mm in diameter, which simulates the lower portion of the steering wheel.

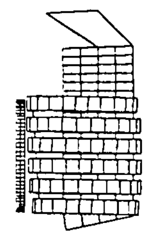
Figure 4 and 5 shows the simulation kinematics. Figure 6 shows the relationship between the reaction force and the deflection of the ribs. As shown in Figure 6, the rib cage of the Hybrid III dummy maintains a constant rigidity regardless of whether the load is a distributed or a concentrated load.



Rib Cage and Impactor

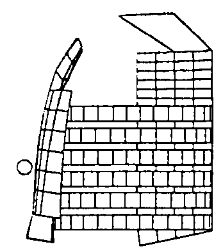


Deformed Shape [Plan View]

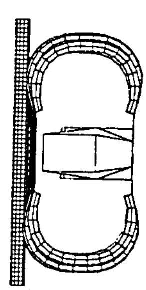


Deformed Shape [Side View]

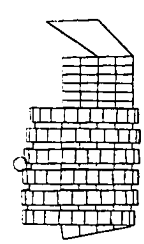
Figure 4. Thracic Compression Model and Simulation Kiremastics for 152 mm diameter cylinder.



Rib Cage and Impactor



Deformed Shape [Plan View]



Deformed Shape [Side View]

Figure 5. Thracic Compression Model and Simulation Kinematics for 30 mm diameter lateral laid cylinder.

As shown in Figure 5, the reason for this constant rigidity is that the sternum does not deform at all under a concentrated load, and the concentrated load applied to the sternum is distributed to the individual ribs. This is because the sternum of the Hybrid III dummy is extremely rigid. Thus, because the rigidity of the rib cage of the Hybrid III dummy would not decrease under a concentrated load, it is difficult that the experiments using the Hybrid III dummy detect the risk of chest injury caused by contact between the lower portion of the steering wheel and the ribs.

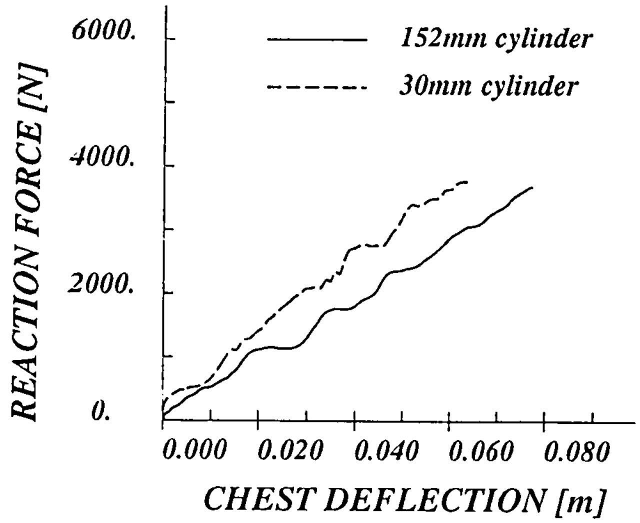


Figure 6. Comparison of Thoracic Stiffness Values between 152 mm Cylinder and 30 mm Lateral Laid Cylinder Compression.

Steering Impact

To investigate the extent of the concentrated load caused by interacting with the lower portion of the steering wheel and the ribs, a steering impact analysis was conducted. Figure 7 shows the analysis model and the simulation kinematics. Figure 8 shows the comparison between measured and calculated reaction force. As shown in Figure 8, the calculated result matches closely the experimental result. Figure 9 shows the horizontal compornent of the reaction force caused by interacting with the lower portion of the steering wheel and the body block. Figure 9 reveal that a reaction force of approximately 4 kN is applied to the body block by the lower portion of the steering wheel.

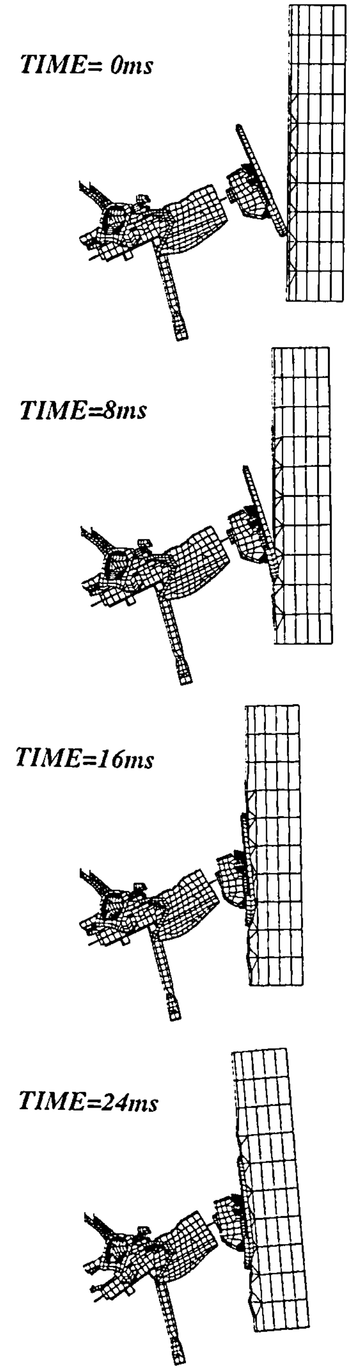


Figure 7. Analysis Model and Simulation Kinematics of the Steering Impact

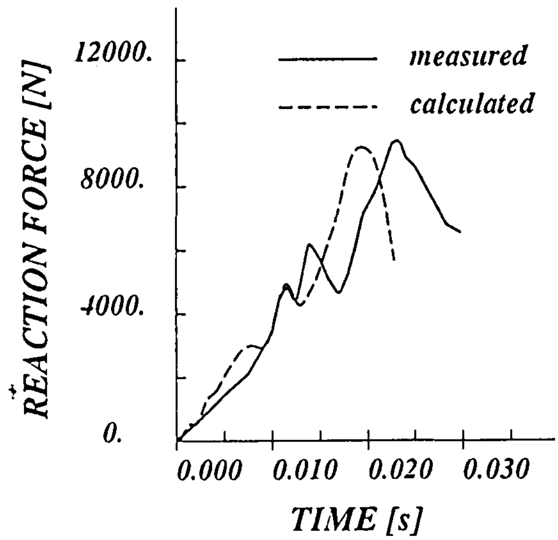


Figure 8. Comparison between Calculated and Measured Results of Reaction Force

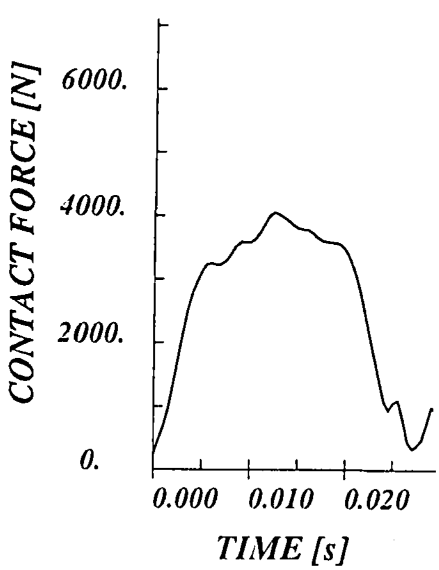


Figure 9. Reaction Force applied by Lower Part of the Steering Wheel to the Body Block

Hyge Sled Test

To investigate the reaction force that is applied by the lower portion of the steering wheel to the ribs of the occupant who is restrained by the airbag, the FEM analyses of HYGE SLED test was conducted. Figure 10 shows the comparison between measured and calculated chest acceleration and Figure 11 shows the simulation kinematics. It is revealed that the calculated chest acceleration matches closely the measured. As shown in Figure 12, only approximately 1 kN of reaction force is applied to the rib cage by interacting with the lower portion of the steering wheel. In other words, it is revealed that the reaction force applied by the lower portion of the steering wheel to the ribs of the occupant restrained by the airbag is approximately 3 kN lower than when an airbag is not used. Thus, when the occupant is restrained by an airbag, the risk of chest injury caused by interacting with the lower portion of the steering wheel and the ribs is decreased.

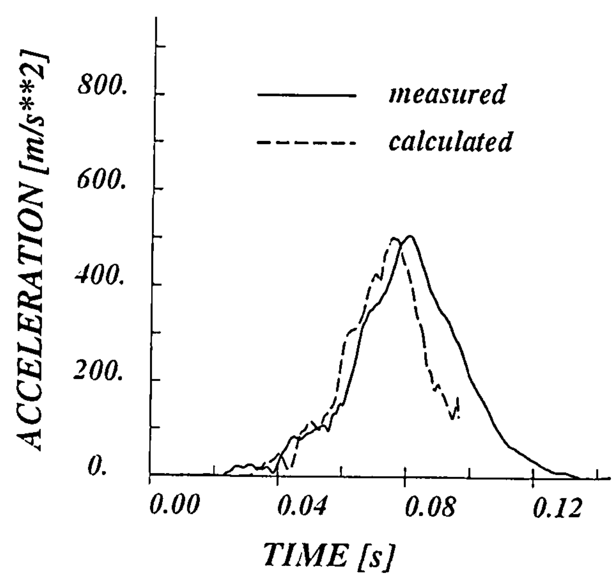


Figure 10. Comparison between Calculated and Measured Chest Acceleration.

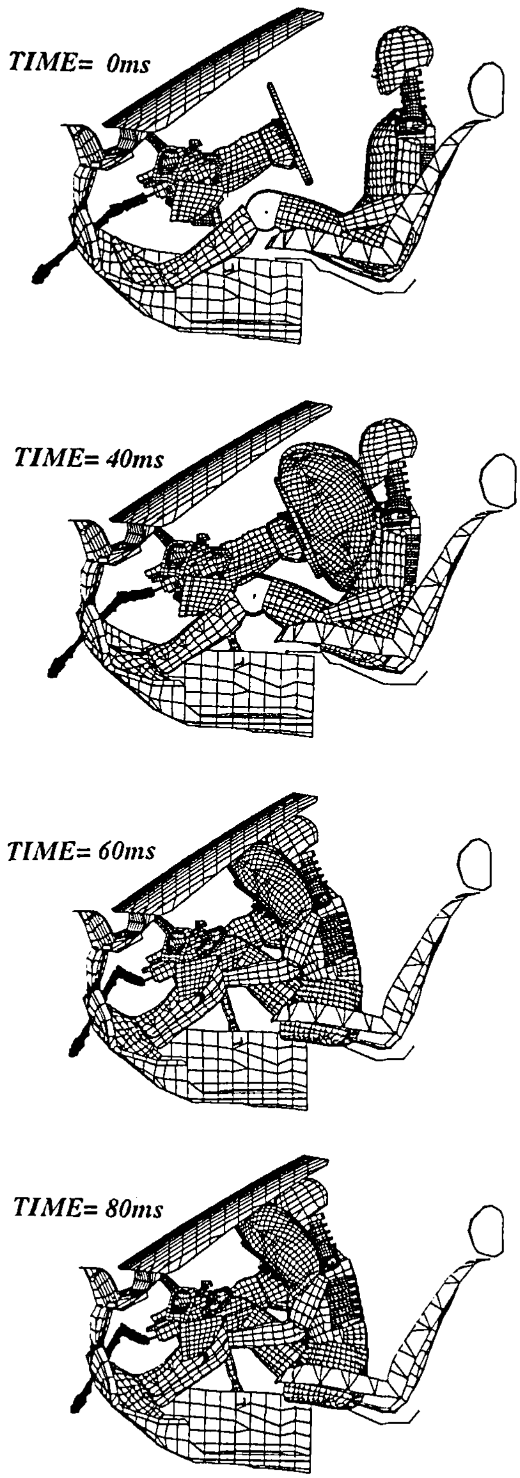


Figure 11. Simulation Kinematics of the HYGE SLED Test.

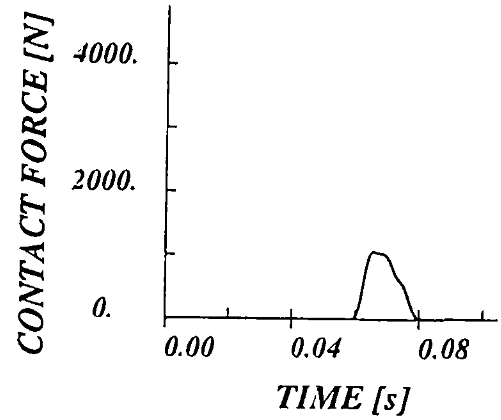


Figure 12. Contact Force Value between The Lower Part of Steering Wheel and the Rib Cage.

CONCLUSION

Because the Hybrid III dummy contains a rigid sternum, the sternum's compliance will not increase even if a concentrated load is applied to the rib cage.

Therefore, it is difficult that experiments using the Hybrid III dummy can be used to detect the risk of chest injury caused by the concentrated load that is caused by interacting with the lower portion of the steering wheel and the ribs.

If a steering impact test is conducted without deploying the airbag, approximately 4 kN of reaction force is applied to the body block by the lower portion of the steering wheel. However, in the HYGE SLED test in which the airbag has been deployed, a reaction force of only 1 kN is applied by the lower portion of the steering wheel to the ribs. Accordingly, there is a tendency that the airbag is deployed, the potential risk of chest injury caused by the concentrated load applied by the lower portion of the steering wheel to the ribs is lowered to some degree.

REFERENCE

[1]Lawrence W.Schneider, et al "Development of an Advanced ATD Thorax System for Improved Injury Assessment in Frontal Crash Environments",SAE 922520(1992)

OPTIMISATION OF THE WHEELCHAIR TIEDOWN AND OCCUPANT RESTRAINT SYSTEM (Effect of Diagonal Strap Anchorage Configurations on Occupant Restraint System)

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Paper Number 96-S1-W-21

ABSTRACT

To improve the safety of wheelchair occupants using both private and public transportation, effort is being focused through the International Standards Organisation (ISO) to produce standards for Wheelchair Tiedown and Occupant Restraint Systems (WTORS) (ISO/CD/10542) and Transportable Wheelchair Systems (TWS)(ISO/CD/7176-19). To ensure the structural integrity of the systems, these standards define a crash test protocol, which includes the use of a test wheelchair (ISO-W/C).

In the UK the diagonal strap of a lap and diagonal (L/D) wheelchair occupant restraint is either anchored to the floor of the vehicle or to the B pillar.

This paper reports a programme of frontal impact tests using the ISO-W/C test chair in a forward facing mode, carried out at the impact facility of Middlesex University, Road Safety Engineering Laboratory (MURSEL). The objective was to identify the influence of the diagonal strap anchorage position on the variation of loads in the WTORS.

Two computer models were built using MADYMO©* and DYNAMAN©** respectively, validated by the experimental results, to predict the occupant response to impacts and hence provide data to optimise future system design.

The results suggested that the 'B pillar' anchorage configuration was superior to the floor anchorage in that it exhibited reduced occupant shoulder loads and wheelchair front wheel loads.

INTRODUCTION

This paper describes part of the current research programme concerned with the crash performance of Wheelchair Tiedown and Occupant Restraint Systems (WTORS) and Transportable Wheelchair Systems (TWS) undertaken at Middlesex University Road Safety Engineering Laboratory (MURSEL).

* MADYMO© is the trademark of TNO Road-Vehicles Research Institute
** DYNAMAN© is the trademark of GESAC

Currently two standards are being developed by the International Standards Organisation Technical Committee 173/SC1/WG6. These draft standards for WTORS (ISO/CD 10542) and TWS (ISO/CD 7176-19) are primarily concerned with specifications and crash performance requirements. Previous experimental research in both Europe and North America has contributed to the format of the crash pulse contained in these standards, and is documented in several papers (Shaw et al, Schnieder L.W., Roy A.P., 1994; Roy A.P. and Stait E., 1995; Petzall J. and Olsson A., 1995).

In both the above draft standards a similar crash test protocol is employed. In the case of ISO/CD 10542, the WTORS is required to satisfactorily restrain a dummy seated on a surrogate electric wheelchair. This tes wheelchair will be coded throughout this paper as ISO W/C. In the case of ISO/CD 7176-19, it is a wheelchai submitted by a manufacturer which is the subject of th test. This will be given the generic code M-W/C an could subsequently be approved as a TWS.

Initial work by MURSEL using a TNO 10 dumm (Fig.1) has shown that an M-W/C exhibits less sever damage when the diagonal strap of the occupant restrain was anchored to the 'B pillar' at shoulder height rathe than anchored to the floor, for impacts of similar severity Analysis of video pictures taken from a Kodak EktaPr 1000 Motion analysis system suggested that the dummy crash dynamics were sensitive to changes in the diagonal top strap anchorage position.

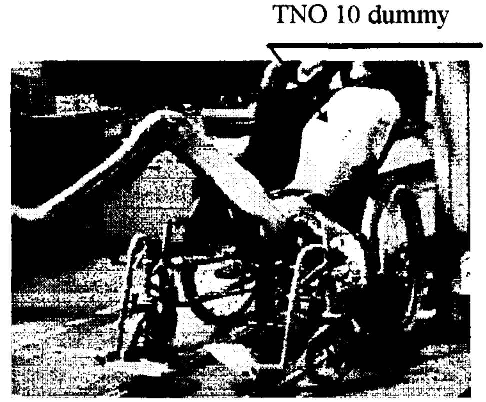


Fig.1 M-W/C 'B Pillar' anchorage - post test

This paper presents the results of an investigation into the variation of wheelchair and occupant loads as a function of diagonal top strap anchorage configurations, these being anchored to the B pillar (Fig.2) and the floor (Fig.3).

This investigation was carried out experimentally using the impact facilities at MURSEL and by computer simulation using MADYMO (CVS1) and DYNAMAN (CVS2).

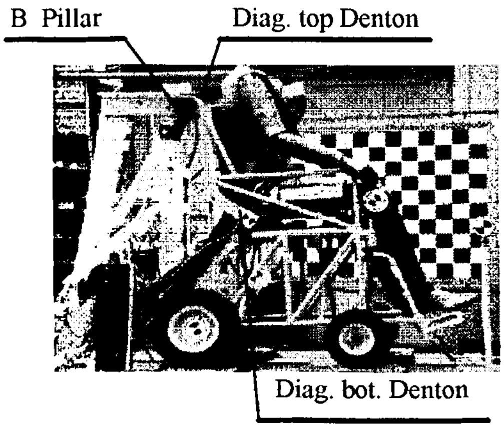


Fig. 2 B pillar anchorage of diagonal strap -ISO-W/C

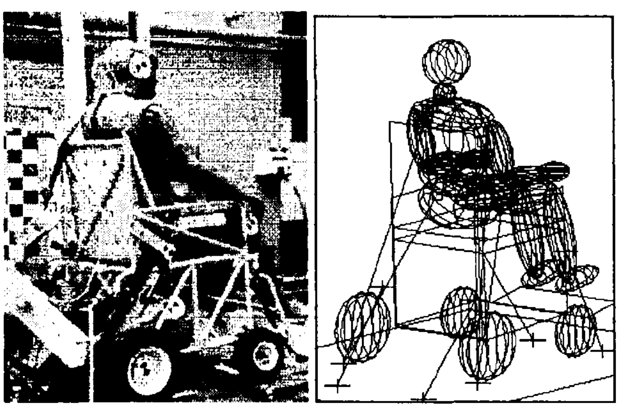


Fig. 3 Floor anchorage of diagonal strap - ISO-W/C

INSTALLATION OF EQUIPMENT ON SLED

Wheelchair

The ISO-W/C was used. This is a surrogate electric wheelchair of mass 85 kg. Its dimensions and structure were developed from a number of prototypes built in the UK by Transport Research Laboratory (TRL) and subsequently in the United States. Its mass distribution represents a typical mid range electric wheelchair. In order to perform satisfactorily over a number of impacts it is considerably stiffer and more robust than a production model. Also its seat is approximately 50 mm higher and thus reproduces an undeflected cushion. However with this CG position it is considered to represent a worst case.

Tiedowns

The wheelchair was attached to the sled by four webbing tiedowns at the rear and two proprietary straps at the front. The front tiedowns locked into rails at an angle of approximately 45 degrees to the horizontal. The rear tiedowns were initially set at 45 degrees to the horizontal and anchored to the sled using shackles. The end remote from the floor was looped round a vertical structural member. This configuration was described elsewhere (Gu J., Roy A.P., 1995).

Occupant Restraint

A Hybrid II adult dummy (mass 75 kg) was used to represent the occupant. The dummy was restrained by a static lap and diagonal belt. The diagonal top strap was anchored to the B pillar or the floor.

Transducers

MURSEL type K9 tensile load cells were attached to each rear tiedown strap to measure the loads. One K9 measured the L/D buckle strap tension (T_4), whilst Denton webbing load cells were attached to both ends of the shoulder belt. The diagonal top strap tension (Diag. top)(T_1) and that just above the buckle (Diag. bot.)(T_2) were measured directly. The lap strap tension (T_3) at the junction with the buckle was estimated as (T_4 - T_2).

The front port and starboard wheel loads (FP wheel, FS wheel) and rear wheel (RP wheel and RS wheel) were measured using cantilever load cells under each wheel (Fig. 4).

Fig. 4 Cantilever load cells

The occupant seat loads in the ISO wheelchair were measured using a pancake load cell at each corner of the seat plate (Fig. 5).

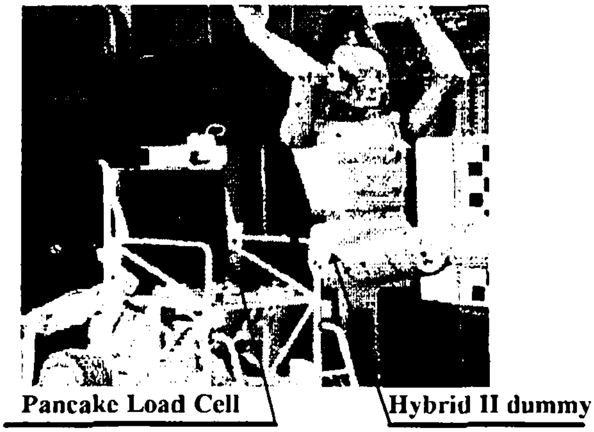


Fig. 5 Pancake load cells built onto the seat plate of the ISO-W/C

The Hybrid II dummy chest accelerations were measured using a standard triaxial Endevco 7267A accelerometer.

Data Acquisition Equipment

A full description of the data acquisition equipment appears elsewhere (Roy A. P., 1990).

IMPACT PROGRAMME

The MURSEL impact facility was used to subject the ISO-W/C and WTORS to a programme of frontal impacts of severity indicated below (Fig. 6). The crash pulse could typically be characterised by velocity change (delta V) and sled deceleration (g). In order to preserve the structural integrity of the test wheelchair and the measurement devices, sled testing was carried out at three levels combined with appropriate velocity change to examine the effect on the shoulder load function. The wheel loads were measured in the case of Level I and Level II tests only.

Level I: 6 -- 10 g,	delta 'V'=15 -- 25 km/h
Level II: 11 g,	delta 'V' = 27 km/h
Level III: 13 -- 21 g,	delta 'V'= 34 -- 51 km/h

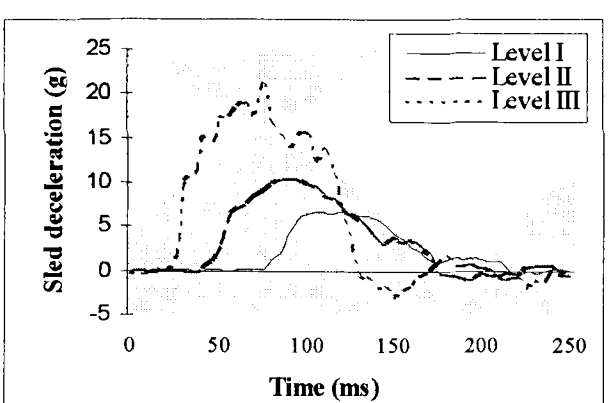


Fig. 6 Sled deceleration pulses in WTORS testing

RESTRAINT SYSTEM DYNAMICS

In order to understand the dynamics of the restraint system, it is appropriate to remember that it consists of two parts:
(1) The ISO-W/C chair secured by four rear tiedowns and two front tiedowns;
(2) The Hybrid II dummy restrained by a L/D occupant restraint.

Both of the restraint systems were independent of each other and anchored separately to the sled. The dummy sat on an aluminium alloy plate, which was placed on the four pancake load cells. The load cells were themselves bolted to the wheelchair frame. In general the movement of the system consisted of the following three phases:

__Phase 1__ - The dummy slid across the plate essentially in the horizontal plane whilst the effect of the rear tiedowns was to compress the rear tyres and rotate the rear pancake load cells downwards. Thus the rear of the wheelchair frame accelerated down. This was verified by the reduction of the loads monitored by the rear pancake load cells. The wheelchair front wheels lifted off the sled floor (Fig 7).

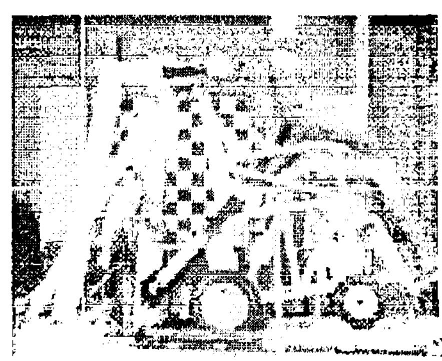


Fig. 7

__Phase 2__ - The dummy continued to slide horizontally and loaded the occupant restraint straps whilst the wheelchair rear tyres recovered and the front wheels moved down onto the sled and compressed (Fig 8).

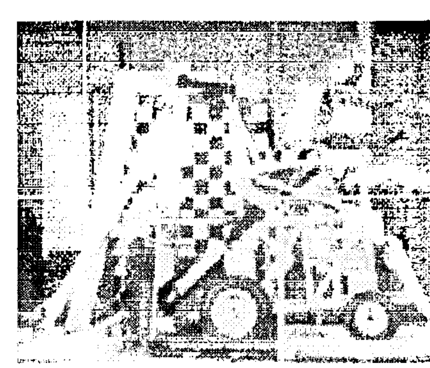


Fig. 8

__Phase 3__ - The dummy reached its furthest forward movement, the tensions in the occupant restraint straps reached their maximum values, and weight transfer by the dummy to the front pancake load cells and the front wheels occurred. Finally the cantilever load cells under the front wheels exhibited an increased value as the front tyres started to recover and rebound commenced (Fig 9)

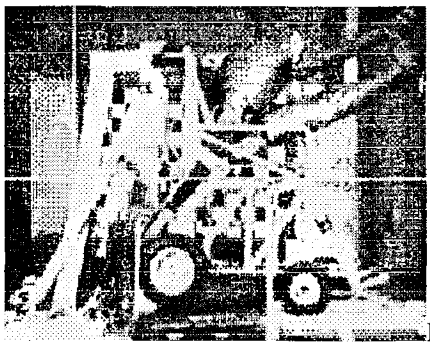


Fig. 9

EXPERIMENTAL RESULTS

A test protocol for the WTORS testing was developed based on the results of twenty five sled tests. Tables 1 - 2 are summary statistics for the dynamic test results. The peak values are given for each parameter. The difference (diff.%) indicates the deviation from B pillar configuration expressed as a percentage. The values of front and rear seat are each a total of two load cells. The wheel sums are each a total of the four appropriate transducers.

Table 1
Peak Values Measured
(Level I: 6 g, 15 km/h)

Configurations		B pillar	Floor	diff.%
Parameters	units			
Chest Res.	g	12.23	11.40	6.8
T_1	kN	2.50	3.10	24.0
T_2	kN	1.70	1.40	17.6
T_3	kN	1.86	1.32	29.0
Front seat	kN	4.04	3.20	20.8
Rear seat	kN	4.31	3.60	16.5
FP wheel	kN	1.70	4.90	188.2
FS wheel	kN	1.80	3.90	116.7
RP wheel	kN	11.40	10.80	5.3
RS wheel	kN	11.50	10.70	6.9
Wheel Sum	kN	26.40	30.30	14.8

Table 2
Peak Values Measured
(Level II: 11 g, 27 km/h)

Configurations		B pillar	Floor	diff.%
Parameters	units			
Chest Res.	g	21.40	19.00	11.2
T_1	kN	4.10	4.50	9.7
T_2	kN	3.00	2.10	30.0
T_3	kN	3.38	3.27	3.3
Front seat	kN	6.01	5.80	3.5
Rear seat	kN	6.47	6.20	4.2
FP wheel	kN	2.30	4.70	104.4
FS wheel	kN	0.90	3.50	288.9
RP wheel	kN	23.30	22.70	2.6
RS wheel	kN	19.90	20.60	3.5
Wheel Sum	kN	46.40	51.50	11.0

The dynamic testing of Level III (B pillar only configuration) concentrated on the investigation of the seat load distribution (Table 3).

Table 3
Peak Values Measured
(Level III: 13 - 21 g, 34 - 51 km/h)

Delta ' V'	km/h	34	40	45	51
Sled pulse	g	13	16	17	21
OUTPUT:					
Chest Res.	g	49.10	34.00	39.94	46.6
T_1	kN	4.77	5.41	5.98	6.5
T_2	kN	3.78	4.38	4.65	5.0
T_3	kN	4.10	4.90	5.20	6.1
FP seat	kN	4.80	5.10	6.50	7.7
FS seat	kN	5.90	5.80	6.60	8.3
RP seat	kN	3.90	4.40	5.40	5.9
RS seat	kN	3.40	4.40	4.10	4.5
Seat Sum	kN	18.00	19.70	22.60	26.4

In order to compare the crash performance of the two anchorage positions, the following parameters were considered.

- **ΔV:** Sled velocity change;
- **t:** Time of peak Diag. top strap tension (T1);
- **α:** Diag. top strap angle to the horizontal at the time of T_1;
- **β:** Occupant torso forward angle from vertical at the time of T_1;
- **γ:** Diag. top strap angle with reference to a vertical plane parallel to the sled fore and aft centre line;
- **H_{exc}:** Dummy head target max. excursion;
- **T_1:** Diagonal top strap tension;
- **T_2:** Diagonal bottom strap tension;
- **T_3:** Lap strap tension;

T_4: Buckle strap tension;
C_f: Occupant seat sum load;
$S_f(B)$: Occupant shoulder load function in B pillar configuration,
$S_f(F)$: Occupant shoulder load function in floor configuration.

The shoulder load functions in the two configurations are defined as follows:

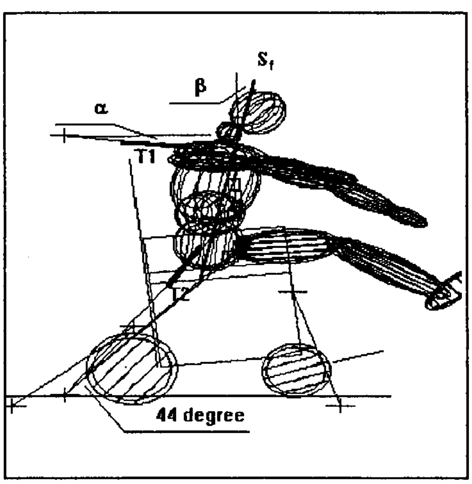
$$S_f\,(B) = T_1 \times \cos\gamma \,\sin(\beta - \alpha) + T_2 \times \sin(44 + \beta) \tag{1}$$

$$S_f\,(F) = T_1 \times \cos\gamma \,\sin(\beta + \alpha) + T_2 \times \sin(44 + \beta) \tag{2}$$

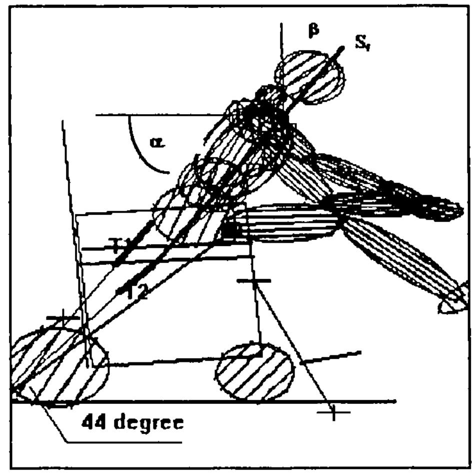
The occupant shoulder load functions were computed in order to try to obtain a value of the downward load on the shoulders of the dummy, in the direction of the torso centre line, applied by the diagonal straps of the occupant restraint.

It should be noted that T_1 did not lie in a vertical plane parallel to the sled centre line, whilst T_2, T_3 and T_4 did.

The dummy shoulder loads were estimated as indicated below and the results are listed in Tables 4 and 5. It is difficult to measure the dummy forward angle (β) as the thorax of the dummy twisted during the impact. Some assumptions were made to simplify the model. The same angle of 44 degrees to the horizontal was assumed for both the diagonal bottom strap and lap belts. The angle γ. was estimated from the EktaPro records and the initial setting of the Hybrid II.



(1) B pillar Configuration



(2) Floor Configuration

Fig. 10 Free body diagram for occupant torso

Table 4
Measured Parameters in B pillar Configuration
(at the time of peak T_1 load)

ΔV	t	α	β	γ	H_{exc}	T_1	T_2	T_3	C_f	$S_f(B)$
km/h	ms	deg	deg	deg	mm	kN	kN	kN	kN	kN
15	165	5	4	17	168	2.50	1.63	1.40	4.54	1.17
18	145	5	4	10	216	2.99	2.23	1.40	4.72	1.61
20	135	5	10	13	254	3.67	2.87	2.18	5.63	2.63
23	125	5	10	8	256	3.90	2.72	2.30	6.53	2.54
25	125	5	16	8	269	3.92	2.90	2.82	8.46	3.25
27	125	5	16	8	272	4.10	2.77	3.28	9.78	3.17
34	120	5	16	8	296	4.77	2.57	3.56	11.3	3.13
40	115	8	20	6	304	5.41	2.69	3.85	11.8	3.54
45	95	10	22	31	369	5.98	4.57	5.39	12.6	5.24
51	95	10	30	32	384	6.51	4.01	6.24	14.7	5.74

Table 5
Measured Parameters in Floor Configuration
(at the time of peak T_1 load)

ΔV	t	α	β	γ	H_{exc}	T_1	T_2	T_3	C_f	$S_f(F)$
km/h	ms	deg	deg	deg	mm	kN	kN	kN	kN	kN
15	210	28	10	<1	272	3.02	0.68	0.68	4.59	2.41
18	200	23	15	<3	312	3.17	0.85	0.79	4.87	2.68
20	190	23	16	<2	360	3.73	1.03	0.93	5.60	3.24
23	175	23	20	<1	360	3.79	1.33	0.87	6.63	3.78
25	155	20	22	<2	368	4.04	1.21	1.76	8.01	3.81
27	155	20	28	4	392	4.18	1.43	1.67	8.77	4.46
34	145	15	35	6	450	5.89	1.68	1.75	9.11	6.14

OPTIMISATION OF WTORS IN COMPUTER MODEL

The segment structure model of the WTORS and ISO-W/C was also developed using CVS simulation with ellipsoids. Two CVS models (CVS1 and CVS2) are presented using the same configurations but different CVS methods.

Initially static tests were carried out to find the load deflection characteristics of all parts of the wheelchair. Force-deflection characteristics for contacts and belt interaction had been measured statically (Interim reports by MURSEL). Calibration of webbing tension against strain had been obtained previously using an Avery tensile testing machine in order to define the contact function.

A dummy model based on the Hybrid II was used to represent the seated occupant in a wheelchair. The WTORS model was run several times whilst varying the diagonal top strap anchorage location, the belt stiffness and the sled deceleration pulse.

DYNAMIC TEST DATA ANALYSIS

Occupant Restraint and Seat Loads as a Function of Time

Fig. 11 shows that for the B pillar configuration the peak values of buckle strap tension (T_4), both diagonal top and bottom tensions (T_1 and T_2) occur around a time of 120 ms to 130 ms. Fig 12 shows that for the floor mounted configuration, the peak value of T_1 is reached at the time of 170 ms. It lagged the peak seat sum (C_f) and T_4 by about 35 ms. This delayed response of T_1 was supported by observations from the EktaPro video record of the greater forward displacement of the dummy torso, thus causing weight transfer to the front of the chair. The greater value of the front wheel loads for the floor anchorage configuration (Table 1-3) also supports this observation.

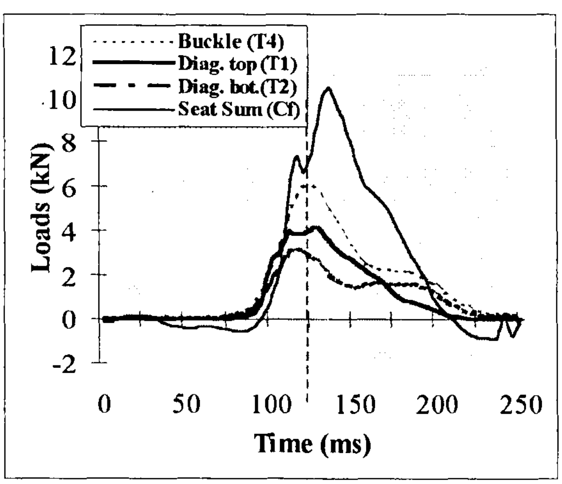


Fig. 11 Load distribution in B pillar configuration (11g, 27 km/h)

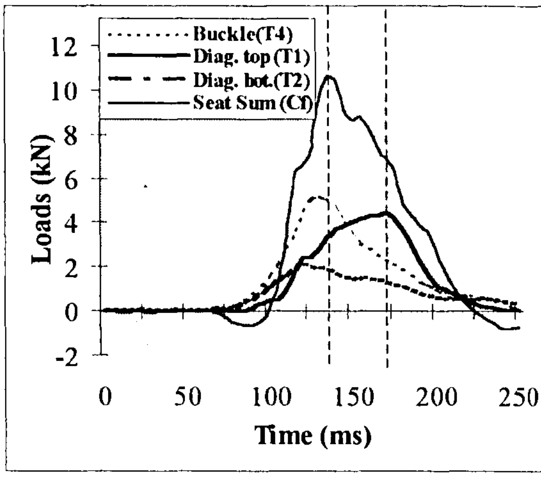


Fig. 12 Load distribution in floor configuration (11g, 27 km/h)

Occupant Restraint and Seat Loads as a Function of Sled Velocity Change (Delta V)

Fig 13 and Fig 14 suggest that the peak loads such as overall seat (C_f), diag. top (T_1), lap belt (T_3), and shoulder load functions [S_f(F) and S_f(B)] generally increase with the velocity change in both configurations.

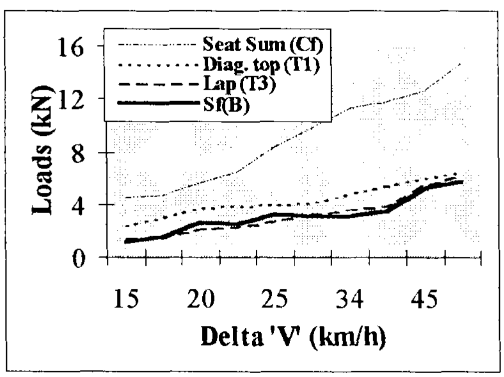


Fig. 13 Peak Parameter Variation - Delta V (B pillar configuration)

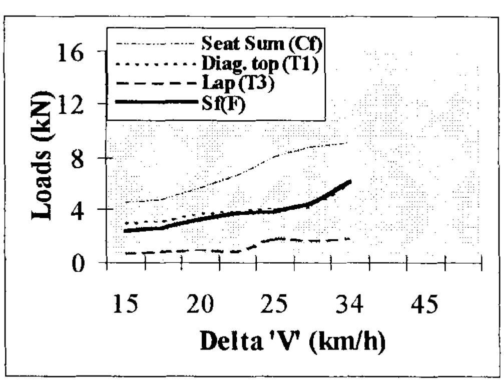


Fig. 14 Peak Parameter Variation - Delta V (Floor configuration)

DISCUSSION OF TEST RESULTS

Comparison of the Effect of Diagonal Top Strap Anchorage Configuration

Tables 1 and 2 show that the front wheel loads were more sensitive to the anchorage configurations than the other parameters. In general the shoulder load increased as the velocity change increased (Fig.15). The floor mounted configuration always produced higher values of front wheel loads and diagonal top strap tensions at a given delta V.

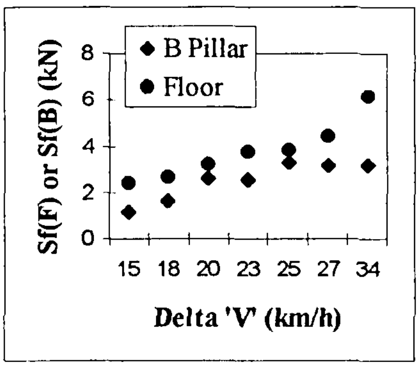


Fig. 15 Comparison of peak shoulder load function in two configurations

As the velocity change increased it would be expected that γ decreased. However above 27 km/h (floor anchorage) and 45 km/h (B pillar anchorage) the dummy shoulder twisted and the dummy moved laterally thus giving unexpectedly high values of γ.

Tables 6 and 7 show that for the floor mounted configuration the peak time of T_1 lagged the B pillar by 35 - 45 ms.

Table 6
Load Duration (B pillar)- Level I, II

Parameters	Units	Load duration (ms)		
		Range	Period	Load
Chest Res.	g	175--100	75	12.2--39.6
Diag. top (T_1)	kN	165--110	55	2.5--4.5
Lap (T_3)	kN	165--110	55	1.9--4.4
Buckle (T_4)	kN	165--105	60	3.5--7.8
Seat Sum (C_f)	kN	175--115	60	6.5--12
Wheel sum	kN	145--135	10	42.2--43.3

Table 7
Load Duration (Floor mounted) - Level I, II

Parameters	Units	Load duration (ms)		
		Range	Period	Load
Chest Res.	g	165--100	65	11.4-29.5
Diag. top (T_1)	kN	210--145	35	3 -- 5.9
Lap (T_3)	kN	175--110	55	1.4--4.2
Buckle (T_4)	kN	170--110	60	2.7-- 6.9
Seat Sum (C_f)	kN	190--115	75	6.6--11.2
Wheel sum	kN	190--135	55	27.5 -45.5

The shoulder load function difference in the two configurations is summarised in Table 8.

Table 8
Shoulder Load Function Difference

Delta V	S_f(B)	S_f(F)	diff.
km/h	kN	kN	%
15	1.17	2.41	106.0
18	1.61	2.68	66.5
20	2.63	3.24	23.2
23	2.54	3.78	48.8
25	3.25	3.81	17.2
27	3.17	4.46	40.7
34	3.13	6.14	96.2

Comparison of the Effect of Wheelchair Structure

The peak values of seat loads in the ISO-W/C were considerably higher than those in the M-W/C. The shaded portion in Fig 16 shows the intersection area of the two wheelchairs. This suggests that the seat cushion in the M-W/C absorbed some of the energy from impact and reduced the peak seat loads.

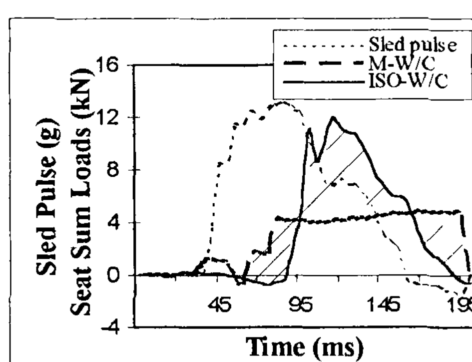


Fig. 16 Seat sum loads for B pillar configuration - Delta V =31 km/h

Comparison of the Effect of Sled Crash Pulses

It can be seen from the test results displayed in Tables 1- 2 that the front wheel loads varied considerably as a function of crash pulse. The test results suggested that weight transfer to the front wheels relatively increased the front wheel loads.

COMPATIBILITY OF DYNAMIC TEST AND CVS MODEL RESULTS

As seen from Table 9, most of the simulated results obtained were in general agreement with the dynamic test results. The simulated diagonal top strap tension in CVS2 model mirrored the test results (Fig. 17).

Table 9
The Difference between Test and CVS Model Results (Peak values, B pillar, 21g, 51 km/h)

Output	units	Test	CVS 1	CVS 2	diff 1 %	diff 2 %
Chest -x	g	38.30	26.30	32.70	31	15
Chest Res.	g	46.60	38.10	53.00	18	14
Diag. top (T_1)	kN	6.90	7.20	7.90	4	14
Diag. bot (T_2)	kN	5.00	5.50	7.50	10	50
Lap (T_3)	kN	6.10	7.20	6.30	18	3
Buckle (T_4)	kN	11.10	14.30	13.80	29	24
Seat Sum (C_f)	kN	26.40	26.30	34.70	0	31

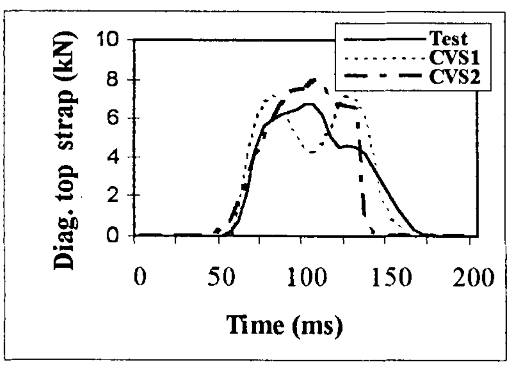


Fig. 17 Diagonal top strap tension - CVS and experimental value (B pillar, 21 g 51 km/h)

All peak strap tensions in the CVS models were generally higher than the test results and had a relatively sharper pattern. The different contact functions of the modelled and actual wheels had a significant effect on this difference. Furthermore the model did not allow for the effect of belt slippage which was evident in the experimental results.

CONCLUSIONS

This impact programme used the ISO-W/C (a surrogate for a mid-range electric wheelchair) which was more robust and stiffer than the M-W/C (a production chair). In addition the Hybrid II was seated on an aluminium plate above pancake load cells. Thus the CG was higher and the seat absorbed little energy when compared with a soft cushion and the more flexible structure of the M-W/C in use. Therefore the peak loads would be expected to be higher than in the real world. The results of an initial small test programme at MURSEL using an M-W/C chair supported this view.

In order to avoid the onset of structural damage to the ISO-W/C the impacts in the floor anchored occupant shoulder belt configuration ceased at a sled velocity change of 34 km/h. For the B pillar configuration it was increased to 51 km/h (2% above the CD10542 maximum).

- The diagonal top strap anchorage configurations had a considerable effect on the dynamics of the system and the values of the diagonal strap tensions and front wheel loads.
- At all values of sled velocity change the floor mounted configuration exhibited a peak shoulder load function greater than that for the B pillar configured system. At a velocity change of 34 km/h the value (6.14 kN) was higher by 96 %. The value at 51 km/h for the B pillar configuration was lower (5.74 kN).
- At all values of sled velocity change the floor mounted configuration exhibited a maximum dummy head target excursion greater than that for the B pillar configured system. At a velocity change of 34 km/h the value (450 mm) was higher by 52%. This value was not reached by the latter system; the value at 51 km/h was 384 mm.
- The front wheel loads exhibited similar variations. They indicated the weight transfer to the front of the wheelchair as the maximum head target forward excursion was reached.
- In general the peak seat load sum was greater for the B pillar than the floor mounted configuration. However the loading phase for the former acted over a shorter period.
- The CVS models generated higher loads than those generated in the sled tests, except in the case of the resultant chest deceleration. The increase of diag. top tension (T_1) is in the order of 14% of the experimental value. The CVS1 model replicated the bouncing behaviour of the ISO-W/C.
- Taking into account the implications of the above conclusions on the occupant and the wheelchair it is considered that the B pillar anchorage of the occupant diagonal strap is superior to the floor mounting.

ACKNOWLEDGEMENTS

The authors wish to acknowledge the help of Mr G T Savage and Mr C T Witherington in the performance of the MURSEL dynamic tests. Some of the work described in this paper forms part of a research programme

supervised by Edward Stait of the Department of Transport. The views expressed are not necessarily those of the Department.

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BREED TEMPERATURE COMPENSATED STORED GAS INFLATOR: THE ONLY TRUE GREEN SOLUTION

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Paper Number 96-S1-W-23

ABSTRACT

The Temperature Compensated Stored Gas (TCSG) inflator provides output gas which is completely clean and cool, with no particulate or objectionable chemical components. The inflator has no combustion-based gas generation and hence needs no filtration. As a result it is environmentally attractive, more easily recyclable, requires little or no pyrotechnic material and can be classified as a non-pyrotechnic device. To improve on earlier deficiencies of stored gas inflators, the variation of output with temperature is narrowed by means of internal mechanical components. For the first time, significant energy is added to the gas by convection rather than by chemical means. Pulse-shaping is also provided by mechanical means.

INTRODUCTION

All current inflators use some form of chemical combustion to inflate the airbag. Pyrotechnic airbag systems rely on sodium azide based gas generating material or newer alternative formulations both to generate the gas from solid and to make it hot. Hybrid inflators use the heat generated by a small quantity of exothermic material to enhance the energy content of a gas stored at high pressure. The only inflation technology which is essentially non-chemical is the traditional stored gas inflator. Although not currently in wide use within the automotive industry, it is the only inflator which produces completely clean gas.

In planning this work, we set out to explore broadly what new inflator technologies might be possible. We asked ourselves fundamental questions such as those shown in Table 1. We considered the history of stored gas and sought to re-examine the situation in light of any possible new technology. In comparing hybrids with stored gas, we noted that the essential function of the pyrotechnic is the addition of energy to gas. We then asked ourselves if there were any way of adding heat to gas other than chemical combustion. Availability of some other means of energy addition would open up a whole new category of inflator. Because of the inherent cleanliness of stored gas, we set out to see if it was possible to improve the deficiencies of the stored gas inflator while retaining its advantages.

Table 1.
Questions We Asked

- Why Did Stored Gas Become Unpopular? Are Those Reasons Still Valid?
- What Are the Essential Differences Between a Hybrid Inflator and a Stored Gas Inflator?
- Is There Any Other Way of Adding Heat to Gas?

Table 2.

Perceptions of Traditional Stored Gas Inflators	
Cleanliness of Gas	plus
Environmental, Non-pyrotechnic, Recyclability	plus
Temperature of Exiting Gas	plus
Simplicity	plus
Output Variation with Temperature	minus
Pulse-Shaping Ability	minus
Intelligent Adjustability	minus
Size, Weight	minus

In answer to the first question, Table 2 summarizes some perceptions of the traditional stored gas inflator. Probably the most significant positive quality is cleanliness of the gas which is produced. Airbag customers require that the gas not be irritating to the vehicle occupant, as expressed in specifications governing the maximum allowable quantities of both particulate entrained in the gas and chemical species in it. Pure stored gas inflation has essentially none of either of these. Also, manufacturing such an inflator involves little or no handling of raw materials which are toxic or pyrotechnic. Thus, the inflator is environmentally attractive ("green"). It may also be classified as a non-pyrotechnic device, which eases shipping and handling requirements. Similarly the benign nature of the component materials makes the inflator relatively easier to recycle. Another favorable feature is that the discharged gas is cool. Finally, stored gas inflators have the virtue of simplicity in terms of the processes that are involved in their functioning. In conventional inflators two physical processes which occupy much of the product development time are combustion and filtration. There are many geometric design variables of the filter itself and many chemical reaction and composition variables, all of which require development effort and careful control during manufacture. By contrast, in a stored gas inflator the performance is determined mainly by mechanical variables such as dimensions of orifices. It is easy to vary these and to predict the impact of such variation.

One drawback of traditional stored gas inflators is the fact that the output has a large variation with respect to the temperature of the inflator. This variation makes the airbag either too hard at hot initial temperatures or too soft or underinflated at cold temperatures. This issue has perhaps been viewed as the most serious and fundamental problem with stored gas inflators. Another issue is that (especially on the passenger side) it is advantageous to provide pulse-shaping. This takes the form of an S-shaped tank curve, which displays a gentle inflation rate at the very beginning, followed by a faster inflation rate. This is to reduce the possibility of the airbag injuring an out-of-position occupant. Future airbag systems will also require adjusting inflator performance according to variables such as crash speed and occupant weight and position. A traditional stored gas inflator does not provide these capabilities. Finally, size and weight have traditionally been a disadvantage of stored gas inflators. It will be seen in this paper that all of these disadvantages can be overcome or at least minimized. What is presented here is a description of a passenger side inflator, but the same principle can also be used for side impact applications.

ENERGY UTILIZATION IN INFLATORS

In comparing different kinds of inflators, it is useful to compare the sources and destinations of thermodynamic energy in the inflator. Fig. 1 shows approximate magnitudes of energy sources and losses typical of each of the three classes of inflator under extreme ambient temperature conditions. For this representation the parameters are chosen so that the total output under hot conditions for each type of inflator is the same. Energy losses are shown as extending below a datum line and are offset slightly for clarity.

For a conventional pyrotechnic inflator, information from Ref. 1 is summarized in the two left hand columns of Fig. 1, which explain performance at hot and cold. The source of all the energy is combustion, with significant loss of heat by convection to inflator components such as the filter. For simplicity the amount of energy released by combustion is shown as being essentially the same at each temperature. For a hybrid inflator, the pyrotechnic has three functions. It adds some energy to the stored gas, so that lower gas storage pressure is needed to carry the gas. Since the energy contribution of the pyrotechnic is somewhat temperature-independent, while the internal energy of the gas is significantly temperature-dependent, the pyrotechnic moderates the temperature dependence of the stored gas of the hybrid. Finally, the pyrotechnic is useful for opening the burst disc. In a typical hybrid inflator, a significant fraction of the output energy comes from the pyrotechnic material, and a significant fraction also comes from the internal energy of the stored gas. Since typical hybrid inflators do not require as much filtration, less heat is lost due to convection.

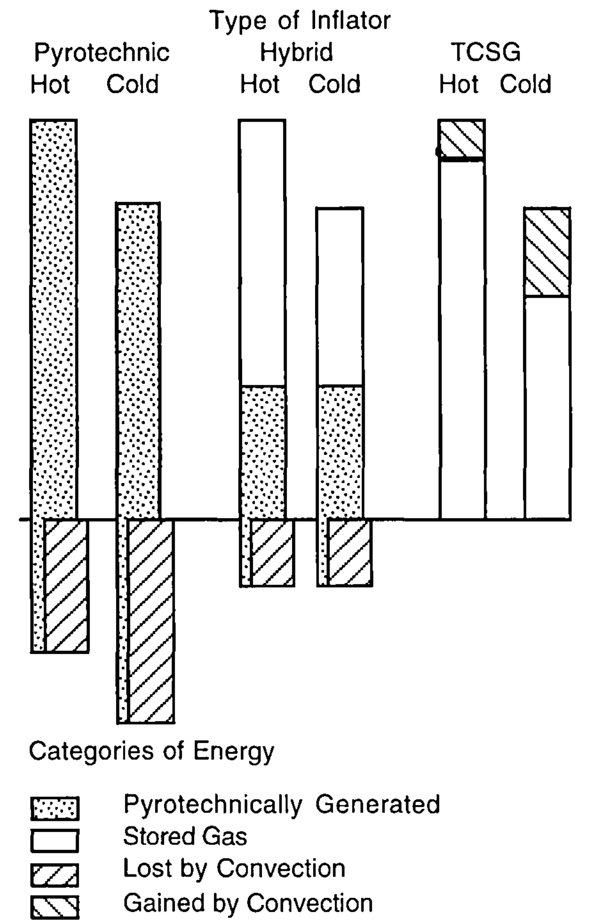

Figure 1. Energy utilization in various types of inflators, for extreme temperatures.

Given the convective loss of heat in both pyrotechnics and hybrids from the hot gas to the cold inflator components, it is logical to ask if it is possible to transfer heat convectively to exiting gas if a heat source exists at a higher temperature than the exiting gas. Such a situation does indeed occur when a stored gas is discharged, because of the cooling of decompression. Although this heat could simply be used to boost inflator output under all ambient temperatures, this would only make a slight improvement in inflator weight or output while not resolving the problem of the spread of energy output with temperature. This would not result in a large improvement in the commercial attractiveness of stored gas inflators. Instead, a more subtle and useful purpose for which to use this potential heat addition is to narrow the temperature dependence of the stored gas inflator. In order to accomplish this, heat must be added at cold initial conditions but not at hot initial conditions, thereby bringing the output at cold up closer to the hot output. Fig. 1 shows that a useful amount of energy is added at cold by convection, while a slight amount of energy is shown as being added (unavoidably) at hot initial conditions. This is the principle of temperature compensation used in the Temperature Compensated Stored Gas (TCSG) inflator, which is shown in more practical terms in Fig. 2. Stored gas inflators are the only ones in which convective heat transfer actually contributes positively to the inflator output. This deliberate convective addition of heat to the exiting gas is a completely new energy addition mechanism, which puts this inflator in a separate category of inflator technology. It is the only inflator with a non-chemical mechanism of energy addition. Significantly, this is addition of energy to the gas without adding any mass (particulate, combustion products, etc.). Thus, the inflator output is perfectly clean.

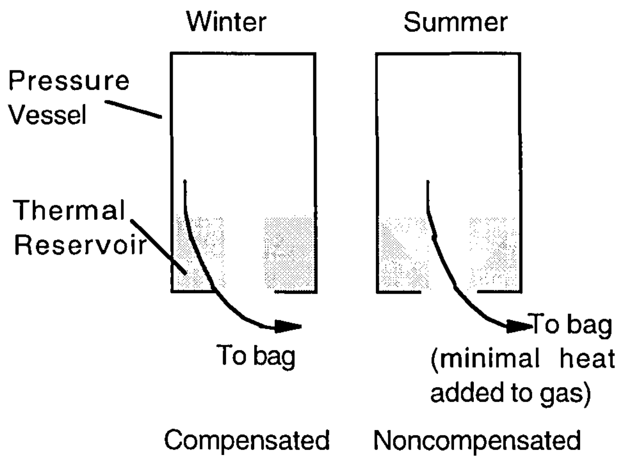

Figure. 2. Illustration of principle of temperature compensation.

THERMODYNAMICS

Thermodynamic issues must be understood to explain the performance of the TCSG inflator and to explain which gas is most appropriate for use in the TCSG inflator. These are explained in more detail in Ref. 2, but are given briefly here. Firstly, the pressure of stored gas inside an inflator varies as a function of the initial temperature of the inflator. The output energy of the inflator is usually correlated to the rise in pressure of a closed receiving tank when an inflator is discharged into it. According to the ideal gas law, both of these quantities should be essentially proportional to the initial absolute temperature of the stored gas. The usual temperature range for which driver or passenger side inflators are specified is -40 C to +85 C, which gives a ratio of absolute temperatures of 1.54.

When an inflator is discharged into an airbag, most of the output energy is used to fill or expand the bag from its folded position of essentially zero volume to its full volume. The remaining small quantity of the energy is used to raise the pressure of the bag above atmospheric after the bag has been filled (pressurization). Thus, an increase by a factor of 1.54 in the energy delivered to the airbag would result in an even larger increase in the peak bag pressure. The peak overpressure in the bag is an important parameter both because of strength limitations of the bag material and because the performance of the bag in absorbing the occupant's kinetic energy is influenced by the peak bag pressure. With such a large variation, an inflator would either fill the bag incompletely at cold initial conditions or dramatically overfill the bag at hot initial conditions.

The first law of thermodynamics (Ref. 3) explains the source of energy addition to the gas for purposes of temperature compensation. Decompression of a stored gas is associated with cooling, just as compression is associated with heating. The first-exiting gas is essentially at the initial temperature of the inflator. The later-exiting gas becomes colder and colder, and the last-exiting gas has a temperature in the range of about -230 C for typical initial temperatures. This extremely cold temperature suggests that heat can be put back into the gas. Furthermore, the heat does not have to come from an object which would ordinarily be thought of as hot, but rather could come from an object at the initial temperature of the stored gas. The thermal reservoir device merely has to be in thermal equilibrium with the gas at the initial temperature of the gas in the inflator before discharge, which provides a passive and hence reliable technique.

In order to further appreciate or evaluate the potential of the compensation technique, it is useful to calculate for a limiting situation how much heat could be put back into the gas by this method. This is done in Ref. 2, which shows that for ideal gas properties and the absence of heat transfer, the ratio of the mass*enthalpy delivered to the bag can be as high as gamma, the ratio of specific heats, assuming that the discharge at hot initial conditions is adiabatic and the discharge at cold initial conditions is isothermal. This indicates that by this simple and passive compensation means, not only would it be possible to bring the cold output up to equal the hot output, but it would actually be possible to make the inflator produce more output energy at a cold (-40 C) condition than at a hot (+85 C) condition. In practice, the extent of compensation achievable is not as large as this. This is because there is finite performance of the thermal reservoir and imperfect heat transfer at compensated conditions, unwanted heat transfer at noncompensated conditions, incomplete emptying of the gas from the pressure vessel, and similar effects. Nevertheless, it is possible to obtain enough compensation to significantly narrow the spread compared to what would occur without temperature compensation. This makes the narrowness of the output range of this inflator comparable to or narrower than the output range of other inflator technologies.

Typical gas storage pressures in the TCSG inflator are large enough that the ideal gas law does not adequately describe the performance. Real gas thermodynamic effects are important and imply that not all gases perform equally well. Nitrogen and argon have been used in most earlier stored gas and hybrid systems, but helium is a better candidate for this situation. At the typical storage conditions, the equations of state for these gases depart from the ideal gas law, giving a compressibility factor, Z, of about 1.2 for helium, 1.25 for nitrogen and 1.07 for argon (Ref. 4). This has an influence on the size of the pressure vessel.

The thermodynamic process of throttling occurs at the inflator discharge, where a gas is allowed to flow rapidly across a large pressure difference. This is considered to be an isenthalpic process. For an ideal gas, throttling produces no change in the temperature of the gas because enthalpy is a function only of temperature. For real gases there is a change of temperature. In the range of conditions encountered in stored gas inflators, argon and nitrogen cool upon throttling, whereas helium increases slightly in temperature upon throttling. This affects how much bag volume can be produced from a given volume of gas stored at a given pressure.

The specific heat ratio, gamma, also differs among the candidate gases, being 1.67 for the monatomic gases (argon and helium) and 1.4 for diatomic gases such as nitrogen. A larger value of gamma means that the cooling effect associated with decompression is somewhat larger for helium than for nitrogen. This provides the opportunity to transfer a larger amount of heat back into the gas for the purpose of providing compensation.

The speed of sound also differs for the various candidate gases, being largest for lightest gases. Helium near room temperature has 2.5 to 3 times the speed of sound of other candidate gases such as nitrogen and argon at the same temperature, and even when the nitrogen is hot as from azide inflators helium still has a larger speed of sound. This helps to provide rapid discharge of the gas given the fact that there may be practical limitations on the areas of exit orifices. Another advantage for helium is that it will not come close to its condensation point for the gas remaining inside the pressure vessel near the end of discharge. Other gases such as nitrogen or argon could reach their condensation points during the later part of the discharge transient especially under cold initial temperatures, and condensate is not useful for filling the airbag. The use of helium also saves a modest amount of inflator system weight, i.e., the weight of the stored gas itself.

OVERALL DESIGN

The major components of a TCSG inflator are the pressure vessel, the compensator mesh and thermal valving assembly, the pulse-shaping device, and the opening device. Fig. 3 shows the overall arrangement of these components, except that the pulse-shaping device is not shown and the burst disc opening device is only shown schematically.

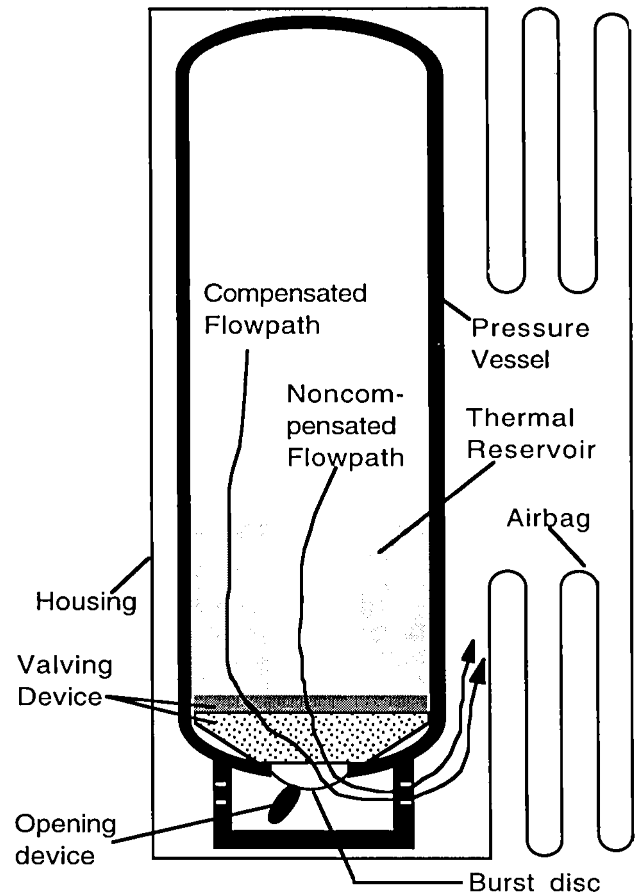

Figure 3. Overall arrangement of TCSG passenger side inflator showing compensated and noncompensated flowpaths. Pulse-shaping device is omitted for simplicity.

THERMAL COMPENSATOR

The thermal compensator consists of the thermal reservoir and the temperature-dependent valving device. Together these provide separate compensated and noncompensated exiting flowpaths. The distribution of flow between the flowpaths is adjusted as a function of the initial ambient temperature.

The compensator is placed inside the pressure vessel. Locating the mesh inside the pressure vessel ensures excellent initial equilibration of temperature between the stored gas and the mesh, because of the high density of the stored helium. The thermal reservoir also must supply sufficient heat to the gas exiting through the thermal reservoir. Thus, the thermal reservoir must have sufficient heat capacity, which is determined by its mass and material properties, and it must have sufficient surface area for heat transfer, which is determined by the geometry. A practical thermal reservoir can be manufactured from wire mesh. It is shown as being of an annular shape with the noncompensated flowpath occupying the empty central region.

In order to use the cooling of decompression to moderate the temperature dependence of the TCSG inflator, it is necessary that this augmentation of the temperature of the exiting gas should occur at cold initial conditions but not at hot initial conditions. This requires the use of two separate exiting flowpaths, one of which (the compensated flowpath) conducts the gas through the compensating thermal mesh and one of which (the noncompensated flowpath) conducts the gas so as to avoid thermal contact with the mesh. Two such parallel flowpaths are shown in Figs. 2 and 3, with each going from the gas reservoir to the airbag or receiving tank. At intermediate temperatures the flow should be partly compensated and partly noncompensated.

The distribution of flow between the two flowpaths is determined by a temperature-dependent valving device. The valving device should set its position as a function of initial temperature, and it should maintain the same position throughout the discharge transient. In the present device temperature-dependent flow areas are used in both flowpaths, such that one flowpath is opened as the other is closed. One possible mechanism for valving uses the phenomenon of thermal expansion, in that the motion that causes the valving is the lengthening or contraction of one solid part relative to another solid part. Other methods of adjusting the compensator valving as a function of temperature have also been demonstrated.

PULSE-SHAPING MECHANISM

Pulse-shaping is necessary to protect out-of-position occupants and to lessen the stresses on the housing, airbag and related structures. It is desirable to have a relatively gentle flowrate for the first 5 to 15 milliseconds of the discharge transient, followed by more rapid fill until the flowrate tapers off due to depletion. In this inflator, we accomplish pulse-shaping by mechanically varying an orifice area in a time-dependent manner. Appropriate pulse-shaping has been achieved by several mechanical designs.

OPENING MECHANISM

Inflators which incorporate stored gas usually use some form of rupture disc to keep the gas hermetically sealed during normal driving and yet release it quickly upon command. Rupture discs are simple in design and can be manufactured inexpensively when the quantities are large. They are characterized by a specified burst pressure for spontaneous rupture, which is somewhat temperature-dependent due to material properties. Rupture discs used in hybrid inflators either are mechanically punctured using pyrotechnic devices or are burst by overpressure. Older stored gas inflators relied on various pyrotechnic devices to initiate the opening.

One component which can be conveniently used in actuating a TCSG inflator is a piston actuator or gas motor. This contains an enclosed electrically initiated pyrotechnic, which when it burns creates a large internal pressure which propels a piston forward. The products of combustion are retained inside the device. Because of this retention and the typically small amount of pyrotechnic used, such a device may be classified for shipping and handling purposes as a non-pyrotechnic device. Opening devices have been demonstrated using the piston actuator which open a burst disc upon command over the entire temperature range with no release of pyrotechnic. With this design we have achieved the reality of zero particulate emissions.

PRESSURE VESSEL

One of the original problems with stored gas inflators was that they were undesirably large and heavy. To some extent, that is an inherent feature of these inflators, but the weight can be reduced by the use of modern high-strength materials for the pressure vessel. The material selection for our prototype stage is a hardenable steel. The physical size of the inflator can be reduced by storing the gas at a higher storage pressure. A typical storage pressure for the present design is 41 MPa (6000 psi) at room temperature.

In the present inflator a complication is introduced by the fact that heat treatment of the completed pressure vessel is not possible because of the temperature-sensitive internal components of the valving device, which occupy essentially the entire internal cross-section of the cylinder. A proprietary method has been developed to perform the major closure operation after the internals are installed. The final step is the fill and seal operation, which includes a very tiny resistance weld at the point of filling as is common in other inflator sealing operations. Pressure vessels of the present type are regulated in the United States by 49 CFR 178.65, referred to as DOT 39. These units comply and have been routinely shipped after filling.

Helium is perceived by many people as being difficult to store at high pressures over long periods of time. Helium leaks through capillary type cracks more rapidly than other gases. On the positive side, however, helium does not diffuse at all through solid metal, which is an advantage compared to the known behavior of hydrogen. Thus, the suitability of helium depends on quality of materials and welds. In industries such as aerospace, helium has been successfully stored in high pressure vessels for time periods of many years. This has been accomplished in quantities of hundreds of thousands with extremely high reliability. This same manufacturing technology will be used on the TCSG inflator. The basic requirements for successful containment of helium are simply hermetic design of the pressure boundary, high quality welding practice, and sensitive but readily achievable leak checking at the time of manufacture. With modern mass spectrometer leak detection equipment, it is possible to measure a leak rate which corresponds to leakage of several percent or so of the contents of a passenger side inflator over a period of 15 years. This would detect at the time of manufacture any inflators which would leak excessively over their design lifetime.

To further minimize concerns about helium, we have developed a low-cost non-intrusive pressure monitor which can be incorporated should the customer so desire. Finally, pressure relief (for example, for bonfire testing) is accomplished by the burst disc which also provides the inflator opening.

MODULE AND SYSTEM ISSUES AND EXPERIMENTAL RESULTS

Development of the TCSG inflator at Breed has included hundreds of deployment tests to date into a closed 100 liter receiving tank. This testing has brought the design to practical configurations of all of the components.

The prototype TCSG passenger side inflator is pictured in Fig. 4. Fig. 5 shows a tank curve produced by it at ambient temperature. At the beginning of the transient the tank curve exhibits pulse-shaping of a duration which is judged to be appropriate for typical vehicles, but which can be adjusted by dimensional variations. Fig. 6 shows a series of tank curves at cold, ambient and hot temperatures, taken with a slightly earlier prototype. In order to concentrate on variation with temperature, the pulse-shaping device was left out of the inflator for this series of shots. These tank curves show a spread between hot and cold output which is comparable to that of other present-day inflators. This spread is significantly reduced compared to what is observed in tests with stored gas inflators that do not have the compensating mesh and valving assembly inside them.

Development of the TCSG passenger side inflator has also included a significant quantity of pendulum and sled tests. Results to date indicate that satisfactory injury indices can be achieved, with the occupant-bag interaction being essentially the same as that from traditional inflators. Work is continuing to optimize occupant interaction and to improve the agreement between MADYMO predictions and the results from

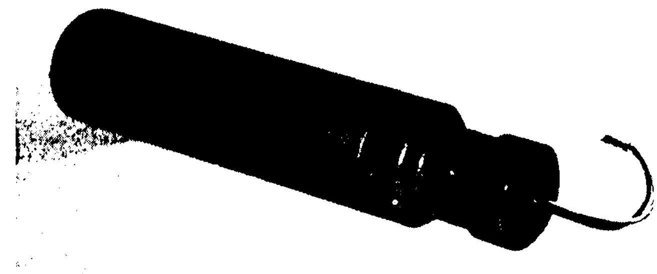

Figure 4. Photograph of prototype TCSG inflator.

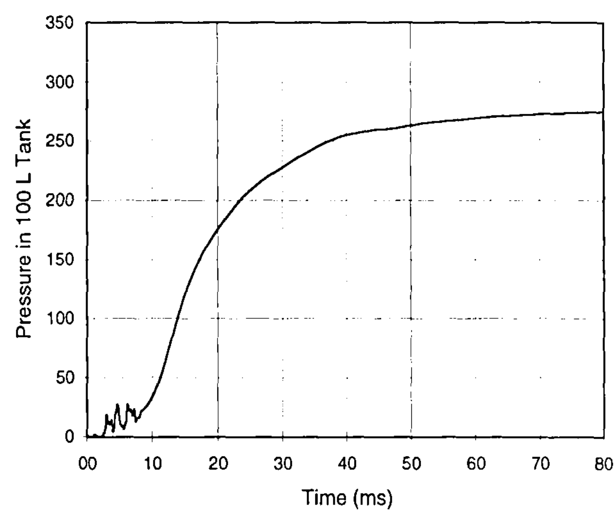

Figure 5. Typical discharge transient for prototype TCSG inflator discharged into a 100 liter receiving tank, at room temperature.

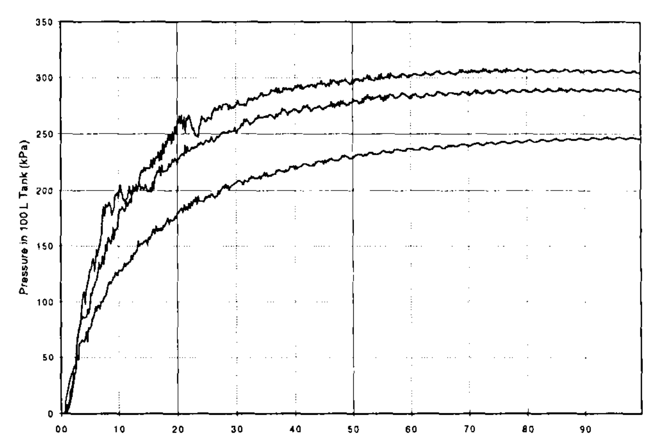

Figure 6. Transient pressures in 100 liter tank for hot (top curve), ambient (middle curve) and cold (lowest curve). Pulse-shaping omitted for simplicity.

dynamic testing. Among the most important issues to be determined or verified by dynamic testing are details such as bag vent area and the required inflator output or mass of stored gas.

There are significant systems issues which must be appreciated related to the behavior of the bag with helium. No other inflator currently on the market produces cool helium. Most inflators produce warm or hot nitrogen or argon. Thus, it should be emphasized that it is not appropriate to take a cold helium inflator and install it in a module which has been designed and optimized for another type of inflator. It is also not appropriate to require a cold helium inflator to meet exactly the same tank final pressure as other types of inflators. Because helium has such a low density the appropriate bag vent area is smaller than would be needed for the other gases. Venting through the porosity of an uncoated bag and the stitching becomes more significant than it would be for argon or nitrogen. There is evidence of a non-negligible amount of heat transfer between the bag and the gas. With a cold helium inflator this heat transfer adds heat to the gas, in marked contrast to all other inflators where the heat transfer removes heat from the gas in the bag.

A stored gas passenger side inflator discharges its gas at one end and the housing directs and redistributes the gas to enter the airbag. With other inflators, one perceived reason for the clearance between the inflator and the housing is for thermal isolation so that the housing does not become heated too hot by exiting gases and pose a danger of igniting nearby material. Because the TCSG inflator produces no heat and helium has a high speed of sound, is likely that the clearance can be smaller and hence the housing can be smaller for a given size inflator. It is possible to design a module with no housing around most of the inflator pressure vessel, thereby resulting in a module whose outside diameter is exactly the outside diameter of the inflator pressure vessel.

As an illustration of the cleanliness of this inflator, Fig. 7 shows a frame from a videotape of a folded bag deployment of a TCSG inflator. In this test the viewing angle of the camera happened to be such that at a certain point during the discharge the two vent holes on opposite sides of the bag aligned so there was a line of sight all the way through the bag to the checkerboard in back. (In this test the vent holes are larger than they typically would be.) No smoke can be seen either coming out of the vent holes or looking through the entire bag. Another readily perceivable advantage of this inflator is that there is no smell. Laboratory personnel usually walk up to the bag immediately after deployment and the absence of irritating odor is readily apparent. Also, the bag is cool to the touch, and whatever gas might still be venting from the bag is also cool. These things would similarly be noticeable to a vehicle occupant who finds himself in an accident situation, and they are very favorable for the TCSG inflator. To illustrate how benign helium is, it is can be pointed out that helium is a component of the breathing mixture used by underwater divers who dive to great depths.

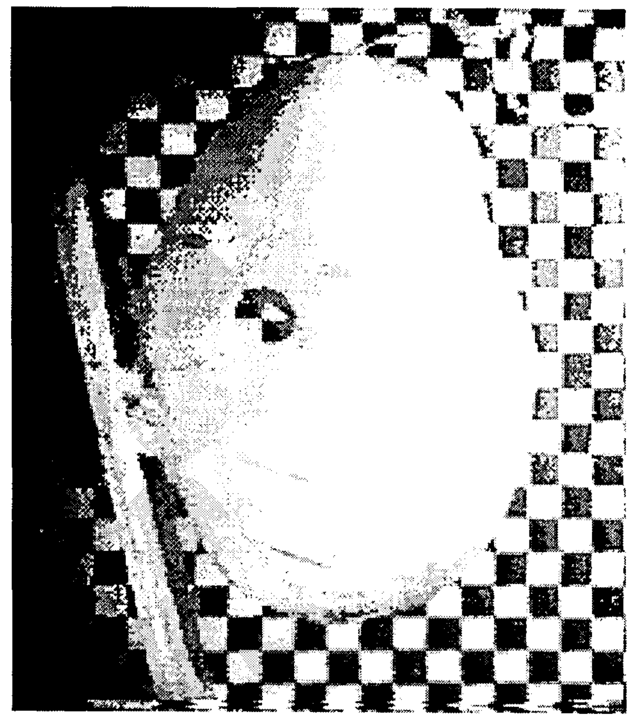

Fig. 7. Deployment of a TCSG inflator with line of sight through vent holes in bag, illustrating absence of smoke.

Some of the parameters of prototype TCSG passenger side inflators are summarized in Table 3. (Work is also being done on side impact inflators using similar principles.) These parameters are based on a 150 liter passenger side bag. The inflator outside diameter is 70 mm. The inflator length is shown here as a little bit under 300 mm. The weight is 1878 grams, with the weight of the rest of the module estimated at 1800 grams. This would give an overall module weight of about 3700 grams. While this does not make it the lightest inflator module, it is lighter than some of the products currently being used and we believe it is light enough to be competitive. Further reductions in size and weight have been proposed and are currently being evaluated.

Table 4 summarizes the improvements which have been made in each of the problem areas of stored gas inflators. The spread of output as a function of initial temperature has been narrowed to approximately equal the range of most other inflators. A pulse-shaping feature has been provided. An intelligent adjustment feature is being developed which would allow adjustment of the output to a very fine degree of adjustment. And finally, in regard to weight and size, we believe we have brought the size and weight down to the point where this inflator compares well with other inflator technologies, especially in view of its other advantages and benefits.

Table 3.

Principal Features of Passenger Side TCSG Inflator	
Bag Volume	150 L
Diameter of Inflator	70 mm
Length of Inflator	295 mm
Weight of Inflator	1878 g
Weight of Rest of Module	1800 g
Dimensions and Weights vary for other bag volumes	

Table 4.

Features of Improved Stored Gas	
Cleanliness of Gas	plus
Environmental, Non-Pyrotechnic, Recyclability	plus
Temperature of Exiting Gas	plus
Simplicity	plus
Output Variation with Temperature	plus
Pulse-Shaping Ability	plus
Intelligent Adjustability	plus
Size, Weight	OK

CONCLUSIONS

The Temperature Compensated Stored Gas inflator has the greatest possible appeal of any inflator for aspects that can be perceived by vehicle occupants after an accident. The problem of irritating gaseous components is not just fractionally improved but rather is completely solved, and at the same time the output gas from this inflator is cool rather than hot. From an environmental and recycling point of view, the components of this inflator are benign as well. This inflator achieves this by keeping the good features of traditional stored gas inflators, adding simple mechanical components to improve the performance in such areas as pulse shape and output variation with temperature, and by optimizing some thermodynamic considerations and the design and material selection for the pressure vessel.

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THE APPROVAL OF AIR-BAGS ETC.- THE NEED FOR A STANDARD

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Paper Number 96-S1-W-24

ABSTRACT

This paper discusses the need for an international standard which may be used by National Competant Authorities (NCAs) to approve or authorise the acquisition of air-bags and seat- belt tensioners.

INTRODUCTION

Recent discussions in USA between the National Highway Traffic Safety Administration (NHTSA) and others (eg motor and insurance industry) on ways of reducing unwanted side-effects of air-bags has focussed on four main points:-
(a) changes in crash test requirements
(b) enforcement of seat-belt use regulation
(c) education of the public and
(d) the need for "smart" restraint systems.

All of these topics will be featured at 15th ESV but this paper will only consider one aspect of (d) ie it discusses the problems experienced by companies resulting from delays in obtaining approvals or authorisations for the components of smart systems and suggests that such delays could be avoided if national authorities based their approval or authorisation procedures on an international standard.

Approval or Authorisation

The requirement for approval or authorisation arises from the fact that the gas generators or inflators contain components which are considered as explosives by many national authorities and are thus subject to national regulations which consider both safety and security.

The process of approval or authorisation differs from nation to nation but in most cases the authority imposes a charge on the applicant and invariably causes a delay to the applicant.

Thus, the process of approval often creates barriers to the importation of foreign products.

Approval or authorisation should not be confused with the "classification" of the product as packaged for transport or with the whole vehicle "type approval" of the vehicle fitted with the devices.

(Details of "classification", including packaging and testing to the "UNITED NATIONS RECOMMENDATIONS ON THE TRANSPORT OF DANGEROUS GOODS, EDITION 9", may be obtained from the report, "REGULATIONS ON AIRBAGS AND SEAT-BELT TENSIONERS", by this author. Please contact ACE Consultancy on FAX +44 1606 861440)

The process of approval or authorisation is now further complicated by the introduction of "hybrid" devices in which the gas generator or inflator may contain both explosives and compressed gas and the approval of the device may be the responsibility of more than one authority (eg in Germany, BAM approves the explosive, TuV approves the compressed gas cylinder and gas.) In general, the NCA in the country of manufacture bases the approval on the results of tests and/or detailed inspection under national or domestic regulations. eg in USA, approval is based on Section 49 of the Code of Federal Regulations (CFR-49), whereas in countries such as Belgium, France, Germany, Italy, Spain, Switzerland and UK, approval or authorisation is under domestic explosives regulations, many of which were promulgated long before air-bags were even considered (eg UK 1875 Explosives Act).

Approval or Authorisation of Other Explosive Devices

International standards have already been developed for some explosive devices (eg CIP and SAAMI for sporting ammunition and SOLAS for marine pyrotechnics) and this has resulted in minimum delays and costs in obtaining approvals for importation of these items.

Since the publication of British Standard 7114 (on fireworks) in 1988, the procedures for importing fireworks into UK have been simplified and are now far less stringent than those for importing car air-bags or seat-belt tensioners!

EEC Directives.

Unfortunately, there is as yet no specific EEC Directive on air-bag inflators or seat-belt tensioners. Instead, each EU nation interprets non-specific Directives and applies them to their national legislation.

Because air-bag inflators may be considered as ammunition in one country (eg UK) and pyrotechnic articles in another (eg Germany), the devices may be not be subject to the same EEC Directives in each EU country (Subsidiarity?).
Also, because some EEC Directives are based on UN Recommendations on the Transport of Dangerous Goods, their applicability to articles outside their packaging is questionable.

ISO and GRSP

The standards being developed by WG8 of ISO TC22 and GRSP of UN-ECE WP29 may eventually be used to specify the quality, performance or efficacy of air-bags in vehicles but may not necessarily be appropriate for use by NCAs to approve or authorise air-bags for acquisition prior to fitting into the vehicles.

Proposed Standard.

It is proposed that a simple standard, based essentially on tests currently carried out by BAM (Germany) and US-DOT, (see figure 1) should be developed by representatives of "industry" and submitted to representatives of the national testing "authorities".

"Industry" could be represented by an organisation such as the Automotive Occupant Restraints Council (AORC) (formerly the American Seat-Belt Council), since the majority of air-bag and seat-belt manufacturers are members of AORC, and the Explosives, Propellant and Pyrotechnic (EPP) Group of the OECD-IGUS (Organisation for Economic Cooperation and Development - International Group on Unstable Substances) could represent the national testing authorities and/or NCAs.

The next meeting of EPP of OECD-IGUS is scheduled for Orlando, USA, from September 23rd to September 26th or 27th, '96, and representatives from the following countries will be invited to attend:-

BELGIUM, CANADA, CHINA, DENMARK, GERMANY, FINLAND, FRANCE, IRELAND, ITALY, JAPAN, NETHERLANDS, NORWAY, POLAND, PORTUGAL, SPAIN, SWEDEN, SWITZERLAND, TAIWAN, UNITED KINGDOM and UNITED STATES OF AMERICA,

The same standard could be used by DGIII of EEC as the basis of an EEC Directive on vehicle air-bags and seat-belt tensioners.

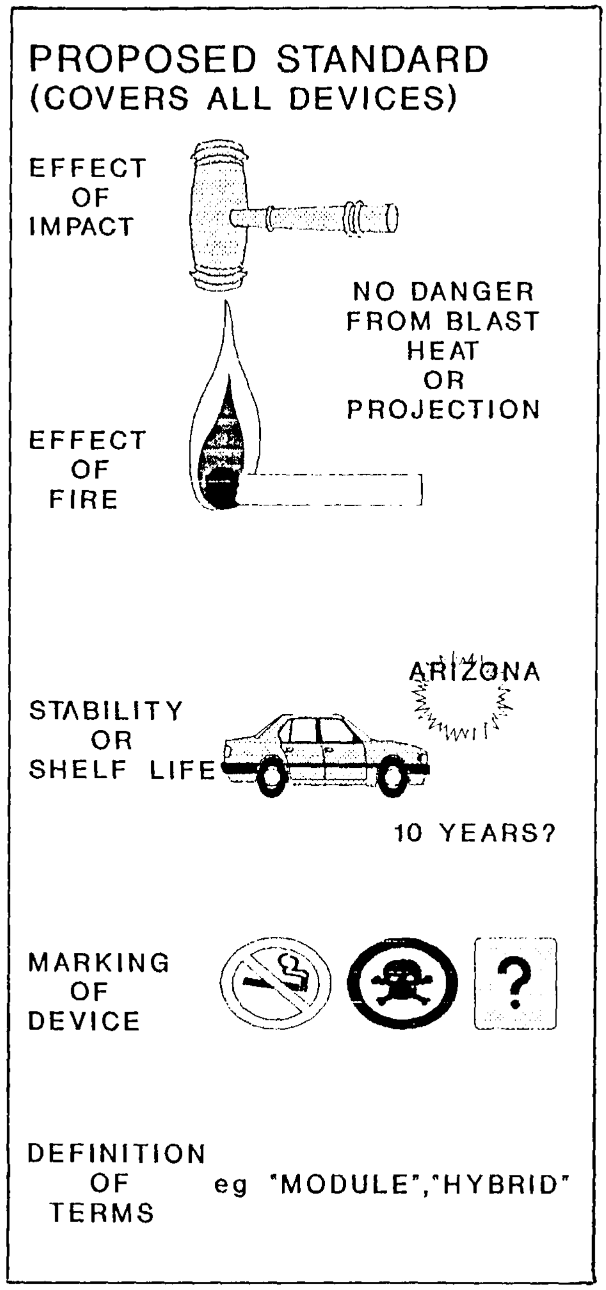

Figure 1. Proposed standard.

THE DEVELOPMENT OF RESULT PRESENTATION IN AUSTRALIA'S NEW CAR ASSESSMENT PROGRAM

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Paper Number 96-S1-W-27

ABSTRACT

This paper examines market research findings about consumer knowledge of vehicle safety. Consumer demand for simplified information is evaluated in the context of output generated from Australian NCAP full frontal and offset frontal barrier crash tests.

The development of Australia's NCAP result presentation format is described. Australian NCAP originally presented results as discrete dummy response measurements from each crash test. Following consumer demand to simplify presentation, NCAP management applied injury risk functions to the results producing a single index rating. This 100 point scale represents the risk of life threatening injury for a weighted combination of full frontal and driver's side offset crashes.

Correlation of the single index rating with actual crash outcomes is examined.

INTRODUCTION

In 1992 Australian motoring clubs and state road authorities initiated a New Car Assessment Program. Results of the first series of full frontal crash tests of nine medium and large cars were released in April 1993. Results of the second series of full frontal and offset frontal tests on 12 small cars were released in April 1994. Subsequently, regular releases of results from passenger cars, four wheel drives, commercial utilities and passenger vans were conducted, NCAP, (1994 - 95).

From November 1994, results were released in a simplified single index rating format which combined dummy response values from full frontal and offset frontal tests. To date, results from 48 vehicle models have been released in this manner. The simplified mode of presentation was developed to improve consumer comprehension of the results and thus enhance the aims of the program.

AUSTRALIAN NCAP TESTING

The Australian NCAP involves both full frontal and offset frontal barrier testing of various passenger vehicle models. Full frontal testing is carried out at 56 km/h rather than the 48 km/h nominally specified in Australian motor vehicle safety regulations. Initially, offset testing was undertaken at 60 km/h; however was increased during 1995 to 64 km/h . This was done for consistency with the US Insurance Institute for Highway Safety's (IIHS) offset testing program and an expected British offset testing program.

The full frontal tests impact specimen vehicles into a rigid barrier, while the offset tests are conducted with a 40 percent offset into an aluminium honeycomb barrier. The offset test is based on the protocol devised by the European Experimental Vehicle Committee (EEVC) as detailed by Lowne, (1994).

Hybrid 3 instrumented crash test dummies are placed in the front outboard seating positions of test vehicles. Transducer readings are recorded during impact, and from these the potential for injury to head, chest and legs of the driver and front passenger are calculated. These measurements provide the basis for comparison of the frontal crash protection afforded by the various vehicle models tested.

Measurements recorded are :

Head Injury Criterion (HIC)
Chest compression (mm)
Resultant chest acceleration (G)
Femur load (kN)
Lower leg bending moment and compression forces (expressed as an index)

Griffiths et al, (1994) outlined the history leading to the inception of Australia's New Car Assessment Program and details of its implementation. Bailey, (1994) details the derivation of the lower leg injury index.

OBJECTIVES OF NCAP

The objectives of the Australian NCAP are to:

- provide car buyers with specific information about the relative occupant protection characteristics of the most common models on the Australian market;

Table 1.
Summary of NCAP Results - April 1993

Test Results

Make/Model	Head Injury Criteria (HIC)	Chest Deformation (mm)	Femur Loads (kN) Left	Femur Loads (kN) Right
MEDIUM SIZE CARS				
Mazda 626				
Driver	1160	49	2.63	2.57
Passenger	930	34	3.48	3.11
Nissan Pintara				
Driver	1750	44	1.29	2.42
Passenger	890	40	0.78	0.74
Toyota Camry				
Driver	1090	41	1.59	3.93
Passenger	1240	39	1.57	2.52

- encourage local manufacturers and importers to provide higher occupant protection levels in their models;

- show that motoring associations and state road authorities are actively promoting safer vehicles and pushing developments in this field;

- establish an independent test facility and knowledge base for occupant protection in Australia; and

- obtain knowledge and understanding of the latest advances in occupant protection and test techniques through active participation in this field.

The essential objective of the program is to lift occupant protection levels in Australian vehicles through market pressure from consumers, which can only be enhanced through improved consumer understanding of the program's output. This is the rationale for continued review of the presentation format of NCAP results.

RESULT PRESENTATION HISTORY

The first issue of NCAP results in April 1993 treated only the results of a full frontal series of tests. Results were expressed in terms of eight pieces of information for each model as shown in Table 1. This information was supplemented by human figure graphics for each vehicle model. These figures depict the dummy responses in each body region, with risk levels in the human figure represented by colour coded ellipsoids showing either low, medium or high risk levels. An illustration of the human figure graphic is shown in Figure 1.

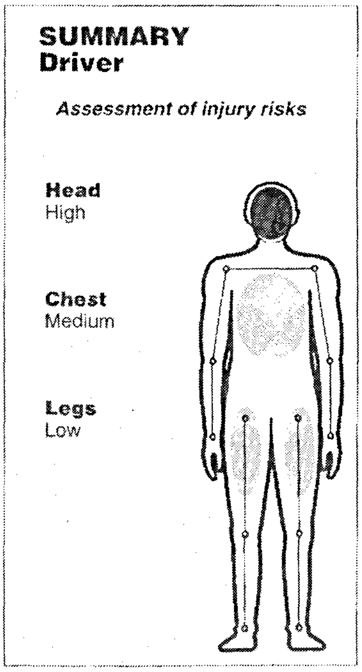

Figure 1. NCAP Human Graphic

The inclusion of a series of offset tests effectively doubled the quantity of information expressing the performance of each model, thereby making comprehension by average motorists even more difficult. Although colour coded to distinguish "high", "medium" and "low" injury risk levels, the summary table of results in NCAP's small car report, released in April 1994, presented a formidable block of information.

Table 2.
Summary of NCAP Results - April 1994

	Make/Model Kerb weight (kg)			Full Frontal Test HIC*	Chest**	Femur (kN) Left	Femur (kN) Right	Offset Frontal Test HIC*	Chest**	Femur (kN) Left	Femur (kN) Right
Small Cars	Ford Laser		D	1900	42	8.57	2.80	3230	39	7.0	11.2
	(1993)	1020	P	1790	•	2.70	3.87	1300	43	#	#
	Mitsubishi Lancer		D	1280	51	3.86	0.98	1180	33	12.9	4.0
	(1993)	1055	P	1240	36	0.86	1.16	790	36	#	#
	Mazda 121		D	1530	34	4.73	4.37	1570	23	7.0	7.4
	(1993)	825	P	1070	34	0.99	2.10	1280	43	#	#
	Toyota Corolla		D	1500	54	9.41	1.93	1020	48	6.2	4.4
	(1993)	1110	P	1220	36	3.16	1.05	390	27	#	#
	Nissan Pulsar		D	1460	52	4.78	2.30	2160	42	9.8	18.3
	(1993)	1095	P	1530	43	2.02	2.49	810	37	#	#
	Hyundai Excel		D	1320	36	0.54	3.60	1200	34	4.9	0.6
	(1993)	947	P	860	37	2.04	1.30	710	36	#	#
	Holden Barina		D	1000	54	3.88	1.62	1210	44	8.3	6.8
	(1993)	795	P	1150	38	1.43	2.32	800	29	#	#
	Honda Civic		D	1460	52	3.0	3.2	620	30	1.0	1.3
	GL (1993)	1013	P	1100	41	1.65	1.54	470	40	#	#
	Honda Civic		D	1160	41	3.2	3.8	850	43	5.2	3.5
	VEi with airbag (1993)	1043	P	1170	40	1.81	2.01	920	38	#	#
	Subaru Impreza		D	1110	47	0.82	3.73	860	43	2.2	1.9
	(1993)	1045	P	890	41	0.87	2.55	360	35	#	#
	Daihatsu Charade		D	1000	50	2.15	2.67	1410	35	7.0	9.8
	(1993)	845	P	1260	41	1.33	3.03	1330	35	#	#

* Head Injury Criterion ** Chest Deformation (mm) • No Data # Passenger femur loads not recorded in offset tests.
•• The VR Commodore Acclaim has been tested and added to the above list as it is the first locally manufactured vehicle to be offered on the Australian market fitted with an air bag as standard.

The results needed careful interpretation to distinguish vehicles, so it was not surprising that feedback was received indicating that the results were potentially confusing. An example of the summary format is shown in Table 2.

No formal study of Australian consumer understanding was conducted. However, it was not expected to differ significantly from experience with US NCAP, which until late 1993 was published in a similar format.

As detailed in NCAP Report to the Congress, (1993), the US National Highway Traffic Safety Administration (NHTSA) funded a focus group study to evaluate consumer needs and to identify better ways to deliver NCAP information to the community. The NHTSA study found that to effectively reach consumers, NCAP should:

"present information on crash tests in a form that is non-technical and as short and simple as possible."

Specifically, NHTSA reported that consumers could understand information expressed simply in terms of "risk of injury", however were confused by technical data sheets and dummy output parameters that needed some level of interpretation.

US NCAP SIMPLIFIED PRESENTATION

In 1993, NHTSA began to publish NCAP results expressed as star ratings. An example of the US NCAP presentation format is shown in Table 3. Five stars indicate the highest crash protection for vehicles within the same weight class - 10 percent or less chance of life-threatening head and chest injuries; four stars around 15 percent, three stars represent about 20 percent chance of serious injury, which is approximately equivalent to a HIC of 1000 and chest deceleration of 60G, two stars around 40 percent, and one star represents the highest possibility of life-threatening injury of more than 45 percent.

Table 3.
US NCAP Presentation Format

Vehicle	Protection	Weight	Driver/D	Passenger/P
Mini (1500-1999 pounds)				
Geo Metro 2-dr hatch	Belts	1610	***	****
Light (2000-2499 pounds)				
Honda Civic Coupe 2-dr	Belts/D&P air bags	2498	***	****
Hyundai Excel 4-dr	Belts	2278	****	*****
Hyundai Excel 2-dr hatch	Belts	2200	****	*****
Hyundai Scoupe 2-dr	Belts	2201	****	****
Mazda Protege 4-dr	Belts	2417	***	****
Nissan Sentra 4-dr	Belts	2420	****	****
Nissan Sentra 4-dr	Belts/D air bag	2427	****	****
Saturn SL2 4-dr	Belts/D air bag	2481	****	***
Toyota Tercel 4-dr	Belts/D air bag	2130	****	*****
Compact (2500-2999 pounds)				
Chevrolet Cavalier 4-dr	Belts	2540	****	*****
Ford Tempo 4-dr	Belts/D air bag	2674	****	****
Honda Prelude 2-dr	Belts/D air bag	2818	****	*****
Mitsubishi Eclipse 2-dr hb	Belts	2594	****	****
Mitsubishi Galant 4-dr	Belts/D&P air bags	2832	no data	****
Subaru Legacy 4-dr	Belts/D air bag	2791	****	****

The star values are calculated using injury risk functions which relate measured dummy parameters to expected levels of life-threatening injury risk to the head and chest. Viano and Arepally, (1990) presented injury risk functions for assessing safety performance of vehicles in crash tests. These functions are based on Logist probability analysis of biomechanical data.

The main problem with the NHTSA star rating system was considered to be its arbitrary choice of star cut-off values and the resulting discrete bands of safety performance. Vehicles with scores at either end of a performance band would be viewed as having similar levels of safety. Additionally, any re-definition of cut-off values would lead to problems in future comparisons with previously compiled results.

After decisions were taken by Australian NCAP regarding its presentation format, US NCAP Report to the Congress, (1995) detailed some consumer concerns about the NHTSA star rating system after a period of its application. These concerns related to the results now being over-simplified, not giving sufficient distinction between groups of either poor performing or good performing vehicles. Additionally, there was still considered to be a demand for some numerical data. NHTSA reported future action would be taken to provide numerical data and review the use of the "star" symbol.

AUSTRALIAN NCAP SINGLE INDEX RATING

It was considered that on-going success of Australian NCAP relied on improving consumer comprehension. This could only be achieved by simplifying the format in which NCAP presented test results. By adopting and extending US NCAP developments in presentation, improved levels of consumer comprehension were expected.

For Australian NCAP, a continuous scale representing risk of injury was seen to be a more effective means of expressing final results and would be more likely to be consistent over time. The following narrative details the derivation and application of the NCAP single index rating.

As used the NHTSA star rating system, the Viano and Arepally risk functions are applied.

For head injury, the equation:

$$P_{head} = [\ 1+ \exp\ (5.02-0.00351 \times HIC)]^{-1}$$

relates the probability of an AIS 4 or greater injury to HIC

For chest injury, the equation:

$$P_{chest} = [\ 1+ \exp\ (5.55-0.0693 \times \text{Chest G})]^{-1}$$

relates the probability of an AIS 4 or greater injury to chest Gs

The Abbreviated Injury Scale (AIS), documented by AAAM, (1990) is a "threat-to-life" scale. Injuries are coded on a six point scale as follows:

1. Minor
2. Moderate
3. Severe (not life threatening)
4. Serious (life threatening)
5. Critical (survival uncertain)
6. Maximum (potentially non-survivable)

An individual who suffers multiple injuries has a higher risk of permanent disability or death. By applying the law of additive probability for non-mutually exclusive events from Mendenhall et al, (1986), individual probabilities of injury for head and chest for each driver and passenger dummy may be combined into a single probability value.

This combined probability of an AIS 4 or greater injury is calculated as:

Combined Probability, P_{comb} =

$$P_{head} + P_{chest} - P_{head} \times P_{chest}$$

Where US NCAP combined head and chest dummy measurements of full frontal tests only, Australian NCAP needed to combine head and chest measurements for full frontal and offset frontal tests for driver and front passenger in each vehicle model. This was achieved by weighting the combined probabilities of injury for full frontal and right offset crashes by the ratio for which these crash types occur on the roads resulting in either injury or death to drivers or front seat passengers.

A data sample was used from the Monash University Accident Research Centre crashed vehicle file of 300 cases where details of vehicle damage profile and occupant outcomes were known. It was assumed that the EEVC offset testing configuration is a representative simulation of typical real world offset frontal crashes and that the full frontal barrier test was a reasonable simulation of real world crashes where the whole vehicle front engaged either an oncoming vehicle or fixed object. Fildes et al, (1991) details the methodology behind collection of the real world case studies.

Combined probabilities for each full frontal and offset test case were established as:

P_{FF+O}(driver) =

$$0.59 \times P_{comb}(\text{full}) + 0.41 \times P_{comb}\ (\text{offset})$$

P_{FF+O}(f pass) =

$$0.71 \times P_{comb}(\text{full}) + 0.29 \times P_{comb}\ (\text{offset})$$

The final results for the two tests representing the safety performance of each model vehicle were plotted on a 100 point scale. The graphic treatment included a colour sweep from green to red, designed to reinforce the risk concept where a lower score is safer than a higher score. The colour coded human figure graphic used in the first releases of results was retained to depict injury risks in specific body regions.

NCAP brochures were published with summary charts for groups of comparable vehicles and more detailed results for each tested vehicle on its own summary page. Figure 2 illustrates an injury risk summary chart where results are shown ranked by driver scores. Figure 3 illustrates a vehicle summary page. The brochures using the single index rating format received favourable feedback from consumers about their "user-friendliness".

CORRELATION WITH REAL WORLD CRASH DATA

Newstead et al, (1995) examined the correlation of results from Australia's New Car Assessment Program with driver outcomes from real road crash data. The sources of crash and injury data were Australian police accident reports and insurance data.

Initial findings of this study suggested a number of relationships. Firstly, there was weak, but statistically significant correlation between NCAP measures and the real outcomes of all crash types. Secondly, the results of NCAP testing were highly associated with two car head on crashes which involve impacts similar to those simulated in NCAP testing. Correlations were stronger for offset tests than for full frontal test outcomes.

At this stage of the study, real world data has been matched to five vehicles with both full frontal and offset frontal NCAP test results. Very good correlation was reported between NCAP single index ratings and corresponding real world two car head-on crashworthiness ratings for this sample of vehicles. Correlation at the one percent level of significance was reported.

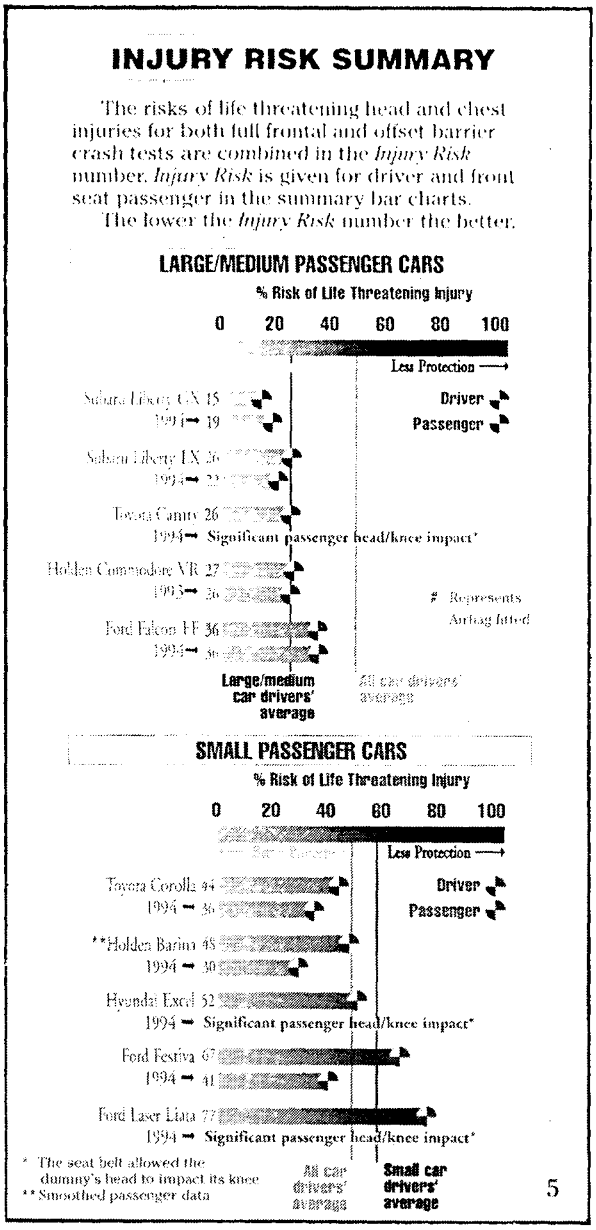

Figure 2. Injury Risk Summary Chart

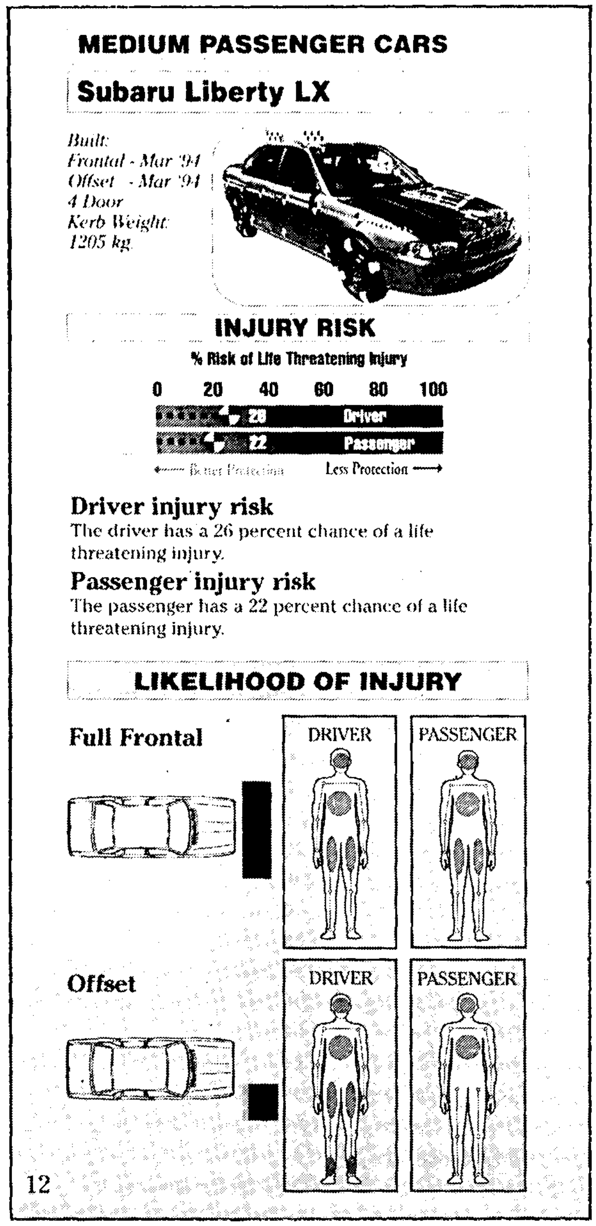

Figure 3. Vehicle Summary Page

FUTURE PRESENTATION

To date, NCAP has emphasised results measured by crash test dummies. In some offset tests it was noted that severe cabin intrusion occurred while measured dummy responses were quite moderate. It is considered that even with current Hybrid 3 dummy technology, an insufficient number of transducers exist to adequately simulate effects on humans in some situations. Forces could be imposed in locations away from transducers.

Acceleration based injury criteria have limitations. HIC while still being useful is a one dimensional measure unable to account for many injury producing effects such as rotational accelerations which occur in reality. Further, many human injury mechanisms are themselves viscous in nature so it is not surprising that anomalous observations sometimes occur in vehicle crash testing. Development of more biofidelic human surrogates is occurring, however it appears that the Hybrid 3 at this stage is the only feasible crash dummy for frontal testing by NCAP. Clearly, dummy response measurements need to be supplemented by other observations.

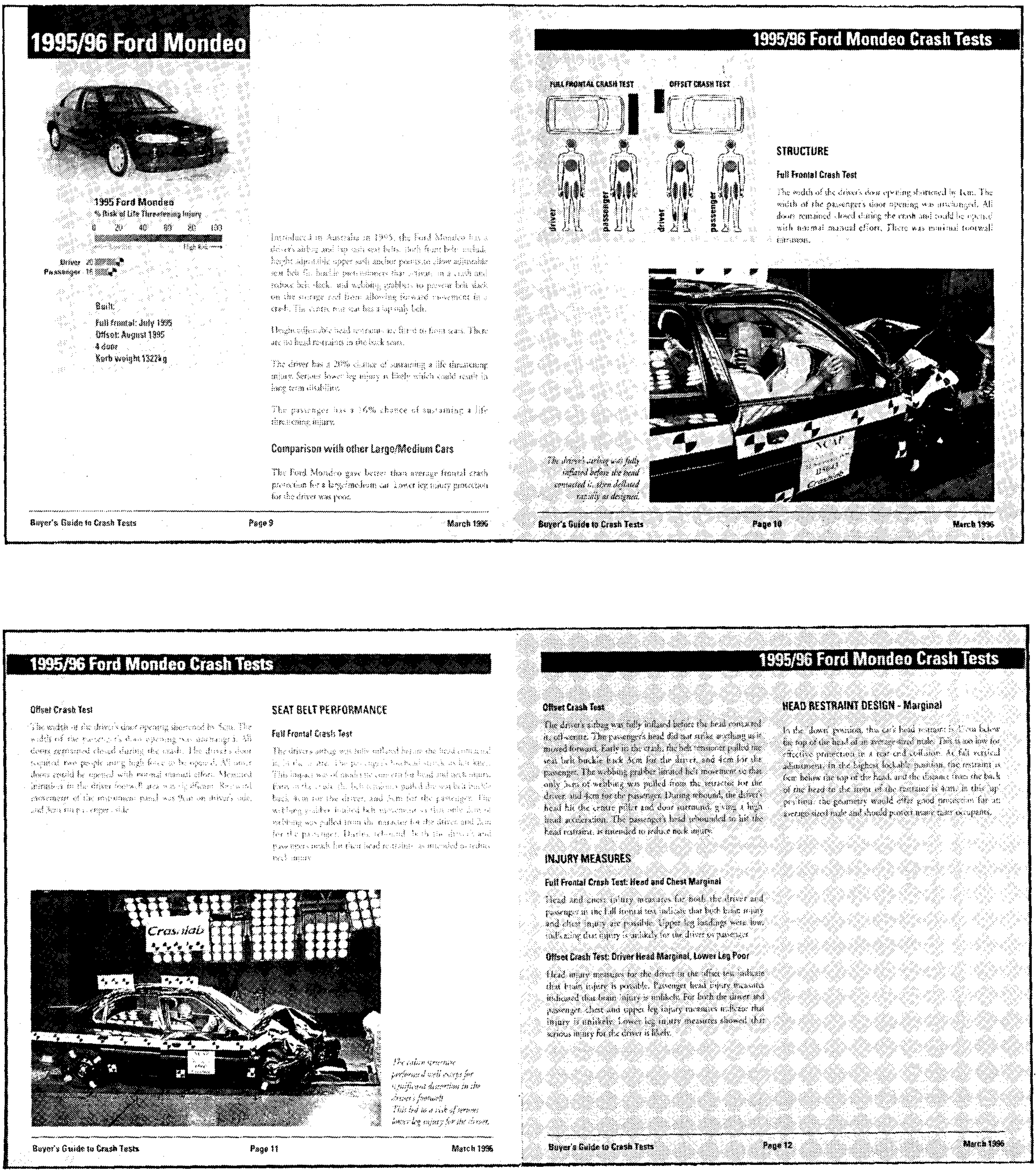

Figure 4. NCAP Presentation Format - March 1996

Accordingly, Australian NCAP has been reviewing its mode of presentation and started to extend its reporting beyond dummy derived injury measures. Latest reporting now includes more broadly based observations including structural performance, restraint fit and performance and headrest design. Similar reporting has to date been conducted by the IIHS in its offset testing program and detailed in its 1995 crashworthiness report. An example of the latest Australian NCAP reporting format is provided in Figure 4.

Future new types of crash testing such as side impact will bring new challenges to the effective presentation of information for the consumer. Australian NCAP will continue to investigate the most effective means of presenting vehicle crash test performance information to consumers.

ACKNOWLEDGMENTS

The author would like to thank colleagues at the following organisations who participated in the development and support of Australia's New Car Assessment Program and their support of this paper:

Automobile Association of Australia
Royal Automobile Club of Victoria
VICROADS
National Roads and Motorists Association
Roads and Traffic Authority New South Wales
Department of Transport Queensland
Royal Automobile Club of Queensland
Royal Automobile Association of South Australia
Department of Transport South Australia
Royal Automobile Club of Tasmania
Royal Automobile Club of Western Australia

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Technical Session 2

Performance Assessment of ITS Collision Avoidance Systems
Chairperson: Joseph Kanianthra, United States

HUMAN DETERMINANTS OF ACTIVE SAFETY: RESULTS OF INTERDISCIPLINARY DRIVER BEHAVIOUR EXPERIMENTS

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Paper Number 96-S2-O-01

ABSTRACT

In order to obtain and validate design criteria to improve active safety in road traffic, driver behaviour was analysed in an interdisciplinary approach. Effects of stress (e.g., route properties) and individual human characteristics (age, sex, personality, driving experience) on driver performance (especially the use of friction potential) and driver strain (physiological and psychological measures) were analysed.

This paper focuses on interindividual variations of performance and strain in field experiments. They were interpreted with individual characteristics: subjects with more driving experience and higher age show lower strain. Female subjects drive with higher safety margins and less strain. Conspicuousness in certain personality measures is related to conspicuous driver performance measures. Strain generally increases with performance, the slope value is determined by the task difficulty.

INTRODUCTION

In Germany, individual traffic with 37.6 million private vehicles on 636.282 road km adds up to 724 billion person kilometres and thus contributes to 82 percent of the overall German transportation of persons (Presse- und Informationsamt der Bundesregierung, 1993). According to Ellinghaus (1982), the quality of road traffic, which can be seen as a system consisting of the subsystems man, vehicle, road and society, should be evaluated not only by social, economic and ecological criteria, but also by criteria of safety (for which -in contrast e.g. to ecology- there are no quantitative targets set by politics; Schneider, 1993).

While fatality rates have decreased over the last 20 years in Germany, the number of persons killed in German road traffic in relation to quantitative output measures of transportation is still higher than in other countries (Germany: 19.3 persons per billion vehicle kilometres, USA: 10.9, UK: 10.4, Netherlands: 12.6; BASt, 1994). More than 10.000 fatalities and almost 400.000 accidents with injuries per year are a strong motivation for increased efforts in the field of road safety.

In the past decades substantial improvements in road safety have been attributed mainly to secondary safety measures, i.e. measures to reduce or prevent injuries, once a collision is inevitable (passive safety). Brown (1991) gives three reasons for this fact:

1) accident reporting systems provide more exploitable data on injury causation than on accident causation;

2) secondary safety measures involving modifications of vehicles and/or road environment "achieve their beneficial effects without requiring attitudinal and behavioural changes on the part of the road user";

3) the political decisions necessary for the introduction generally do not constrain personal freedom and are thus easier to take than primary safety measures which aim at the prevention of accidents (active safety) and require behavioural changes.

Because there is only little reliable information on the relationships between road user behaviour and road accidents, it is hard to identify appropriate (primary) safety measures to prevent such forms of behaviour that contribute to accidents.

The driver and active safety

In work systems in which the human operator contributes to the system's performance, human behaviour basically can affect man-system mismatches in two ways (Rasmussen, 1987):

- Human actions bring the system outside its boundary,
- Changes in system behaviour bring the human operator outside his boundaries which are determined by his individual capabilities.

Besides rather technical measures to make the system more error tolerant, countermeasures directly aiming at the behaviour of the human operator must be considered. Different strategies have to be applied: in the first case the person must be kept from doing wrong things (leading the system out of its boundaries) while in the second case the operator must be supported to do the right thing (maintain a match by reacting correspondingly in a situation that is critical due to e.g. environmental disturbances).

Brown (1991) discusses the effectiveness of the so-called 'Triple E' (engineering of vehicles and roads, education and training of road users and enforcement of traffic law) and comes to a clear priorisation: engineering measures to improve both hazard assessment and the "indication of driver's performance limits" can alter behaviour directly by changing task demands and are thus considered to have a greater and more sustained effect on safety than the other indirect measures. Yet it must be stated that while engineering measures can provide an improved *opportunity* to be safe they do not necessarily increase the *desire* to be safe (Wilde, 1992). As motivation plays an important role in active safety, the design of engineering measures must take behavioural adaptation into consideration, which might offset the potential safety benefits ("risk homeostasis", see e.g. Wilde 1982; Wilde & Kunkel, 1984; encouraging safe behaviour: Burkardt, 1992).

Vehicle control can be modelled in a hierarchical multilevel structure (Johannsen, 1976; Donges, 1992) with three control loops: navigation or planning (transferring goals into plans, task duration: minutes to hours), guidance or manoeuvre (transferring plans into subplans, seconds) and stabilisation or control (transferring subplans into actions, milliseconds). The level of guidance is of utmost importance for active safety, because it is at this level where target values of speed and course are set by the driver. These target values can either be within or outside the system's boundaries of performance. The driver's decision is made based on the perception of legal limitations (e.g. speed limit), traffic conditions (e.g. existence and speed of vehicle ahead), and physical/technical limitations of the subsystem road-vehicle (e.g. friction between tyre and road).

The actual friction potential, which results from a complex mechanism influenced by road surface and specific vehicle parameters, is one of the most important determinants of the system's boundaries. The use of the given friction potential as one measure of the safety margin (Andrews et al., 1992) is mainly determined by the driver's choice of speed and accelerations (at the level of guidance) and his reactions to disturbances (stabilisation). As both friction potential and its use provide only little feedback for the driver, it may be assumed that the correct assessment of the safety margin -and thus the safe adaptation to road conditions- is very difficult.

EXPERIMENTAL APPROACH

Within the European research programmes PROMETHEUS and DRIVE, two institutions of the Darmstadt University, the Department for Automotive Engineering and the Institute of Ergonomics worked in cooperation with Porsche AG on the development of an assistance system for car drivers. The purpose was to reduce the number of traffic accidents under adverse weather conditions by supporting the driver with information, advice and warnings based on in-vehicle acquisition of data quantifying the actual friction potential.

In order to derive and validate design criteria for assistance systems supporting the driver in the adaptation of driver behaviour to friction-related driving limits, those factors producing intra- and interindividual variations of performance have to be determined. The ergonomic concept of stress and strain (Rohmert, 1986) was applied as a basis for experiments in order to distinguish between objective factors deriving from the driving task and the environment („stresses"), measures of the working person („individual characteristics"), and the effects of work within the human operator („strain") which feedback on the individual capabilities (Figure 1.).

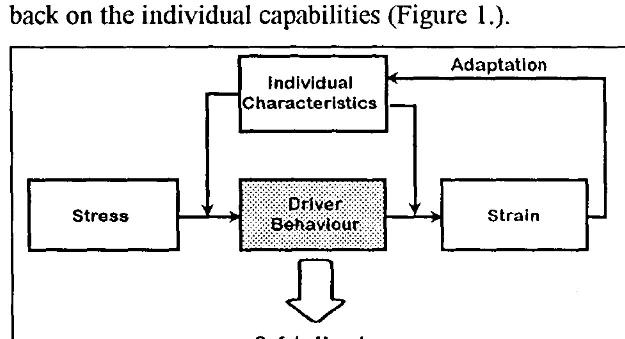


Figure 1. Concept of stress and strain (Rohmert, 1986; Breuer, 1995).

Luczak (1989) distinguishes between four categories of individual determinants of performance (Table 1.) differing in terms of temporal variability. Based on results of a literature review and preliminary experiments, Rohmert et al. (1994) determined the individual characteristics sex, age, and driving experience as relevant parameters to be varied within the collective and analysed in a between-subject design. Driving skills related to vehicle type and route characteristics were studied in a within-subject design. Relevant personality measures such as e.g. readiness to take risks and motivation were assessed for each subject. Only healthy subjects took part in the experiments. Informed consent of all subjects was obtained.

Table 1.
Possible Individual Determinants of Traffic Safety (based on a scheme of Luczak, 1989)

Category	Considered Measures
Constitution (constant in the course of life)	Sex
Disposition (changing but beyond control)	– Age – Personality – Physical Health
Qualification (long-term changes)	– Driving Experience
Adaptation (short-term changes)	– Motivation – Driving Skills (vehicle, route) – Strain – Fatigue

METHODOLOGY

Experiments were conducted in an iterative process, starting in the field (real road traffic), analysing relevant parameters in the controlled field (test track) and in the laboratory (driving simulation). Design solutions were derived and verified in experiments starting in the laboratory and leading through the controlled field back into real life conditions (Figure 2.). Table 2 presents place and scope of the different test series performed. This process ensured that for each test series with its special scope, the appropriate combination of flexibility and closeness to real life conditions of the experimental conditions were obtained (Rohmert, 1988).

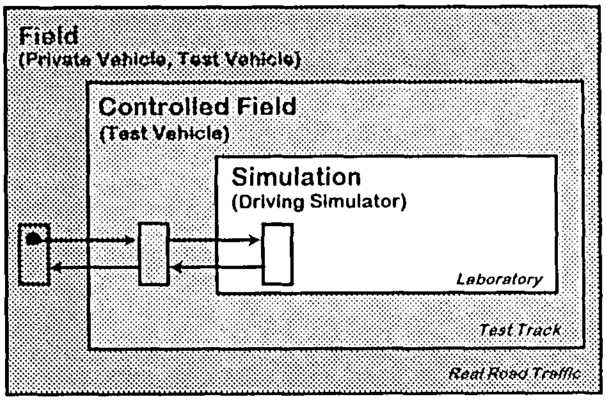


Figure 2. Work systems analysed within the iterative process of experiments to derive and validate safety relevant design criteria.

Table 2.
Experimental Design: Place and Scope of Test Series

Place of Experiments	Scope
Field	Development and Validation of Test Design and Measuring Concepts
Controlled Field	Determination of technical-physical driving limits and strain effects
Field	– Stress Effects (esp. route characteristics) – Effects of individual human characteristics – Relations between Performance and Strain
Laboratory	Effects of route characteristics on driving performance
Field	Effects of adaptation (training, motivation)
Field	Side-effects of experimental conditions
Controlled Field	Safety of design alternatives
Field	Effectiveness and acceptance of design alternatives

A test route (80 km) was composed for the field experiments in such a way that a broad variety of demands is imposed on the system driver/vehicle. It was analysed in terms of traffic situations (type of road, legal regulations, vertical and horizontal shape; Benda, 1977), road geometry (determination of curve radius and lane width), and flow of elements (rating according to Kraus & Trapp, 1987). The characteristics of the test route (59% on rural roads, 314 curves, 162 of which at slope/incline) enable the driver to reach (individual) limits (use of friction potential) without violation of traffic regulations.

104 field tests were conducted by 82 subjects (Table 3.) with two test vehicles differing in engine power, curb mass and powered axle on different road surface conditions (dry vs. not dry i.e. damp or wet; (see Horst, v.d. et al., 1993).

A test series with 3 male subjects (aged 21, 24 and 25; driving experience of 30.000, 80.000 and 120.000 km) allowed the analysis of training effects: 2 subjects drove on the test route (without motorway section, 66 km, average duration 80 min) in real traffic twice per day on 14 consecutive working days. One subject functioned as test manager before he performed 8 driving tests as subject on 4 consecutive days (Rohmert & Breuer, 1994).

Table 3.
Categorisation of 82 Subjects Who Took Part in Field Tests According to Sex (m: male; f: female), Age and Driving Experience

Age [years]	18-25		26-50		>50		
Driving Experience [km]	m	f	m	f	m	f	Σ
< 100.000	14	5	1	1			21
100.000-750.000	19	6	21	1	2	3	52
> 750.000			3	1	5		9
	33	11	25	3	7	3	
Σ	44		28		10		82

Table 4 presents an overview of the methods applied to assess driver behaviour and driver strain, containing objective and subjective as well as continuous and pre-post measures. Video analysis methodology and expert rating were specially developed and validated (Breuer & Kretschmer, 1996), the other methods were selected and adapted.

Table 4.
Methods applied to describe Driver Behaviour and Driver Strain

	objective	subjective
Driver Performance	*Activity at*: – steering wheel – foot pedals *Vehicle Dynamics*: – velocity – acceleration – use of friction potential – PAP (velocity * use of friction potential)	*Video Analysis*: – body posture – steering – gear shifting – communication – lane changing – critical situations *Expert Rating (by accompanying person)*: – speed – acceleration – cornering – headway – driving skills – compliance with traffic rules – social behaviour
Driver Strain	*Physiological indicators*: – ECG (heart rate, variability) – EMG (electromyography of selected muscles) – Tremor Activity Quotient	*Rating by subject*: – strain state at selected spots – pre/post rating of psychic and physical state (NITSCH, 1976)

Driver behaviour was assessed by objective measures obtained by in-vehicle sensors and by subjective measures derived from the video tape and an expert rating. Safety of driver behaviour is assessed by the safety margin as derived from the use of friction potential. In order to assess the safety of individual driver performance, the use of friction potential (measure for the technical-physical safety margin) is multiplied by the velocity (measure for the temporal margin, i.e. the time the driver has left for reactions in critical situations). The 90th percentile of this performance parameter (PAP: Performance Assessment Parameter) on selected sections of the test route is valued: increasing PAP leads to decreasing safety reserves of the system.

Based on the results of an ergonomic analysis of the driving task (Rohmert, 1976), indicators for physical, mental and emotional strain were selected and tested in preliminary experiments (Table 5.). Heart rate is seen as a comprehensive indicator not only for physical strain but also for mental and emotional strain (e.g. Rohmert, 1973; Luczak & Rohmert, 1974; Klimmer, 1987; Phillip, 1979) and is often used in driving experiments (see for example survey of more than 30 studies in Klimmer & Rutenfranz, 1989). Heart rate was derived from the electrocardiogram (ECG) and recorded with a rate of 1 Hz. In order to ensure inter- and intraindividual comparability work heart rate is calculated off-line by subtracting a reference value (as an indicator for the base line, obtained under standardised submaximal stress conditions after the test) from the absolute values of heart rate. The variability of momentary heart rate was also derived from the ECG in order to assess mental strain (Luczak & Laurig, 1973 by the quotient of sinus arrhythmia (Phillip, 1979).

As indicators for physical strain at human interfaces to steering wheel, gear lever and foot pedals electromyography was applied to derive electrical activities (EA) of selected muscles at the forearm (m. extensor digitorum, Rohmert & Luczak, 1973) and calf (m. gastrocnemius, Kahabka & Rohmert, 1988), both on the right-hand side. According to activation theory (see e.g. Rohmert, 1973) electrical activities of muscle groups not involved in the physical work process describe general muscle tone indicating the extent of activation. Abdominal muscle (m. rectus abdominis, Kahabka & Rohmert, 1988) and forehead muscle (m. epicranius pars frontalis, Fahrenberg et al., 1984) were selected. Tremor activity is measured between thumb and forefinger before and after the driving test to derive the tremor activity quotient (Haider et al., 1983) in order to evaluate changes in the state of emotional/mental strain due to the test.

Table 5.
Strain Measures Obtained during Driving Experiments and Primarily Indicated Strain Dimensions

	Indicated Strain Dimension		
	physical	psychic	
Measure		mental	emotional
Heart rate (HR)	X	X	X
HR variability		X	
Electric Activity (EA) of forehead muscle (m. epicranius pars frontalis)		X	X
EA abdominal muscle (m. rectus abdominis)	X		X
EA of forearm muscle (m. extensor digitorum communis)	X		X
EA of calf muscle (m. gastrocnemius)	X		

Subjective assessment of strain was obtained by pre-post measures (rating of strain-relevant and motivational state: Nitsch, 1976) and by rating of strain at selected spots of the test route (score).

RESULTS

Adaptation Effects

Intraindividual changes in behaviour and strain are evaluated at selected route sections. Due to training effects, safety relevant measures as mean speed and lateral acceleration changed over the course of tests at sections where speed behaviour is uninfluenced by other road users: the increase of acceleration (60%) is higher than the increase of mean speed (25%). The integrative strain indicator mean heart rate decreases between the second and the eighth test and then scatters around the level of the first test, reaching a higher value only in test 16, when performance measures peak. Acceleration is clearly lower on wet roads, while mean speed is not affected. The drop in performance in test 17 can be explained by significantly lower values of rated motivation and higher values of rated strain before the test.

The evaluation of driver activities did not show the development of routines for e.g. gear shifting at certain (fixed) spots of the route. Such manoeuvres and driver behaviour in general became smoother in the course of tests according to the expert rating. Tremor activity quotients (0.4-3.2, mean 1.1) are slightly higher and above 1 in the first 8 tests indicating a high level of tension due to anticipation before these tests. Motivation before the driving experiment correlates to relevant performance measures (Figure 4.).

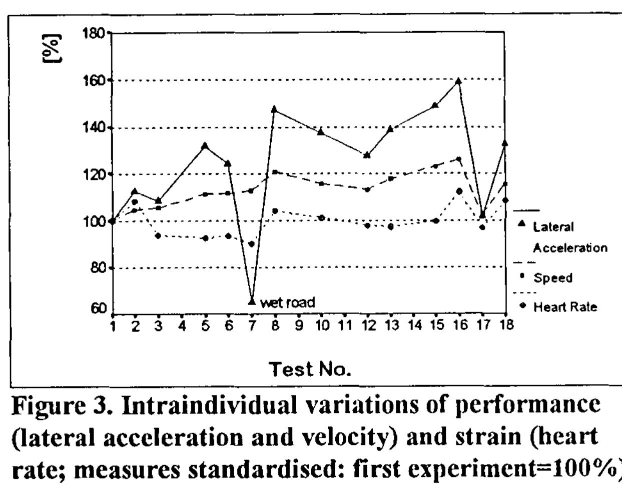


Figure 3. Intraindividual variations of performance (lateral acceleration and velocity) and strain (heart rate; measures standardised: first experiment=100%) at a difficult section (narrow rural road, 0.8 curves per 100 m, descent partly >6%, length: 1610 m) in the course of 18 field experiments on 10 consecutive working days.

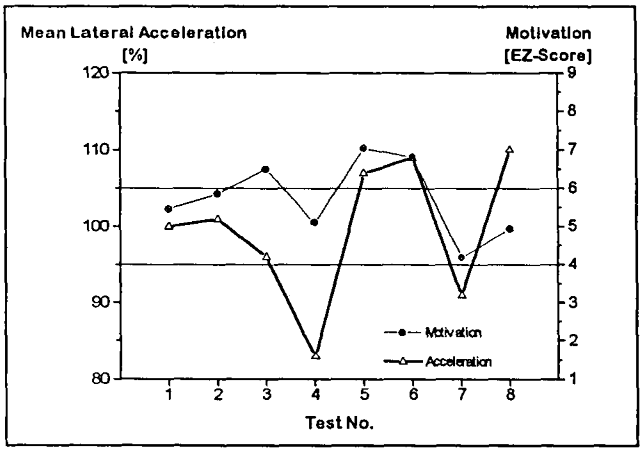


Figure 4. Motivation (EZ-score before driving experiment: between 1 and 9, normal range 4-6, NITSCH, 1976) and mean lateral acceleration (standardised: first experiment=100%) at a difficult section in the course of 8 experiments on 4 consecutive working days.

Classification of Performance and Strain

Interindividual variations of performance and strain are considerably high: e.g. the 95th percentile of friction potential use (complete test) varies between 20 and 60 percent; velocities at given sections vary in a relation of 1:2 (minimum:maximum) on rural roads. Mean work heart rate as a major strain indicator accumulates at values between 4-8 beats per minute (bpm); some subjects show higher values of up to 46 bpm (Figure 5.). Intraindividual standard deviations vary between 4 and 17 bpm (mean 9 bpm, mode 7 bpm); maximum values are within a range from 30 up to 130 bpm (mean: 65 bpm).

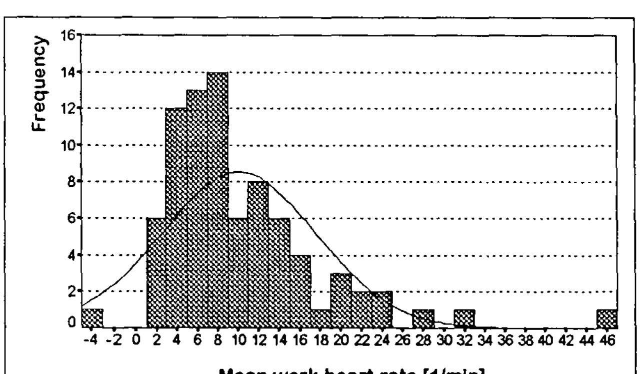


Figure 5. Interindividual variation of strain: Frequency distribution of mean work heart rate (with regard to complete driving test).

Route parameters serve to explain a fraction of the variations in performance and strain: especially curve density (number of curves per 100 m), flow of elements and road width (both rated) correlate significantly with performance and strain indicators (see detailed analysis in Breuer, 1995). A difficulty measure is derived by an additive superimposition of these measures in order to distinguish three classes: low, medium and high route difficulty (Figure 6.).

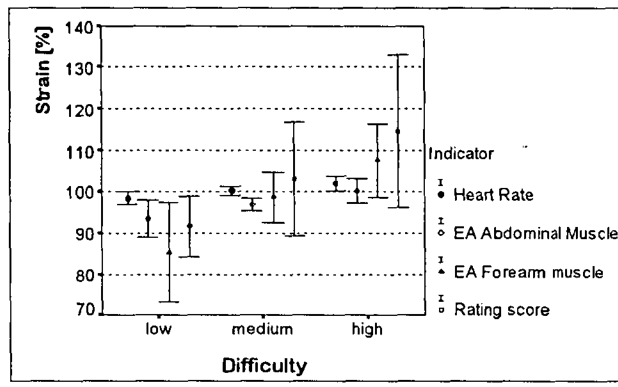


Figure 6. Mean and standard deviation of selected strain indicators as a function of route difficulty (values are presented in relation to mean during complete test; significant differences ($p<0.05$) between low and medium: heart rate and EA forearm; between low and high: heart rate, EA abdominal muscle; EA forearm, and rating score; between medium and high: heart rate).

Independent of task (route) difficulty, subjects drive within a characteristic range of performance and can be assigned to one of three distinguishable performance classes (low, medium, high). A matrix of actual values of use of friction potential and velocity can be utilised to identify the driving style with a probability of 85% (Figure 7.): for example a combination of 50% (use of friction potential) and 80 km/h (velocity) indicates a driver of the upper performance class with a probability whereas a combination of 50% and 25 km/h identifies the driver as belonging to the medium performance group. Although changes in performance occurred during the adaptation series, subjects did not leave their typical performance class.

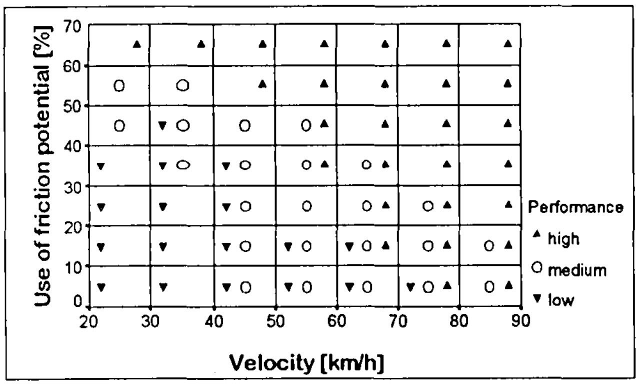
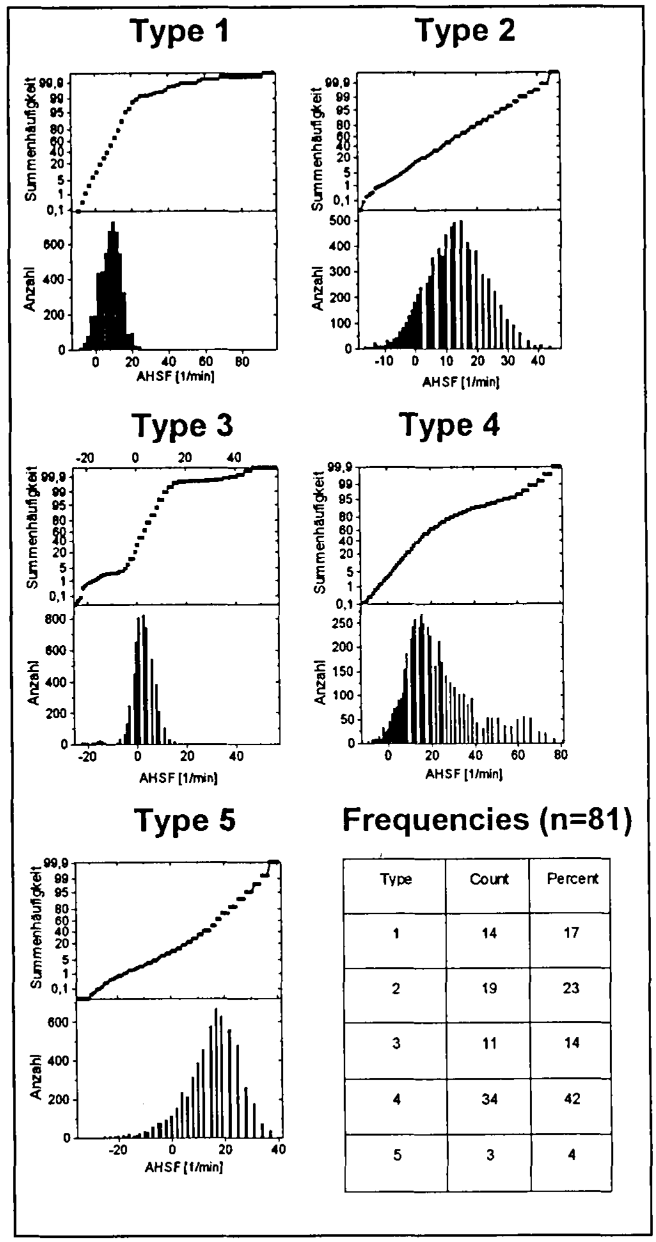


Figure 7. Classification of driver behaviour: areas in which the subjects of the respective performance classes (defined by the performance parameter PAP) produce values with a probability of 85%.

Time series of physiological indicators are classified by characteristics of their cumulative frequency distribution (Rohmert, 1993) and by their temporal variations (see examples for work heart rate in Figures 8.-9.). 5 types of frequency distributions could be distinguished, differing in skewness and courtosis. They indicate different degrees of variability and different proportions of exertion and relaxation phases. In 20 % of all field experiments heart rate increased over time, slope values of the linear regressions ranged up to 14 bpm per hour.

An example shall show how these classifications help to assess strain reactions: Only 2 subjects show increases in their heart rate of more than 6 bpm per hour, a limit of endurance determined by ROHMERT (1959) for static work: one subject with low driving experience, driving the sporty test vehicle on wet road conditions, and one expert on dry roads. Considering the level of the work heart rate (mean: 22 bpm) and the frequency distribution (type 5: mainly high values), especially the expert's strain reaction must be judged as critical because strain is at a high level, there is no balance between exertion and relaxation phases during the driving test, and therefore strain after the experiment is significantly higher than before. Subjective strain indicators prove this finding: deficiency rated by the subject (physical fatigue; Nitsch, 1976) is significantly higher after the test in comparison to the level before the test.



Type	Count	Percent
1	14	17
2	19	23
3	11	14
4	34	42
5	3	4

Figure 8. Classification of accumulated frequency counts of work heart rate time series: examples for the types 1-5 and frequencies within a sample of 81 field experiments

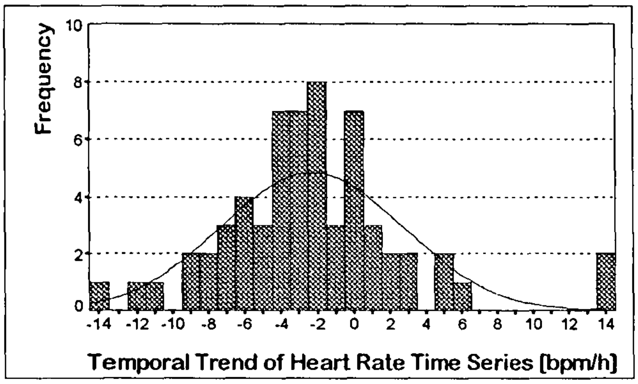


Figure 9. Frequency distribution of slope values of heart rate (linear regression of time series; values quantify change during 1 hour in beats per minute).

Influences of Individual Characteristics

Interindividual variations of performance and strain can be interpreted with individual characteristics: highest values of PAP (i.e. lowest safety margins) are produced by younger subjects with little or medium driving experience (see example in Figure 10.). Female subjects drive with higher safety margins and less strain. Motivation and personality measures (e.g. readiness to take risks, Figure 11.) seem to influence the driver's use of friction potential. Subjects with more driving experience and higher age show lower strain (Figure 12.).

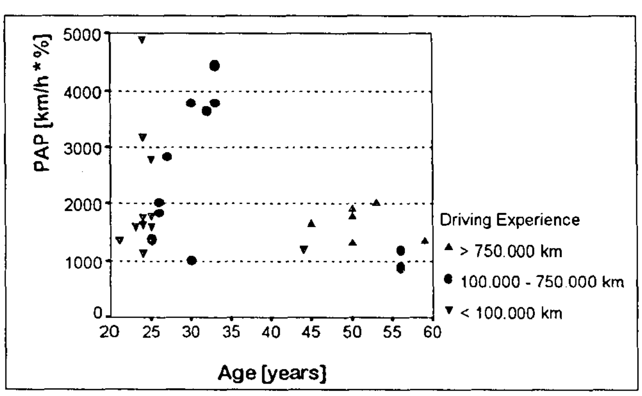


Figure 10. Performance parameter PAP (mean of 90th percentile on 5 sections) as a function of age and driving experience (sample of 31 subjects).

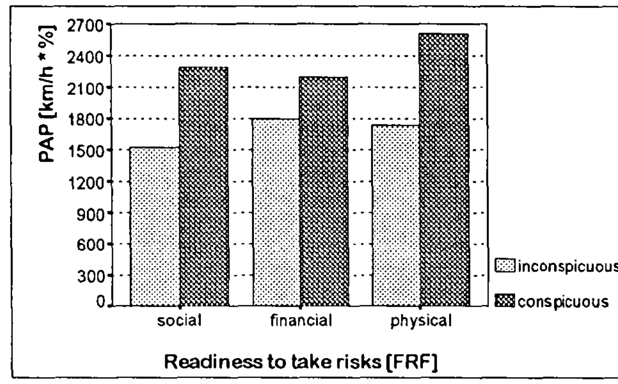


Figure 11. Personality effects: performance parameter PAP (Mean of 90th percentile at 5 difficult sections) as a function of conspicuity in dimensions of readiness to take risks (differences are significant, $p<0.05$).

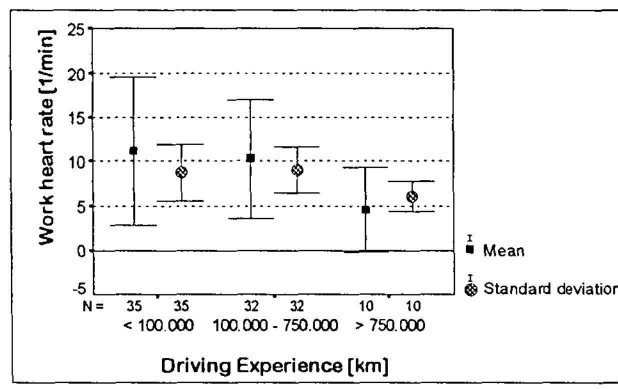


Figure 12. Mean work heart rate and its individual standard deviation as a function of driving experience.

Strain as a function of Performance

Intra- and interindividual analyses show that individual driver behaviour seems to be the most important determinant of driver strain: higher performance generally produces increases in both objective and subjective strain indicators (examples in Figures 13.-14.; detailed analysis in Breuer, 1995). High performance (i.e. driving close to the limits) leads to scales of strain which are to be avoided from an ergonomic point of view. Mean values of work heart rate reach up to 80 bpm (absolute values of heart rate are therefore around 150 bpm) over periods of several minutes and up to 46 bpm over the entire driving test (ca. 90 min) for drivers of the upper performance class. These are rather high values for a mainly informatory work such as car driving. Subjective pre-post-indicators support the conclusion that some of these drivers perform above their individual limit of endurance so that fatigue occurs and the driver's capability to act and react safely might be reduced.

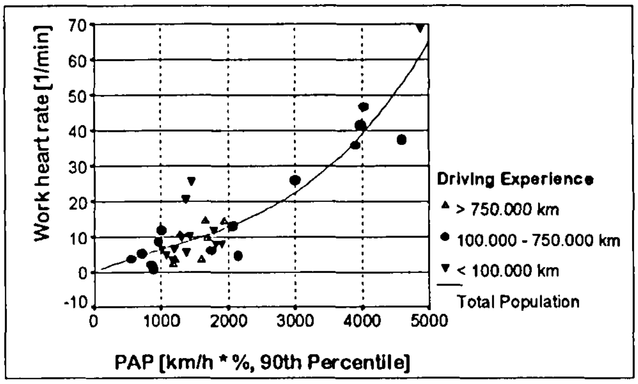


Figure 13. Strain (mean work heart rate) as a function of performance (PAP: use of friction potential* velocity; 90. percentile) and driving experience at a difficult section with cubic regression line (r^2=0,92).

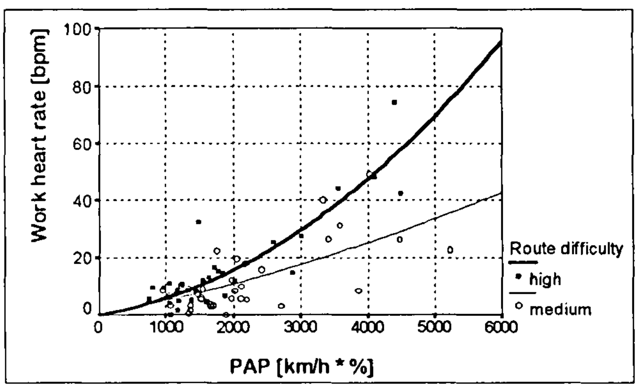


Figure 14. Strain (mean work heart rate) as a function of performance (PAP: use of friction potential* velocity; 90. percentile) and task difficulty (route) with quadratic regression lines (r^2= 0,90 for high and 0,75 for medium difficulty).

DISCUSSION

Safety-relevant driver behaviour can be assessed by the use of friction potential (technical/physical safety margin), weighed by velocity (temporal margin). Intraindividual variations are rather limited, they are caused predominantly by route parameters such as curve density, road width and flow of traffic elements and by the degree of adaptation. Interindividual differences are considerably high, they are caused predominantly by individual characteristics such as motivation, readiness to take risks, driving experience and age.

Different dimensions of driver strain can be assessed by objective (physiological) and subjective (rating of perceived strain) measures. They complement one another and together indicate human strain in its entirety, which is best reflected by the integrative measure work heart rate. Strain can be classified by time series characteristics of physiological indicators in order support the identification of ergonomically unfavourable combinations of stress, individual characteristics, and behaviour. Strain reactions are influenced by route difficulty, and determined (on a given route) predominantly by individual driver performance. It can be stated that strain generally reaches ergonomically critical scales when the driver operates close to physical driving limits.

The results from field experiments stress the fact that assistance functions are needed in order to avoid critical situations by offering feedback when impending limit conditions, and to support the driver within critical situations by producing correct reactions.

Given the fact that driving task consists mainly of non-muscular work contents (coordination of sensory input and effector control; Rohmert, 1976), the quality of human information processing is an important determinant of driving performance. Each of the main human functional mechanisms within the chain of information processing (detection, identification, decision, action; Luczak, 1975) can be supported by technical systems which amplify, extend, relief or even substitute human capabilities (Breuer, 1995; Table 6).

**Table 6.
Human Functions (Determinants of Driver Performance), Technical Functions with Regard to Human Capabilities, and Technical Support Options**

Human Function	Technical Function	Support Options
Detection, Identification	Amplification, Extension	Presentation of additional information from technical sensors
Decision	Relief	Presentation of instructions
Action	Relief	Support correct actions, prevent wrong actions
	Substitution	Technical system takes over

Assistance functions must not lead to additional strain especially in critical situations: additional information may only be offered when necessary and in adequate modality (human haptic information channel should be addressed). Warning thresholds should be adaptive to individual driving style: e.g. the performance class can be utilised as an indicator for the range of the driver's range of experience so that warnings are presented earlier (at a lower threshold) for the drivers of the lower performance class.

As the relationships between system performance and human strain are quantified for the conditions analysed and the on-line identification of relevant actual route parameters is possible (e.g. by the analysis of the steering wheel angle), work heart rate can be calculated so that the design of assistance functions can take not only driving style, but also resulting driver strain into consideration. From an ergonomic point of view an accumulation of critical situations should be avoided (e.g. by lowering warning thresholds with frequent warnings) because resulting unfavourable strain reactions possibly lead to central and/or peripheral fatigue and thus have a negative effect on the driver's capabilities.

The design of primary safety measures aiming at changes in human behaviour constitutes one basic problem: how far can changes in the work system go? Should the driver for instance always stay in charge, or should a technical system take over in certain critical situations.

Besides all legal aspects, the question of personal freedom is addressed. The driver has many degrees of freedom when operating as a car driver, especially in private road traffic: the driver may determine within wide limits when, where and how to drive. The experiments referred to in this paper show that variations on safety-relevant behaviour are high and due to differences in individual characteristics. Moreover, while safety-relevant individual capabilities (e.g., visual capabilities of taxi drivers, Ruprecht, 1993) and working times (e.g. of truck drivers, Kiegeland, 1990) of drivers in commercial and public transportation are strictly regulated and according rules are regularly enforced in Germany, both is not the case for car drivers in private traffic.

Ergonomics as a science aims to provide humane and economically efficient man at work systems by the mutual adaptation of man and work. There is a clear hierarchy of aims: safety and health of work have to be ensured first before individual satisfaction can be improved (Rohmert, 1986). From this point of view, findings to improve road safety should be applied even if they lead to restrictions of the driver's personal freedom.

Yet the (ethic) question still remains to be answered by society whether individual freedom is a higher value than safety and whether the driver's degrees of freedom are therefore indisputable elements of personal freedom or rather design deficits of the work system.

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RESULTS OF THE FEASIBILITY STUDY OF A SYSTEM FOR WARNING OF DROWSINESS AT THE STEERING WHEEL BASED ON ANALYSIS OF DRIVER EYELID MOVEMENTS

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ABSTRACT

Renault has investigated two approaches for developing an on-board system to warn of driver drowsiness. The first approach is based on the use of mechanical signals (steering wheel angle, lateral position, etc.), while the second is based on analysis of eyelid movements. The aim of this system is to warn the driver sufficiently early of the occurrence of micro-sleeps, because it is during these micro-sleep phases that the driver completely loses control of his vehicle. These moments of loss of control of the vehicle are characterized objectively by the simultaneous analysis of physiological signals and driving incidents recorded during tests on simulator and on the test circuit.

The study of mechanical signals has shown that the latter are too dependent on the geometric characteristics of the road and the kinetic characteristics of the car. According to us, such a system can only function reliably on motorways, on which driving characteristics are normalized to a maximum.

Partial analysis of thirty videos of the face, recorded during alertness tests, showed that certain eyelid parameters change when lapses of alertness occur. These observations are confirmed by numerous studies already performed on the subject. At present, we are studying the relevance of eyelid parameters for predicting driver drowsiness at the steering wheel. The initial results are positive, although they still need to be further detailed and backed up by more widespread testing.

We have developed an on-board system for automatic measurement of eyelid movements. We consider that the feasibility of this measurement is proven, but only in conditions of low luminosity (dawn, dusk, night).

Accident research data shows that a system operating by night on all types of road would be more effective in terms of warning than a system operating on motorway both by day and by night. This is why Renault, Renault V.I. and Mack Trucks have favoured this approach and have planned a series of tests to validate the complete system.

INTRODUCTION

An accident research survey carried out by Centre Européen d'Etudes Socio-economiques et Accidentologiques des Risques (C.E.E.S.A.R) mentions that in France driver drowsiness is the second main cause of fatal crashes in the truck category [1] and the fourth main cause in the passenger car category [2]. In Australia, the Australian Federal Office of Road Safety (F.O.R.S) estimates that 20% of fatal crashes in which trucks are involved are due to driver drowsiness. Another study published by the F.O.R.S points out that a great majority of truck drivers occasionally feel lapses of alertness at the steering wheel [3]. However, all those drivers who have experienced lapses of alertness have observed that these events take place gradually. Moreover, experiments carried out by Renault have shown that drivers are aware that they undergo a lapse of alertness. In the great majority of cases, these drivers stop to rest or to hand over the steering wheel to one of their passengers. However, the statistics are there. It is to provide a practical, effective solution to this problem that Renault, Renault V.I., Mack Trucks Australia and Association pour l'Aide aux Recherches Intéressant la Santé au Travail et l'Environnement (A.A.R.I.S.T.E) have been working together very closely for almost two years now on an approach using real-time analysis of driver eyelid movements.

Definition of the features of the system for warning of drowsiness at the steering wheel

Renault's Automotive Biomedical Department has based its work on data collected during experiments in real-world situations and on simulator to define the features of the system for warning of drowsiness at the steering wheel. These experiments involved having a subject drive until he left the road one or more times [4]. In these experiments, a series of mechanical and physiological signals and the videos of driver faces were recorded. The tests were carried out on approximately fifty subjects.

At the end of the test, the subject answered a questionnaire. The answers to this questionnaire show clearly that the drivers are aware of the occurrence of lapses of alertness. Some even specify that in real-world situations, they would have stopped well before their forced leaving the road.

Based on the analysis of about twenty tests by an expert, the degrees of hypoalertness have been classified qualitatively in three levels. The first level characterizes the state of an alert driver, the second that of a driver who is starting to have lapses of alertness, while level three, finally, characterizes a state of marked hypoalertness. Statistical study of the driving incidents occurring during these tests performed in real-world situation on the circuit of the Renault test centre (leaving the road, catching at the steering wheel) shows that 87% of these accidents occur when the driver is in a state classified as level 3. This level 3 is always reached gradually. Of course, the speed at which the lapse of alertness sets in varies from one person to an other, and for a given person it depends on his sleep deficit and fatigue accumulated beforehand. During these tests, we were able to observe that for subjects who were very tired before starting the test, hypoalertness level 3 was never reached until half an hour of driving. Except for pathological cases and in complete agreement with all the teams which have studied the phenomenon of lapses of alertness [5],[6],[7], we therefore consider that the driver has time to realise that his (her) level of alertness is deteriorating and that, as a consequence, he (she) should stop driving to rest, and very fortunately in the great majority of cases the driver does indeed stop or hands over the steering wheel to one of the passengers.

Unfortunately, there are situations in which, due to various constraints, the driver decides to continue driving, even though he or she is aware of the occurrence of a drop in his (her) level of alertness. The deterioration of the driver's condition can therefore reach a level such that at first he (she) is no longer very well aware of his (her) condition and that he (she) may then suffer losses of consciousness for a few seconds. During these periods of loss of consciousness, the driver completely loses control of the vehicle. Moreover, it should be emphasized that the driver does not regain all his (her) faculties immediately. This is undoubtedly why the consequences of these losses of consciousness or micro-sleeps are statistically so important even though the occurrence of micro-sleeps is a relatively rare phenomenon considering the number of hours that drivers spend at the wheel of their vehicle. Experience shows that when a person reaches a certain level of hypoalertness, he (she) can neither predict nor prevent the occurrence of such micro-sleeps.

From all these facts, we have concluded that it is not essential to warn the driver that hypoalertness is setting in, since he (she) is aware of this and will stop of his (her) own accord in the great majority of cases. The aim of the system being researched by Renault, Renault V.I. and Mack Trucks Australia is rather to warn the driver sufficiently long beforehand that once this warning time is completed, he (she) will find himself/herself in a condition in which he (she) may sustain micro-sleeps. The system being researched is therefore a decision aid system the main aim of which is to inform the driver of the maximum driving time that he or she must not exceed if he (she) does not want to risk his or her life or the lives of other road users.

Analysis of accident research data concerning hypoalertness

We describe here the main conclusions drawn by us from research data concerning accidents due to hypoalertness. In France, this data was collected and analysed by C.E.E.S.A.R on behalf of Renault V.I. in the case of trucks and Renault for passenger cars. In Australia, Mack Trucks Australia is monitoring this question based on data collected and analysed by the F.O.R.S and by the Monash University Accident Research Center (M.U.A.R.C). This data provided initial requirements for defining the priority approach to be selected by Renault, Renault V.I. and Mack Trucks Australia for developing a system to warn of drowsiness at the steering wheel.

Importance of drowsiness at the steering wheel as a factor in fatal crashes - The data collected by C.E.E.S.A.R shows that in the truck category 22% of fatal crashes are due to driver drowsiness. In the passenger car category the figure put forward is 8%. Again in the truck category, the F.O.R.S estimates that 20% of fatal crashes involving a truck are due to driver drowsiness. All these organizations emphasize that these figures are bound to under-estimate the real figures, due to the fact that if the driver survives the accident, he (she) will not be willing to admit that he (she) dozed off at the steering wheel. Moreover, for a significant percentage of accidents, the cause remains undetermined. Finally, it should be emphasized that when the driver dozes off at the steering wheel after drinking alcohol, the accident is classified in the category of accidents caused by alcohol.

Breakdown of accidents due to hypoalertness as a function of the type of road - The accident research data collected and analysed by C.E.E.S.A.R shows that for both trucks and passenger cars, accidents due to drowsiness at the steering wheel occur on all types of roads (Tables 1 and 2).

Table 1.
Breakdown of fatal truck crashes due to driver drowsiness, by type of road

Road category	Number	Percentage
Motorway	14	44%
Other	18	56%
Total	32	100%

Table 2.
Breakdown of fatal passenger car crashes due to driver drowsiness, by type of road

Road category	Number	Percentage
Motorway	21	13.4%
Other	136	86.6%
Total	157	100%

Breakdown of accidents due to hypoalertness as a function of luminosity - The accident research data collected and analysed by C.E.E.S.A.R shows that for both trucks and passenger cars, accidents due to drowsiness at the steering wheel occur mostly when the luminosity is poor, especially for trucks (Tables 3 and 4).

Table 3.
Breakdown of truck accidents due to driver drowsiness, by type of luminosity

Luminosity	Number	Percentage
Daylight	3	9.4%
Other	29	90.6%
Total	32	100%

Table 4.
Breakdown of fatal passenger car crashes due to driver drowsiness, by type of luminosity

Luminosity	Number	Percentage
Daylight	59	37.6%
Other	98	62.4%
Total	157	100%

The National Highway Traffic Safety Administration General Estimate Statistic indicates that the number of accidents due to driver drowsiness peaks between midnight and dawn. The G.E.S also underlines the fact that there is a second smaller peak which occurs in the afternoon [8]. Studies carried out in the United States between 1977 and 1980 indicate that accidents due to drowsiness at the steering wheel mostly occur between midnight and six o'clock in the morning [9], [10], [11]. A study carried out by the M.U.A.R.C shows that truck accidents on australian motorway peak during the night [12]. The link between these accidents and drowsiness at the steering wheel was not precisely analyzed.

Synopsis of the various approaches adopted by Renault's Automotive Biomedical Department to develop a system for warning of drowsiness at the steering wheel

The Renault R&D Department and A.A.R.I.S.T.E reviewed a number of approaches for developing a drowsiness warning system which would be as simple and as reliable as possible. We summarize here the results of the two main approaches on which Renault has focused over the last two years: the study of mechanical signals and the study of driver eyelid movements.

Prediction of drowsiness by means of mechanical signals (steering wheel angle, speed, lateral position) - Previous studies performed by Renault, A.A.R.I.S.T.E and Miriad Parallel Processing showed, on the one hand, that the mechanical signals depended on numerous external parameters others than alertness and; on the other hand, that the variability of these signals in a given subject and from one subject to another was high [13], [14], [15], [16], [17], [18], [19], [20], [21], [22], [23]. A study by A.A.R.I.S.T.E showed that the micro-corrections made to the steering wheel angle to correct the vehicle's trajectory are strongly influenced by the geometry of the road [24], [25], [26]. Everything suggests that the type of road also has a major influence on these micro-corrections. Road characteristics such as the lane width, road profile and curvature of turns vary significantly depending on whether the road is a motorway, a national highway or a county road. A study performed by Miriad Parallel Processing company shows that spectral examination of the signal is unable to highlight hypoalertness, even when the non-stationary nature of the signal is taken into account [22]. The complexity and order of magnitude of the interaction of these various factors on the steering-wheel angle signal seemed to us to provide a sufficient argument not to use this signal as the essential source of information to predict driver drowsiness at the steering wheel. The use of mechanical signals therefore seems to be a very complex approach due to the number of parameters other than alertness affecting these signals, and to an equal or even greater extent than lapse of alertness parameters. The use of mechanical signals to predict drowsinesss at the steering wheel therefore requires a knowledge of the road context. As a consequence, in order to develop within a reasonable time a drowsiness warning system based on the analysis of mechanical signals, it seems likely that the operation of this system would have to be limited to a single type of road, probably the motorway so as to reduce the complexity of signal processing operations.

<u>Prediction of hypoalertness from eyelid signals</u> - In parallel, the Renault R&D Department and A.A.R.I.S.T.E have investigated the possibility of using eyelid signals to predict driver drowsiness. Partial analysis of about thirty videos of the face of drivers recorded during tests on driving circuit and on simulator has shown the link between certain eyelid blinking parameters and qualitative analysis of the level of alertness by an expert [27]. The studies performed by other teams - American, Australian and Japanese - seem indeed to confirm these initial results obtained in Renault's Automotive Biomedical Department [7], [8], [28], [29], [30], [31].

However, it should not be forgotten that even though eyelid signals currently seem to be the most relevant factor for predicting driver drowsiness, it is nevertheless true that it still has to be demonstrated that these signals are adequate to develop an operational system. To answer this question, a team of three A.A.R.I.S.T.E engineers is currently working on the relevance of eyelid signals for predicting and providing a warning of driver drowsiness at the steering wheel sufficiently early and reliably. The initial results of these analyses performed on about fifty subjects are expected by early 1997.

However, the feasibility of such an approach is not obvious. The problem of image processing posed is complex. This complexity is the result of several factors, and these various factors combine with one another. Among these factors may be mentioned: variability of face morphology, variability of spectacle shapes, variations in external lighting, reflections on spectacle lenses and the wearing of sunglasses. To all this must be added the calculation time, which should not exceed about ten milliseconds in order to permit analysis of eyelid movement. This therefore represents quite a challenge.

Of all these factors, some seemed to us to have no simple solutions known at present. One of the first factors is the variability of external lighting. On the one hand, the amplitude of variation of luminosity between day and night is very great, while on the other hand sunlight poses optical problems to which there are no known algorithmic solutions. Under certain angles of incidence, the sun's luminous flux strikes directly on the lens of the camera, and it is thus impossible to form an image of the face. When the sun is on the side of the vehicle, it creates major lighting variations on the face. Moreover, when the subject is wearing spectacles and under certain angles of incidence of light, reflections occur, which can more or less partly mask the eyes, making it impossible to analyse them. Finally, there are some sunglasses for which the glass is treated to avoid letting through infrared rays. Our conclusion therefore is that it is not currently possible to develop an operational system of image analysis to measure eyelid movements in all circumstances. In selecting this approach, one must first consider the limitations to the system's conditions of operation.

Reason for the selection, by Renault, Renault V.I. and Mack Trucks, of a drowsiness warning system based on the analysis of eyelid signals

We have seen that Renault studied two approaches in parallel. The conclusion of each of these studies is that it seems difficult at present to develop a system for detecting lapses of alertness which could be operational in all circumstances. We therefore investigated, using data from C.E.E.S.A.R, the F.O.R.S, the M.U.A.R.C and the National Highway Traffic Safety Administration General Estimate Statistic, the consequences of limitation of the system's operating conditions on its effectiveness in terms of number of accidents prevented.

We saw that initially the mechanical approach limits us to use on the motorway, irrespective of the external luminosity. The eyelid blinking approach limits us initially to use with low external luminosity irrespective of the type of road.

Even if the data from the United States and Australia available to us are less specific, they seem to confirm the fact that most accidents due to driver drowsiness take place at night [8], [12]. According to the different results of accidents research survey, we decided to give priority to the approach based on analysis of eyelid blinking. The strategy selected by Renault, Renault V.I. and Mack Trucks probably makes it possible to achieve system feasibility while maintaining an effectiveness superior to that of systems based on the analysis of mechanical signals such as the steering wheel angle or car lateral position. For economic and accident research reasons, Renault, Renault V.I. and Mack Trucks Australia have chosen to implement this system initially on trucks and coaches.

Methodology used to define the alarm tripping threshold of the drowsiness warning system

We shall merely outline this method in principle here. To find the times at which the driver loses control of the situation, we work on the basis of alertness tests performed on simulator and on test circuit (50 tests). During the test, the driver's face is filmed, and the physiological signals identified as being most useful for the measurement of lapses of alertness (EEG, EOG) are recorded. As regards mechanical signals, the vehicle's lateral position, speed and steering wheel angle are also recorded.

We use the lateral position signal to detect the times at which the driver no longer controls his vehicle, and from the qualitative analysis of physiological signals we can determine whether the loss of control of the vehicle is indeed due to a lapse of driver alertness. It is thus possible to associate with the objective and qualitative physiological measurement of the lapse of alertness an objective and quantitative measurement of danger. The characteristic patterns of eyelid signals which appear during these losses of control of the vehicle are then retrieved. The aim of the system is therefore to predict, several minutes in advance, the appearance of driver states described by these eyelid parameters, which are 'associated with a physiological state in which the driver could potentially lose control of the vehicle.

Description of the eyelid movement measurement computer developed by the Renault R&D Department

We are currently investigating the feasibility of this system, which consists of a camera with automatic gain control, two infrared-emitting-diode lighting modules, an image processing card built around two C40 modules and an on-board PC.

The present system requirements are as follows: the system must be capable of indicating whether the driver's head is present in the camera's field of view, and whether the driver's head is turned away or head-on. The computer must be able to measure the actual degree of eye opening for eyes which can be detected.

Image formation and acquisition - It is well known that the quality of image formation is a decisive aspect for building a reliable image processing algorithm. We therefore tried initially to obtain a lighting intensity stable over time and as uniform as possible within the volume of space occupied by the driver's head. A source of lighting in the near infrared can meet most of the requirements for forming a video image for digital processing. Lighting in the near infrared does not affect the driver's vision. The sensitivity of the CCD in this spectral band enables satisfactory lighting intensity to be obtained with a small number of diodes. This is an important point, because it means that sufficient lighting can be obtained while remaining well within the threshold of harmlessness of infrared for the human eye. Another advantage is that the small volume occupied by the diodes makes it easier to integrate the light sources in the vehicle's instrument panel. Judicious selection and installation of diodes makes it possible to obtain lighting of sufficient uniformity and power to allow digitization and analysis of the image. Through research performed by Renault V.I. 's ergonomics department on possible installation positions for the camera and the diodes in the truck cab, positions were found such that diode reflections on any spectacle lenses would pose no problems in the great majority of driving configurations. As regards the stability of this lighting over time, from tests performed by night with the Renault V.I. road test centre we observed that the car headlights and artificial lighting do not cause any variation in the luminosity of the image if a filter is installed which allows only near-infrared radiation to pass in front of the camera lenses. One of the other advantages of lighting in the near infrared is that it normalizes the luminous intensity of the skin of the face, irrespective of the colour of the driver's skin.

To conclude, the work performed on formation of the image of the driver's face in a truck cab has made it easier to deal with the problem of analysing images. In the first place, the stability of lighting in time and space is now an acquired fact in situations of low external luminosity, and in the second place the intensity of the skin of the driver's face remains constant irrespective of the colour of his skin. The only disadvantage of lighting in the near infrared is that it diminishes the contrast between the iris and cornea of the eye and thus complicates the task of measuring the degree of eye opening. From ergonomic data supplied by Renault V.I. , we have calculated the image framing so that the face of all the truck drivers remains in the image field without having to adjust the position of either the camera or the lens.

Image processing algorithm - The main problem that remains to be dealt with, therefore, is how to develop a reliable algorithm for all possible combinations due to the variability of face morphologies combined with the variability of spectacle rims, all these variabilities themselves combining with all head positions and partial maskings of the head due to events such as hand movements, for example. We have therefore chosen to develop a multi-scale algorithm to achieve optimum independence relative to the variability of face morphologies. The algorithm breaks down into a phase of overall analysis of the face to determine a set of areas of interest in which can be found the eyes, and a phase of local analysis of the selected regions of interest to obtain a more specific identification of the eyes and measure their degree of opening.

From qualitative analysis of the films of a given driver one can observe that in a driving situation, a standard position is found in the majority of cases, corresponding to a driver looking ahead of him (her) to drive and performing a number of small adjustment motions related to movements of the vehicle and looking at the rearview mirrors and the instrument panel of the vehicle. Alongside these majority situations, there exist a multitude of situations which take up a small percentage of driving time: the driver rubs an eye, leans over to adjust the radio, turns his (her) head to look at the landscape, drinks or eats a sandwich, etc. So we have decided to use a tree-structure to represent these different situation classes (Figure 1).

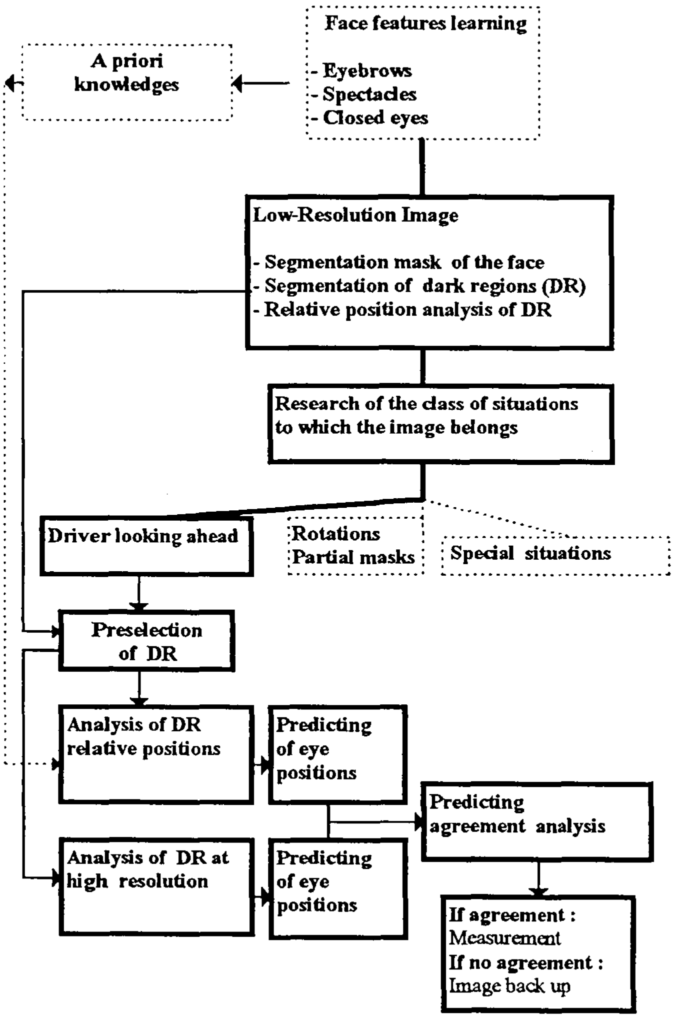

Figure 1. Principle of the eyelid movement measurement algorithm of Renault. Dotted diagram blocks correspond to the algorithm parts not yet developed.

One of the advantages of this procedure is that the code developed to analyse a given class of situations is unchanged for analysis of another class of situations. As a consequence, the algorithm inevitably improves with time. The tree structure also makes it possible to increase the intelligence of the algorithm without increasing the algorithm calculation time. This approach can also be used to quantify the percentage of time spent in a given class of situations. When we are thoroughly capable of solving this class of situation, we shall analyse other types of situations such as marked head rotations or partial maskings of the face with a hand.

Figure 2.a. Initial image of the driver of the Renault V.I. AE 520.

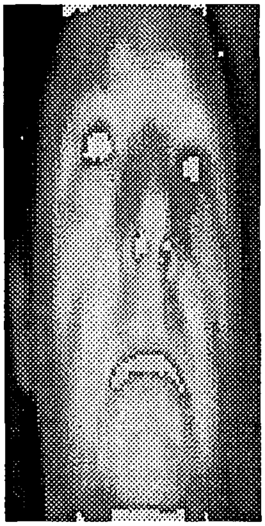

Figure 2.b. Segmentation of dark region in the face mask in the low-resolution image.

Figure 2.c. Final result of the image analysis.

Overall analysis is performed based on a very low-resolution image. The first step of the algorithm is to extract an approximate mask of the face. The second step is to extract the dark areas inside this mask (Figure 2b). The following step is to analyse like a graph the positions of these dark areas of interest relative to one another so as to classify the image in a particular class of situations.

Once the image is associated with a situation classe, the mask dark areas pass through an initial filter of morphological criteria to be selected. Following this initial selection, there are again then several candidate areas which can contain the eyes.The following step is to analyse like a graph the positions of these selected areas of interest so as to decide which areas have the best probability to contain the eyes.

Local high-resolution analysis of the areas corresponding to dark areas of the face involves calculating a series of numeric criteria over the selected regions. These criteria describe basically the distribution of grey levels in the areas of interest. The numeric criteria used enable classification of candidate regions.

One then verifies that the results of the global and local analysis are in agreement. There are numerous advantages from comparing the results obtained by two different approaches. In the first place, this twofold approach reduces room for error since, for the measurement to be validated, there must be agreement between the prediction of positions given by overall analysis of the face on the low-resolution image and by local high-resolution analysis of the selected regions of interest. In the second place, this approach provides a method for evaluating the algorithm and for improving it in a rational and systematic manner, since it is possible to detect cases in which the results of the graph analysis and the local analysis are not in agreement. In such cases, the eye location predicted by the graph analysis and the location predicted by the local analysis are incrusted on the image. The image is then saved. If desired, all the images which pose a problem during analysis of a video film can be identified in this way.

Once there is agreement between the results of the graph analysis of the face and the local analysis of regions of interest, the apparent measurement of eye opening is performed. This measurement is performed based on statistical analysis of the distribution of grey levels in a vertical cross section of the eye. At present, only the apparent eye opening can be measured in this way. We are therefore looking for ways to make this measurement of eye opening independent of the position of the head

It is planned to develop a learning phase which would make it possible to record the essential characteristics of the face, such as the wearing of glasses, and whether or not it is possible to detect closed eyes and eyebrows depending on the colour of the person's hair system . The data collected during this learning phase will serve as a fund of knowledge to improve the reliability of analysis of the graph of the face (Figure 1).

Results of evaluation of the algorithm - We have broken down the evaluation of the algorithm's reliability into an evaluation of algorithm reliability relative to various face morphologies, head movements of truck drivers and variations in external lighting.

To evaluate the reliability of the algorithm relative to various face morphologies, we recorded for several minutes eighty cases consisting of forty different persons with and without spectacles. The algorithm worked for all the persons with a satisfactory location and measurement rate. (Figure 3a, 3b, 3c, 3d, 3e).


Figure 3.a.


Figure 3.b.

Figure 3.c.

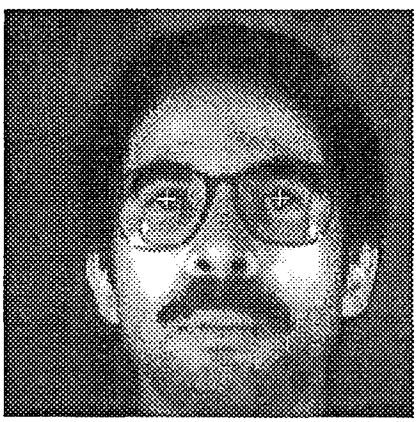

Figure 3.d.

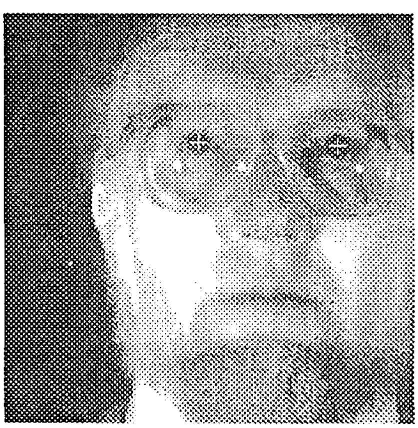

Figure 3.e. Sample of some faces process by the Renault algorithm.

To evaluate the reliability of the algorithm relative to truck driver head movements, we took recordings by night on a Renault V.I. AE truck at Renault V.I. 's road test centre. This algorithm includes no dynamic follow-up phase. This approach is adversely affected by certain rapid, major movements by the truck driver. Using our approach, the results obtained in real-world situation on trucks are equivalent to those obtained in laboratory. On a journey of approximately one hour, consisting of 60% motorway and 40% national highways and urban ring roads, the success rate of correct measurements was good enough so we can use these measurements to analyse driver alertness level.

Conclusion and prospects for development of the system

We therefore feel that the feasibility of the system for measuring eyelid movements in a real-world night situation is established for the sample of drivers filmed by us. We must therefore continue to increase the size of our database of human faces to complete our evaluation of the reliability of this algorithm for the various types of face morphologies. Some parts of the algorithm still require further development to enrich the number of situations which can be analysed by this system. A series of tests on trucks by night is planned at Renault V.I. , to acquire the image base necessary for validation of the real-time system of measurement of eyelid movements. It is projected that final feasibility of the measurement system will be achieved by early 1997.

At present, the greatest uncertainty to be solved is whether the system works for the user. In other words, it still has to be proved that solely by analysis of eyelid movements, the driver can be warned sufficiently early and sufficiently reliably that he (she) has only a given time left before entering into a state in which he (she) could suddenly and unexpectedly lose control of his (her) vehicle. A team of A.A.R.I.S.T.E engineers is currently working full-time on this project for Renault V.I. , Renault and Mack Trucks. An initial series of partial results is expected by early 1997. Subsequently, if the results are positive, a Renault V.I. and Renault endurance test vehicle will be equipped with the eyelid movement measuring and analysis system and the Man Machine Interface currently developed by Renault V.I., so as to accumulate hours of operation in order to detect cases not represented earlier and which could lead to malfunctions of the system. Renault V.I. plans to perform initial validation of this system on AE range of trucks (Figure 4). Mack Trucks Australia, for its part, plans to carry out an initial series of tests with the complete system on its Mack truck CH400 or Mack Truck CH600 product range (Figure 5).

Figure 4. Renault V.I. plans to install the drowsiness warning system on its AE range of trucks.

Figure 5. Mack Trucks Australia plans to evaluate the drowsiness warning system on its Mack Truck CH400 or Mack Truck CH600 range of trucks.

ACKNOWLEDGEMENTS

The author would like to thank Doctor Botto of C.E.E.S.A.R and Mr Thomas of Renault for their precious collaboration as experts on this project. The author would like to thank warmly Renault and Renault V.I. pilots like this the Renault's Road Test Centre and Renault V.I. Road Test Centre for their cooperation during experiments in real-world situations. The author would also like to thank Mrs Bouque and Mrs Merle for their logistic help.

This work is partly supported by the France Ministère de l'Equipement et des Transports.

The views expressed in this paper are those of the author alone and should not be interpreted otherwise.

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DEVELOPMENT OF ACTIVE HEAD LIGHT

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ABSTRACT

Recently, traffic accidents during nighttime driving have increased. Fatal accidents while cornering are a particularly serious problem. We considered the possibility that the causes of such accidents might be either a driver's mistake during a steering maneuver due to low visibility or an unsuitable watch point due to the angle of conventional fixed lighting distribution. In an attempt to solve these problems, we researched and developed the "Active Head Light (AHL)" system. It is capable of improving visibility on curves and guiding the driver's eyes to a suitable point. In this system, the beam or lighting distribution is properly controlled before the vehicle enters the curve by using road shape information from the car navigation system.

We confirmed that when using AHL, visibility on curves during nighttime driving can be maintained at about the same level as that during daytime driving and that the driver's mental stress during nighttime driving is decreased. We also confirmed that the glare level of this system for oncoming vehicles is equal to that of conventional headlights. We concluded that this system will help the driver to maintain proper driving maneuvers during nighttime driving like that of daytime driving.

INTRODUCTION

Vehicle safety technology consists primarily of preventive safety technology and damage decreasing technology. The former is to prevent an accident from happening and the latter is to minimize the damage of an accident, if one actually happens. There are some tangible results of the development of damage decreasing technology, namely collision energy absorbing body and the supplementary restraint system, which has already been adopted on many conventional vehicles. The safest vehicle is, however, the one causing no accidents. That is why practical applications of solid preventive safety technology have been long waited for. Many related ideas were presented in the Japan Ministry of Transport ASV (Advanced Safety Vehicle) project. This paper describes the application of preventive safety technology to headlight improvement, securing good visibility to reduce nighttime accidents which have been on the rise recently. The development results are already in a tangible form, one is the preventive safety devices of the Honda ASV-1, revealed in August 1995 by Honda Motors.

Analysis of Nighttime Accidents

<u>Recently Accidents in Nighttime</u> - The following is a brief description of traffic accidents happening at night in Japan. The number of nighttime traffic accidents has been on the rise recently [1]. Figure 1 shows the number of both daytime and nighttime traffic accidents in Japan on a yearly basis, assuming both numbers for 1982 are 100 each. For the 1st 10-year period following 1982, the number of fatal accidents at night rose almost 1.5 times. The death rate due to nighttime accidents is higher than that due to daytime accidents; the ratio is about 3 to 1. Figure 2 shows the traffic accident death rate categorized by the type of road and time (day or night). Deaths on a curved road at night account for the highest rate among all traffic fatalities.

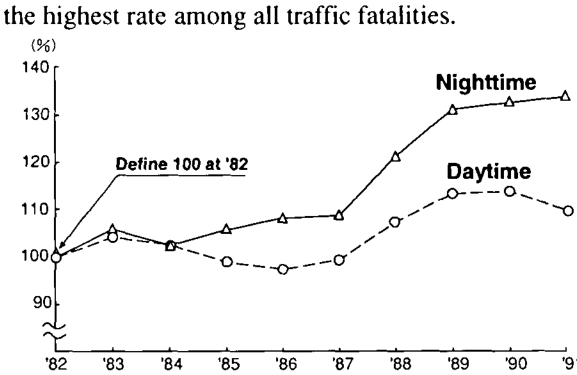

Figure 1. Fatal accident number of daytime and nighttime (in Japan).

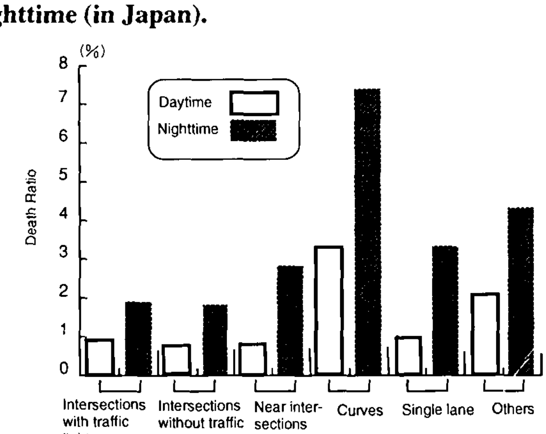

Figure 2. Result of traffic accidents investigation.

Accidents on A Curved Road - The death ratio on a curved road at night is rather high and some thoughts should be given to the reasons behind this fact. Excess speed and inexperienced driver's reckless driving are most frequently to blame, though there might be deeper reasons. More than 90 percent of driving depends on incoming visual information [2]: any marred visibility can be a critical factor triggering improper driver response. Visibility and watch points, taken by an eye-marker camera, were recorded both at daytime and night, and their comparison is shown in figure 3. Compared with at daytime, at night, it is harder, to see what is beyond a curve. Therefore, a driver tends to set his eye-point closer to himself. The Nagata's report [3] suggests that inexperienced drivers tend to set a watch point closer to themselves compared with experienced drivers, while moving around a corner and that insufficient driving skill and a closer watch point are interrelated. Hirao's report [4], on the other hand, suggests a "Td" value that is too small makes driving inconsistent when "L" is the distance between a driver and his watch point, "V" is vehicle speed, "Td" is slack time and Td = L/V. Td, particularly at night, tends to decrease, making driving inconsistent, zigzagged, and, susceptible to accidents. This is also predictable from the control law. All these imply that one of the crucial factors causing a high accident rate at night on curved roads is "Td" which tends to decrease at night.

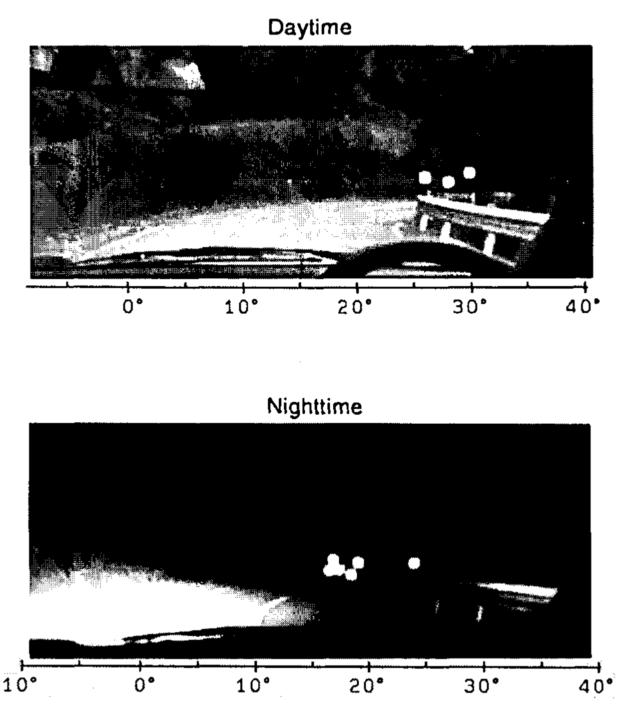

Figure 3. Eye mark on driving at right curve (same place).

Light Distribution to Reduce Accidents on A Curved Road - As shown in Figure 4, as far as the performance of conventional headlights is concerned, the visibility in a turning direction is much more insufficient than the one straight ahead. This is due to the optical paths, structured in headlights, and the intensity of light. The required performance of headlights is to make objects in a turning direction as much illuminated as the ones straight ahead. There have been a lot of studies ever done on the visibility of objects straight ahead of cars [5] and their underlying premise is "to secure good visibility, good enough to spot any obstacles, so that a car can stop before hitting them". It is assumed that the slack time for a car, running at 60km/h, straight toward a point, which is illuminated with an intensity of 3-lux, is approximately 3.6 seconds. This equation was applied to evaluate the extent of visibility in a turning direction. Figure 5 shows the relationship between conventional vehicle light distribution and their turning radius (lateral G was assumed to be 0.3G, the velocity was set at each specified curvature, the slack time was the distance, between the car and the point illuminated with at a light intensity of 3-lux, which was divided by the velocity). The evaluation resulted in much lower values for the slack time, collaborating that "Td", as mentioned before, tends to decrease at night. The light distribution which enables the above mentioned equation was determined to be the targeted light distribution. With the consideration of "the effect to direct the driver's visual focus to a correct point", the light distribution shown in Figure 6 was determined as the correct one, and the headlight system which would deliver such distribution was presented in the Honda ASV-1 as follows.

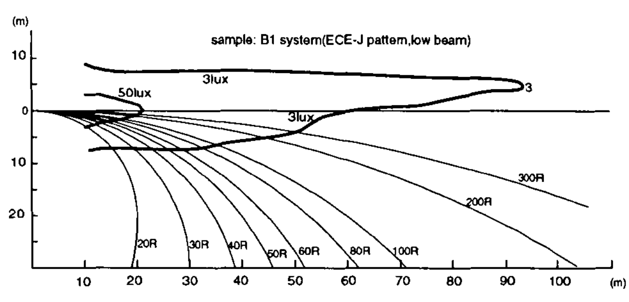

Figure 4. The lighting distribution of a current headlamp.

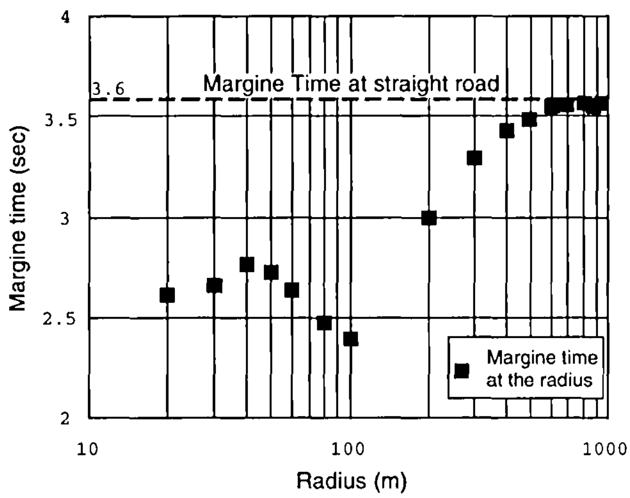

Figure 5. Cornering radius and margine time.

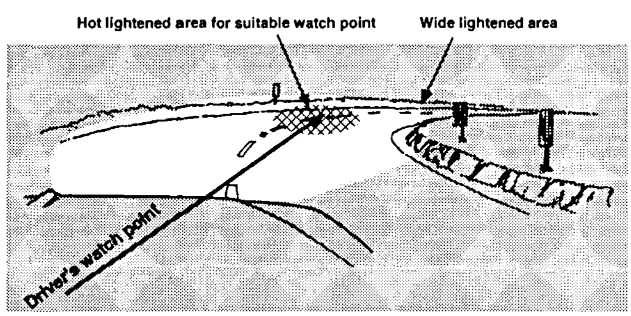

Figure 6. Proposed specification of light distribution in curve.

Active Headlight

<u>Outline</u> - In order to deliver the above mentioned light distribution on a curved road, the system shown in Figure 7 was made: a steering angle and vehicle speed were determinant factor to predict a turning radius. With the preliminary experiment's result that a driver tends to see a turning direction before actually operating the steering wheel, the technical objective was set: the objective was to control light distribution toward a turning direction before entering a curve so that a driver's visual focus is directed to the turning direction as well. For this purpose, information of curved roads which are predicted to come ahead were fed from a vehicle navigation system to control the system. Curved roads ahead were predicted by the method of corner notice and warning (Tamura et al.,1994) used in the Honda ASV-1.

<u>Specifications and the Prototype's Performance</u> - The structure of the newly developed active headlight system is described as follows:
A high intensity discharge lamp is used as the light source of the lamp and its light is reflected by the entire surface of the reflector. This optical system is used to control light beams when vehicles pass each other. The reflector is split into two, top and bottom. The top half can be rotated horizontally. The right-hand lamp can be rotated 50 degrees right and 20 degrees left. The left-hand lamp can be rotated complementary to the left-hand lamp. The top half reflector is designed to beam under the cutoff and to meet Japanese Standards of Light Distribution at any rotational angle. The reflector also does not bother a passing vehicle's driver with glare. The bottom half reflector is fixed and this part alone can reflect as intense light as conventional vehicles do when passing each other. Figure 8 shows lightened areas on a curved road. On a straight road, the movable reflector casts a beam straight ahead, improving the visibility of an object far ahead. At a curved road or a cross-section, the movable reflector on the turning side is rotated more than the reflector on the other side so that the visibility in the turning direction is improved while overall light intensity of the lightened area is kept constant.

<u>Control</u> - The main features of the control system are "control by turn signal", "control by steering angle", and "control by the information of navigation system", and any of them or any combination of them are selected on a given situation. Examples are described as follows.
<Control by turn signal>
The headlight control is synchronized with the directional signal operation. When a vehicle is about to turn at a intersection, the turning direction is lightened before vehicle's actual turn. When the steering angle goes beyond a predetermined point, the beam angle is further increased in proportion to the steering angle input. In case of the lane change operation, the beam angle is fixed so that only a neighboring line is illuminated.
<Control by steering angle>
When a vehicle moves on a curved road, the steering angle and vehicle speed are used to predict the turning radius ahead so that the beam is set as a tangent touching the turning circle. Its control speed is adjusted by steering speed and vehicle speed so that the result will agree with human senses.
<Control by information from the navigation system>
When a curved road is ahead of the car, the navigation system with the curve notice function, mentioned before, feeds a signal to the headlight so that the anticipated turning direction is illuminated in accordance with its curvature before actual turning. When a fed steering angle exceeds a predetermined value, the control will be switched to the "control by steering angle" mode.

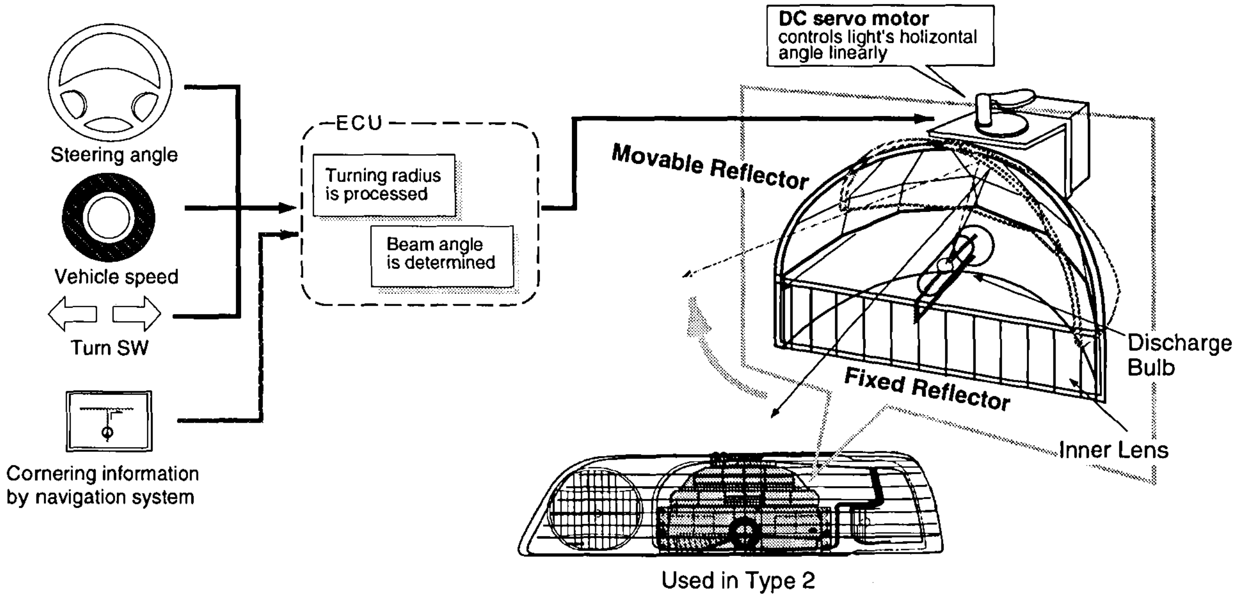

Figure 7. Structure of active headlight system.

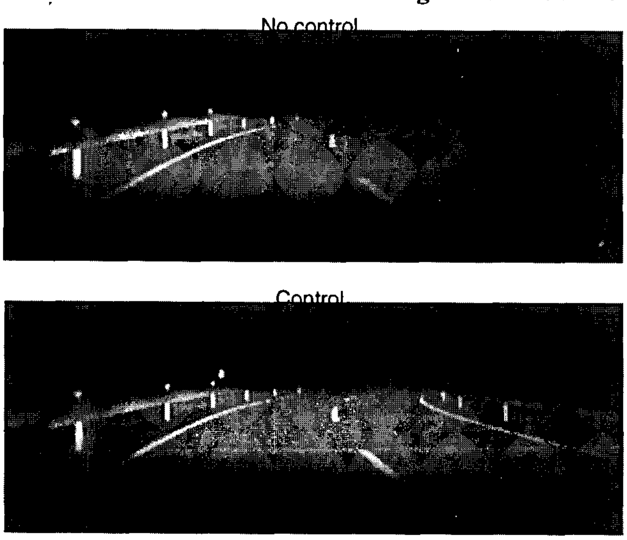

Figure 8. Light distribution image (computer graphics).

Validation

<u>Evaluation of Visibility and Glare</u> - Visibility on a curved road with a 50m curvature (hereafter called "R50") was evaluated using the active headlight system and by a conventional lamp, and the comparison results are shown in Figure 9. When the active headlight system is on, the lighted area illuminated with a light intensity of 3-lux reaches 59.6m ahead of the car, improving the visibility of the road surface and shoulder in the turning direction. When the system is off and the conventional lamp is used, the lighted area illuminated with a intensity of 3-lux reaches only 35.8m ahead of the car, signifying the big difference compared with the system. If this system of the vehicle, running at 45km/h along with "R50", is on, a driver can visually identify road conditions ahead about 2 seconds earlier than he/she can when the system is off, on the assumption that obstacles can be easily spotted in an area lighted with a intensity of 3-lux. The active headlight system can improve visibility ahead of a car and it benefits the car's driver. However it must not bother other cars' drivers with its glare. Table 1 shows how the driver of a car passing on a opposite lane felt about the glare. The glare is bit increased but remains in just acceptable level.

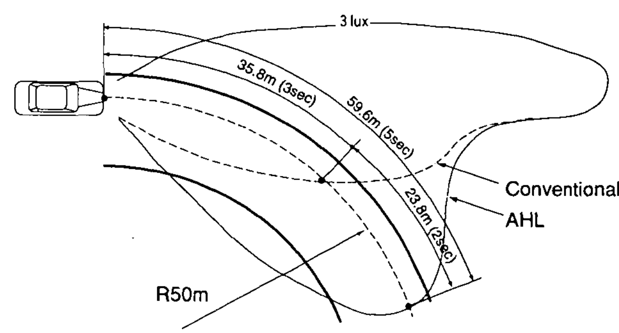

Figure 9. Light distribution of active head light.

Table 1.
Evaluation of the Disconfort-glare of AHL

	Left curve	Right curve
Conventional	7.0	4.5
AHL	5.8	4.9

(Using the de Bore disconfort-glare scale)
1:unbearable ••• 5:just accceptable ••• 9:just noticeable

Sensory Evaluation - 6 subjects participated in sensory analysis conducted on Honda's winding proving course. The test vehicle was equipped with the active headlight system which was turned on and off during the test. The results are shown in Figure 10. With this system, a driver can control the car comfortably with more confidence.

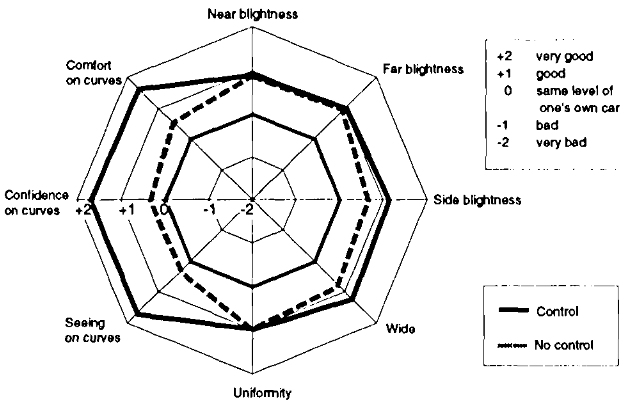

Figure 10. Subjective evaluation.

Evasive Operation - The evasive action of a vehicle, running on a stationery circle, was evaluated to assess how the system would affect driving itself. The test vehicle was driven by a subject participant on a stationary circle with 30R at a speed of 40km/h and obstacles were arbitrarily placed on the course without letting the driver know the positions. The steering angle signals were recorded by a sensor.
The test results are shown in Figure 11.

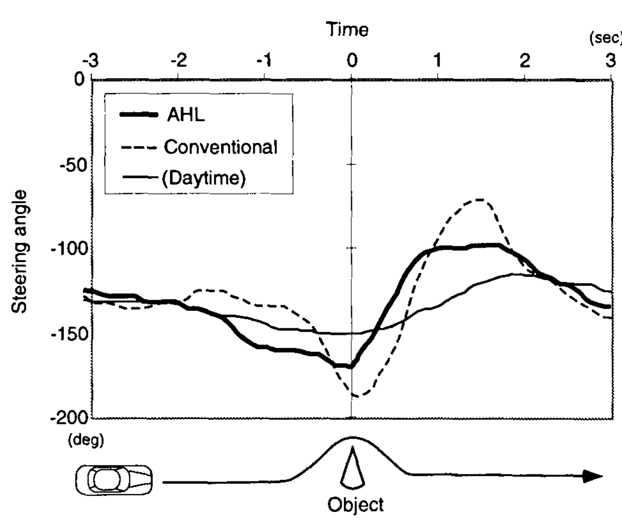

Figure 11. Driving maneuvers on the test.

With this active headlight system, evasive action becomes quicker and steering speed decreases, enabling stable steering wheel operation. However, with the conventional system, evasive action is less responsive and steering speed tends to increase, resulting in more corrective steering action and unstable driving. Driving with the active headlight system is more analogue to daytime driving than driving with conventional systems.

CONCLUSION

The active headlight system enables the following:

1) Improved visibility on curved roads while the extent of the negative effect to drivers in opposing lanes is maintained to within just acceptable level.
2) Improved driving feel around curves, making driving more solid and comfortable.
3) Quicker assessment of obstacles, if any, on curved roads, enabling smoother evasive action.

FUTURE SYSTEMS

With the technical progress of optical system designing and light distribution simulations, a highly efficient headlight with optimum light distribution will be feasible. The reliability and durability of actuators will be further improved as well. The targeted control system is to minimize the extent of glare to drivers in opposing lanes and to maximize visibility of the road ahead if no cars are in the opposing lanes. All these will be possible with image processing, vehicle-to-vehicle and road-to-vehicle communication, and radar technologies. Sensing process of road conditions could be further improved by the introduction of navigation systems, like the one mentioned above, in conjunction with advanced infrastructure. Various challenges of preventive safety technology will be addressed as vehicles and roads are more and more electronized and evolved intelligently. With further improved sensing technologies and intelligent and automatic systems, the time may eventually come for vehicles are not to be controlled directly by human visual senses or driving skill. However, whatever its form might be, visual information will be indispensable and the active headlight system will remain as a supplementary system.

CLOSING REMARKS

People may not realize the importance of headlights as long as they drive a car on a road that they know well or on a street amply illuminated. However, in Japan, there are still many places not illuminated well during the night. When driving on such a road, without the aid of illumination, drivers may feel uneasy. We will devote ourself to making headlights further evolve so that drivers can enjoy a comfortable ride under any circumstances. And he hopes, while pursuing development in this field that results can be commercialized at anytime. Many people will benefit from this system in the not so distant future with revision and solidification of the present legal system.

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DEVELOPMENT OF NISSAN'S ADVANCED SAFETY VEHICLE

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Paper Number: 96-S2-O-05

ABSTRACT

This paper describes Nissan's Advanced Safety Vehicle (ASV), which has been developed in the course of the company's involvement in the ASV Project of Japan's Ministry of Transport. While the ASV incorporates a total of twelve advanced safety systems, space limitations preclude a detailed explanation of each one. An overview is given of the individual systems, and several technologies that are distinctive features of the ASV are discussed in detail.

INTRODUCTION

At Nissan Motor Co., Ltd., we have worked hard over the years to improve safety, similar to the vigorous efforts that have been directed toward environmental protection, energy conservation and other issues.

In connection with participation in the Ministry of Transport's Advanced Safety Vehicle (ASV) Project, wide-ranging studies, including traffic analyses, were undertaken of safety systems to be incorporated in Nissan's ASV. These systems would be based on the fundamental safety technologies that had been accumulated down to the present.

The principal focus of these efforts has been driver support, involving active intervention by vehicle systems to reduce the likelihood of driver error. This thematic concern was selected in part in consideration of the projected aging of Japan's population by the early part of the 21st century, the target date for practical implementation of ASV technologies. Figure 1 lists the twelve systems selected for use in the ASV. Research activities are now proceeding toward the development and implementation of practical systems.

Keeping the early realization of these systems in

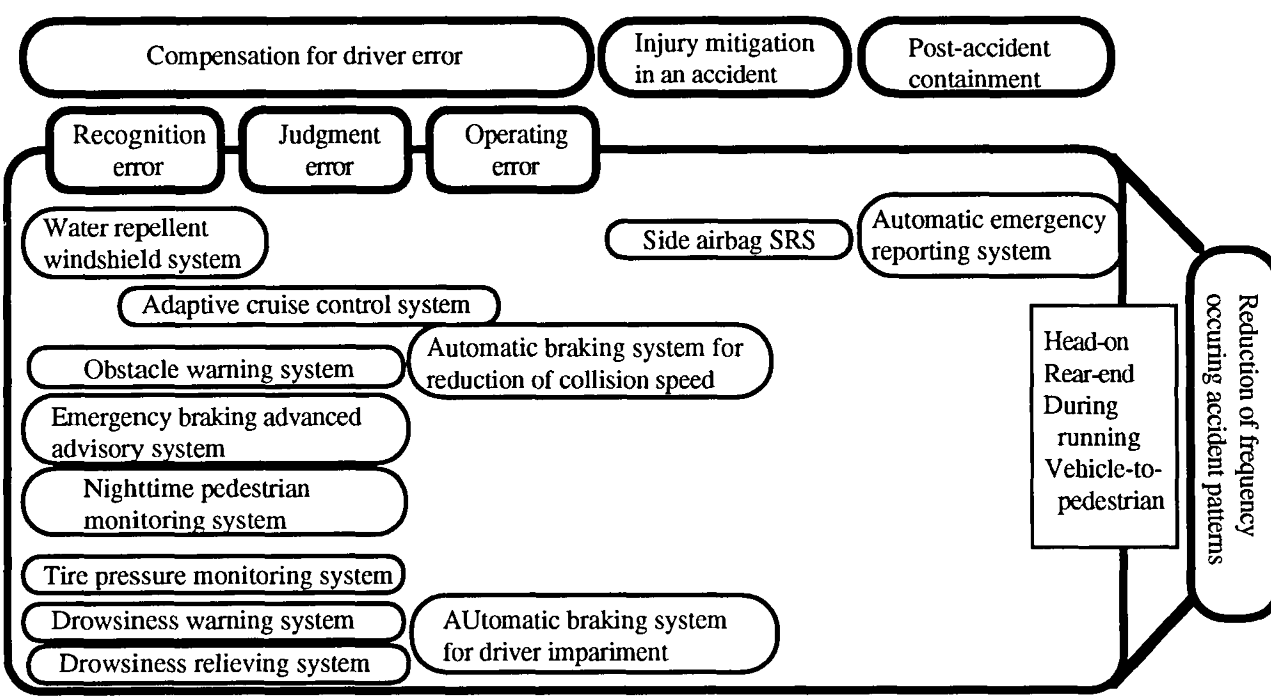

Fig.1 Safety Systems Incorporated in Nissan's ASV

mind, emphasis is also being placed on ease of installation, which will be an important issue at the time of actual implementation. The aim is to be able to install almost every system on one production vehicle.

With nine passenger car manufacturers participating in the ASV Project, it is not surprising that certain technologies are seen in common among all the companies. Among the twelve systems in Figure 1, however, the following four can be cited as examples of systems unique to Nissan's ASV which are not found on the vehicles of the other car makers.

a) A drowsiness warning system using image processing technology

b) A drowsiness relieving system using a refreshing fragrance

c) An emergency braking advanced advisory system that infers emergency braking from a sudden release of the accelerator and alerts the driver of a following vehicle

d) A nighttime pedestrian monitoring system using an infrared camera

This paper gives a brief description of all twelve systems and focuses in particular on the four systems noted above as distinctive features of Nissan's ASV. The characteristics and benefits of these four systems are discussed in detail along with various issues that will have to be addressed in order to implement them on production vehicles.

OVERVIEW OF ASV SYSTEMS

Drowsiness Warning System

This system is intended to prevent drowsiness at the wheel by detecting a driver's drowsy condition and issuing a warning to alert the person. A camera installed in the instrument panel captures images of the driver's face which are processed by a computer to determine the opening/closing pattern of the eyes for early detection of incipient drowsiness. A warning buzzer is sounded when a drowsy state is detected

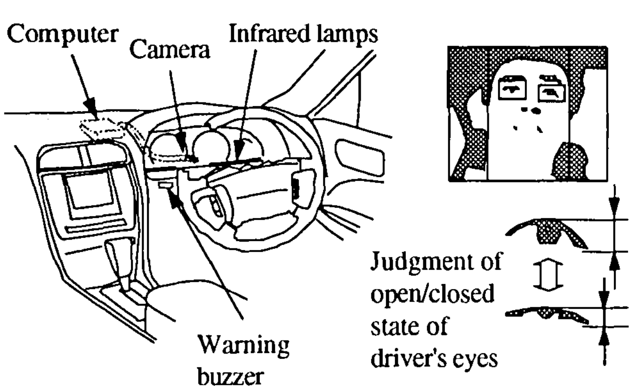

Fig.2 Configuration of the Drowsiness Warning System

(Figure 2). The system also detects if the driver is not watching the road attentively and issues a warning accordingly.

Drowsiness Relieving System

This system is designed to arouse a driver from a sleepy condition detected by the drowsiness warning system. Along with sounding a warning buzzer, it actuates the air conditioner to discharge air containing a refreshing fragrance. This discharge of a refreshing fragrance can also be activated manually by turning on the control switch if the driver feels drowsy.

Automatic Braking System for Driver Impairment

The purpose of this system is to minimize the severity of an accident if the driver remains in a potentially dangerous situation despite the warnings issued by the drowsiness warning and relieving systems. This is accomplished by applying the brakes automatically to stop the vehicle while flashing the hazard lamps to alert nearby drivers of an emergency situation. A driver can also activate the system voluntarily by a switch operation.

Water Repellent Windshield System

This system adopts a new coating technology with excellent water repellency, developed with the aim of improving visibility during rainy weather. When applied to the windshield, the coating works to form large droplets rather than an uneven film of raindrops, resulting in a clearer, safer field of vision (Figure 3). The water repellent agent is chemically bonded to the windshield glass, enabling it to maintain its effectiveness for a longer period of time than other agents now on the market.

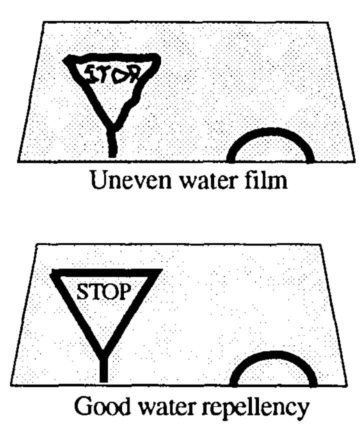

Fig.3 Comparison of Water Repellency Conditions

Emergency Braking Advanced Advisory System

The aim of this system is to reduce the likelihood of a rear-end collision by detecting emergency braking and alerting the driver of a following vehicle sooner, thereby facilitating a quicker evasive maneuver. Based on the speed at which the driver releases the accelerator, the system infers that emergency braking will immediately occur. It then illuminates the stoplights even before the driver depresses the brake pedal. It also automatically illuminates the stoplights when it receives an automatic braking command signal from the automatic braking system for reduction of collision speed.

Tire Pressure Monitoring System

This system alerts a driver to low tire pressure that has fallen below a certain specified level. Low tire pressure is judged on the basis of the principle that a tire's rolling radius, calculated from the distance a tire travels per revolution, varies according to the tire pressure. Changes in the rolling radius are detected by using the output of the wheel speed sensor attached to each tire for the antilock braking system (ABS). Comparisons are made of each wheel's rotational speed to detect any variation.

Obstacle Warning System

The system functions to detect and alert a driver to the presence of another vehicle in the forward or rearward direction or the presence of an obstacle ahead such as something that has fallen on the road (Figure 4). Obstacles or another vehicle in the same lane ahead is detected by a camera located at one side of the rearview mirror. In addition, a radar unit mounted at the front of the vehicle measures the relative velocity and distance to the detected object. Based on that information and the vehicle speed, a comprehensive judgment is made about the danger of a collision and a warning is issued if necessary.

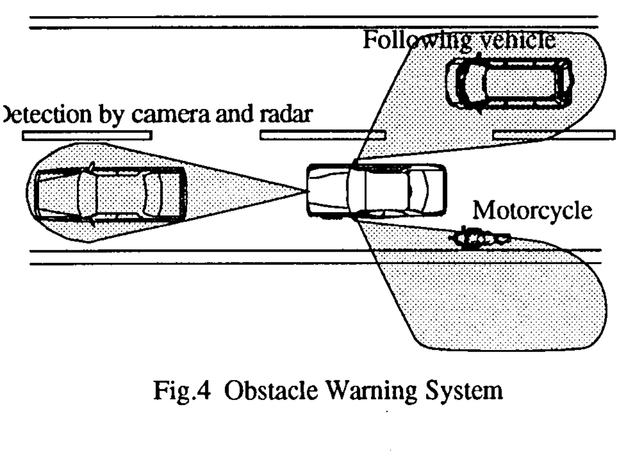

Fig.4 Obstacle Warning System

Objects in the rearward direction on either side of the vehicle are detected by cameras incorporated in the outside mirrors and a radar unit mounted on one side of the rear bumper. These devices detect the presence of an approaching vehicle when the driver wants to change lanes, for example, and activate a warning accordingly.

Adaptive Cruise Control System

This system has been created by adding a headway sensor (laser radar) and throttle and brake actuators to a conventional automatic speed control device (ASCD). It is designed to control the vehicle automatically so as to maintain a suitable headway to another vehicle. When the system detects that the vehicle is approaching too closely to a vehicle ahead, it automatically reduces the speed in order to maintain a safe distance between the vehicles.

Automatic Braking System for Reduction of Collision Speed

If the driver's braking response to another vehicle or an object on the road ahead is delayed, this system automatically brakes the vehicle to reduce the collision speed as much as possible. The system automatically brakes and stops the vehicle in the event it judges a collision is unavoidable because the driver has not taken any evasive action despite being warned by the obstacle warning system that the vehicle is approaching an obstacle ahead.

Side Airbag Supplemental Restraint System

This side airbag SRS functions to cushion the impact applied to an occupant in a side collision. When the vehicle is struck from the side, the system deploys airbags to mitigate the impact on the driver or the front passenger, working in tandem with the energy-absorbing structure of the high-strength body frame and doors (Figure 5).

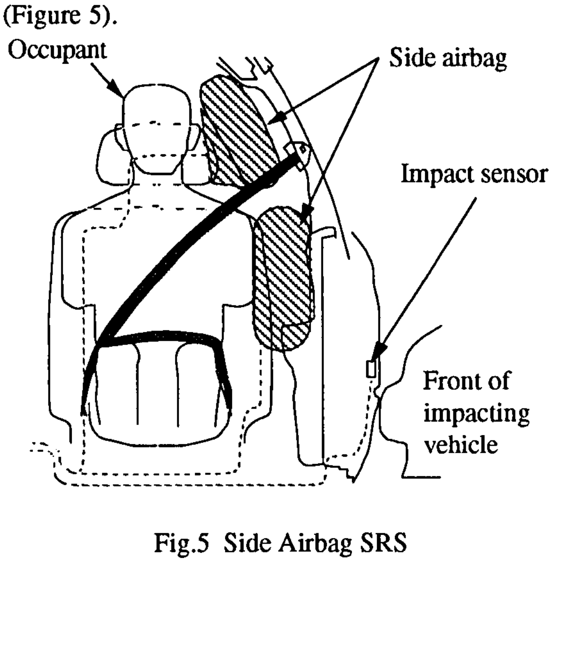

Fig.5 Side Airbag SRS

Nighttime Pedestrian Monitoring System

This system alerts the driver to the presence of a pedestrian ahead of the vehicle by detecting infrared radiation emitted by the person's body (Figure 6). The direction of a pedestrian detected by infrared sensors is indicated on an instrument panel display, making it easy for the driver to pay attention to that direction.

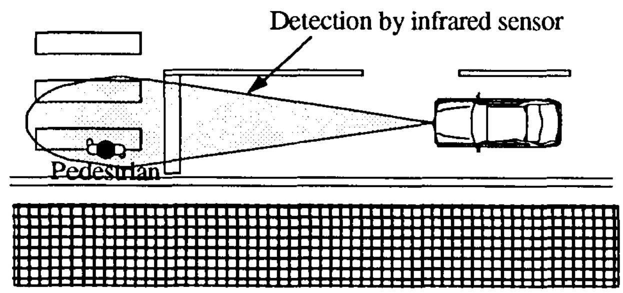

Fig.6 Nighttime Pedestrian Monitoring System

Automatic Emergency Reporting System

When impact sensors detect that an accident has occurred, this system automatically sends information on the vehicle's position, the vehicle and the driver's identity to a highway operations center via a mobile phone or some other type of wireless communications device (Figure 7). The operations center confirms the position of the disabled vehicle on a map displayed on a terminal monitor and immediately initiates rescue action, such as requesting the dispatch of an ambulance. This works to contain the occurrence of secondary disasters. A driver can also report vehicle trouble or an emergency health problem by operating a switch provided in the vehicle.

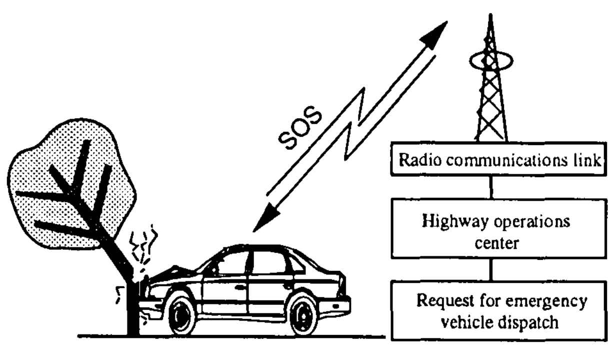

Fig.7 Automatic Emergency Reporting System

DISTINCTIVE SYSTEMS OF NISSAN'S ASV

Drowsiness Warning System

There are various possible methods of detecting driver drowsiness as outlined in Table 1. Among these different techniques, this system uses image processing technology to detect the opening/closing pattern of the driver's eyes, representing a physiological phenomenon. This method was selected because it offers the following benefits in comparison with other approaches:

1) Early and accurate detection is possible.
2) It is not annoying to drivers.

The first benefit stems from the advantage of detecting a physiological phenomenon using a sensing technique that provides a direct indication of the driver's condition. This approach is faster than indirect methods that monitor changes in steering behavior or vehicle behavior, which appear as a result of drowsiness. One typical disadvantage of methods that detect physiological phenomena is that they generally annoy drivers because sensors must be attached directly to the driver's body. This issue has been overcome by using image processing technology which provides a noncontact detection method that does not require any direct attachment of annoying sensors. As a result, this system can simultaneously satisfy the conflicting requirements noted in 1) and 2) above.

The configuration of the system is shown schematically in Figure 8. A small CCD camera installed in the instrument panel takes pictures of the driver's face, which are then processed to determine the open or closed state of the driver's eyes. That provides information for judging whether the driver is drowsy or

Table1 Drowsiness Detection Techniques

Detection techniques	Description
Physiological phenomena	Detection of brain waves, eye electric potential, pulse rate, skin electric potential, open/closed state of eyes, head inclination, posture, gripping force, etc
Driver behavior	Detection of driving operations, including steering, accelerator inputs, braking, gear shifting, etc
Vehicle behavior	Detection of vehicle speed, lateral acceleration, yaw rate, lateral position, etc
Driver's response	Periodic request for response
Driveng conditions	INference of likelihood of drowsiness from continuous driving time,time of day, etc

not. An infrared LED is also used to compensate for insufficient illumination during nighttime driving.

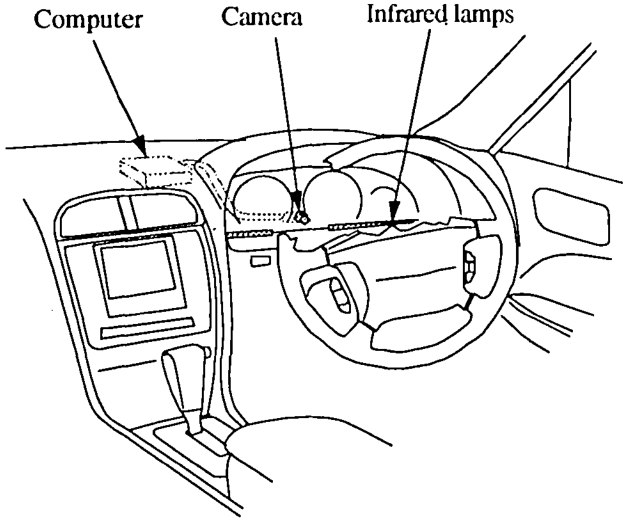

Fig.8 Drowsiness Warning System

The criterion for judging the open or closed state of the driver's eyes is shown in Figure 9. The position of the eyes is found by processing images of the driver's face, and the vertical dimension of the eyeball is used to judge the degree to which the eyes are open or closed.

Figure 10 presents experimental data showing the detection performance of this system. The results indicate that this method of detecting the open/closed state of the eyes accurately captured the change in the driver's level of alertness with elapsed time. The alertness index used here is a yardstick of the degree of drowsiness that has been devised on the basis of previous research [1]. This index is determined by totalling the points assigned to three phenomena, namely brain waves, blinking and facial expression. An index value of 9 indicates a wide-awake state whereas a value of 3 represents an indistinct mental state just prior to falling asleep.

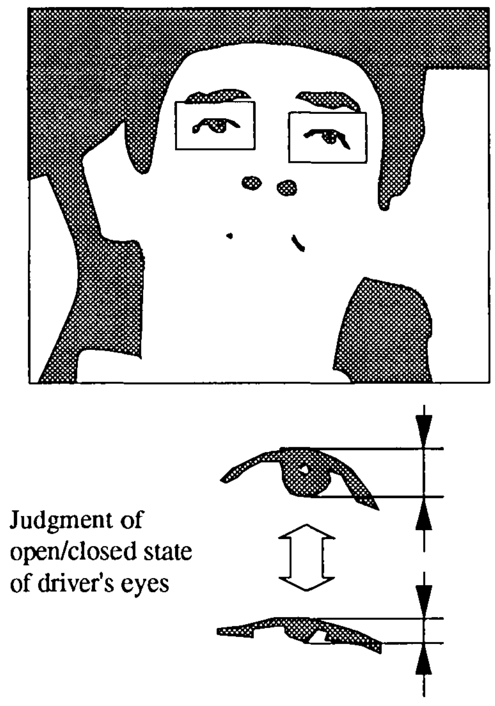

Fig.9 Drowsiness Detection by Image Processing Technology

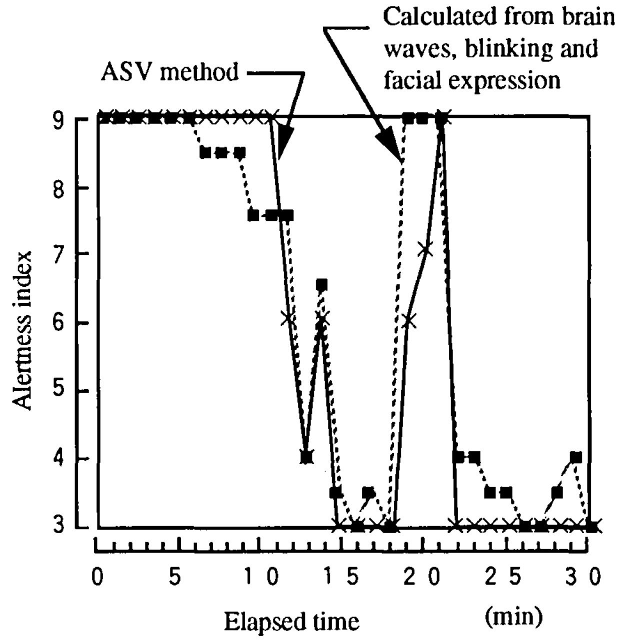

FIg.10 Accuracy of Drowsiness Detection Method in Gauging Reduced Alertness

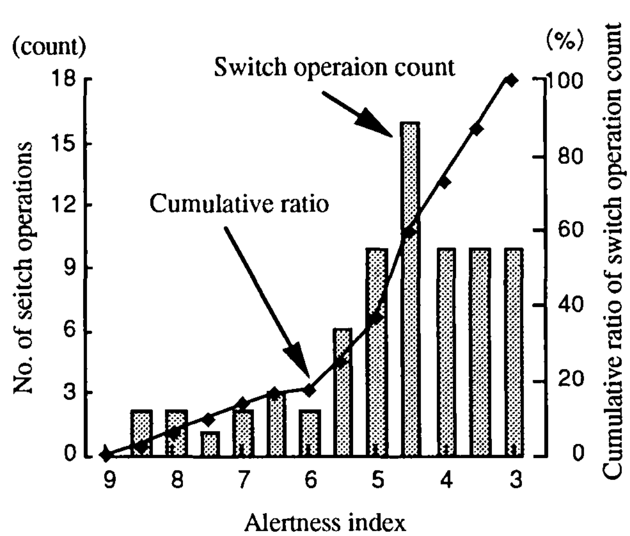

FIg.11 Degree of Consciousness of Reduced Alertness

The results of studies concerning the effectiveness of the drowsiness warning system will be described briefly. Figure 11 shows the results of a test that examined the level of alertness at which drivers became conscious of being drowsy. Less than 20% of the drivers were aware of being drowsy at the point where their alertness index had fallen to a value of 6, a level that could potentially be dangerous for driving. These results suggest that drivers are not always sufficiently aware of being sleepy.

One issue that remains to be addressed in future work is to improve the adaptability of the system to changes in the ambient light environment in the vehicle interior. Another issue that requires further research is the accommodation of differences among individual drivers.

Drowsiness Relieving System

This system is designed to increase the driver's level of alertness by even a small amount so that the person can drive safely to a rest area after a drowsiness warning has been issued. The method of refreshing the driver involves the discharge of a fragrance in addition to sounding a warning buzzer. This combined approach has been adopted because the refreshing effect lasts for a longer period of time.

Figure 12 compares the refreshing effect obtained with different approaches. In the case of either a warning buzzer or a fragrance alone, the refreshing effect diminishes with elapsed time. By comparison, the refreshing effect continues for a longer period of time when a warning buzzer and a fragrance are presented together.

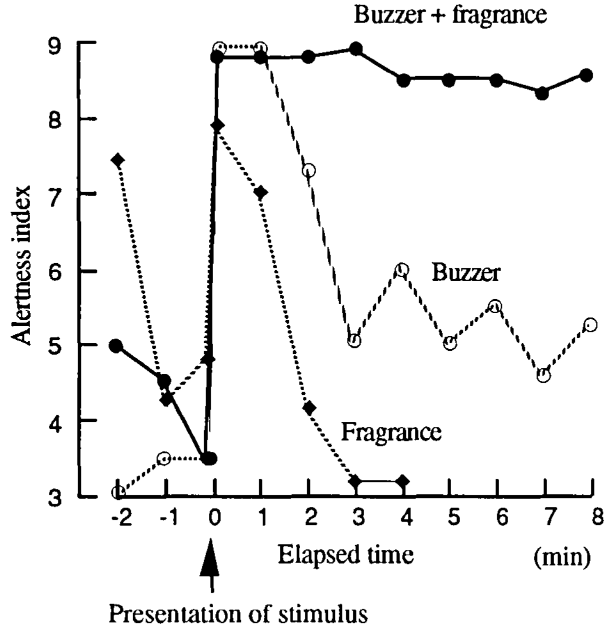

Fig.12 Comparison of Refreshing Effect

The fragrances used are menthol and lemon, which provide a refreshing and invigorating effect. These two types were chosen on the basis of experiments conducted on various fragrances.

In future work, it will be necessary to devise a measure to counter the tendency for drivers to become accustomed to the fragrance. Another issue that must be addressed is to develop a specific procedure for maintaining the fragrant material.

Nighttime Pedestrian Monitoring System

Many fatal traffic accidents occur at night even though traffic is lighter than during the daytime. Moreover, approximately one-third of the fatalities are pedestrians who are killed while crossing the street or road [2]. This monitoring system, using an infrared camera to detect the presence of pedestrians at night, is being developed as a possible solution to this situation.

Figure 13 is a block diagram showing the configuration of the monitoring system. Images taken with an infrared camera installed on the front bumper are processed to determine whether any pedestrians are present ahead of the vehicle. When the presence of a pedestrian is detected, a particular LED is illuminated the light of which is reflected on the windshield to alert the driver to the direction of the pedestrian. Pedestrian detection is accomplished by a processing operation that involves switching between two band-pass filters.

An example of an image captured with the infrared camera is shown in Figure 14, and the image obtained following processing for judging the presence of a pedestrian is shown in Figure 15. A human face has been extracted from the background as a result of the filter processing operation.

Aspects of this system that need to be addressed in the future include the establishment of a detection technique that can adapt to different driving environments and further research on a method of presenting information in an easy-to-understand format.

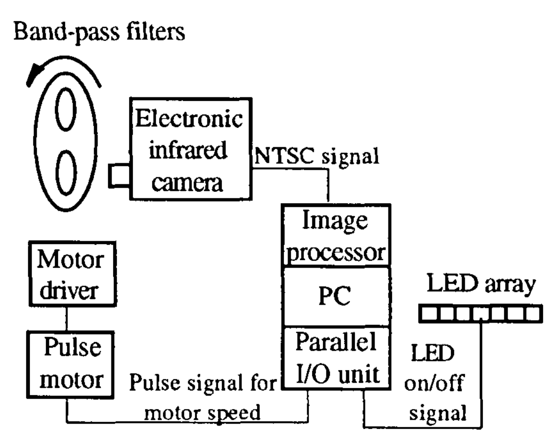

Fig.13 Configuration of Nighttime Pedestrian Monitoring System

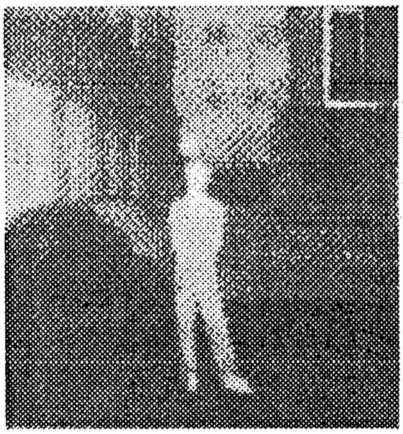

Fig.14 Image Taken with Infrared Camera

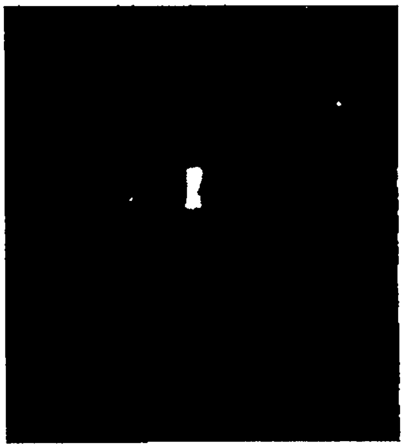

Fig.15 Image Following Processing for Pedestrian Detection

Emergency Braking Advanced Advisory System

Rear-end collisions often occur because the driver of a following vehicle failed to maintain a sufficient distance to the vehicle ahead or was daydreaming momentarily and did not brake quickly enough. One possible way of preventing such accidents is to illuminate the stoplights of a vehicle more quickly so as to give other drivers advance warning of one's braking intention. That is the purpose of the emergency braking advanced advisory system adopted on Nissan's ASV.

In general, before a driver applies the brakes in an emergency situation, the person releases the accelerator quickly. This system infers the driver's intention to brake suddenly from the speed at which the accelerator is released. The operating principle of the system is illustrated schematically in Figure 16.

One issue that needs to be dealt with in future work is to develop a more accurate detection method. It will also be necessary to make sure that the system conforms to legal and regulatory requirements.

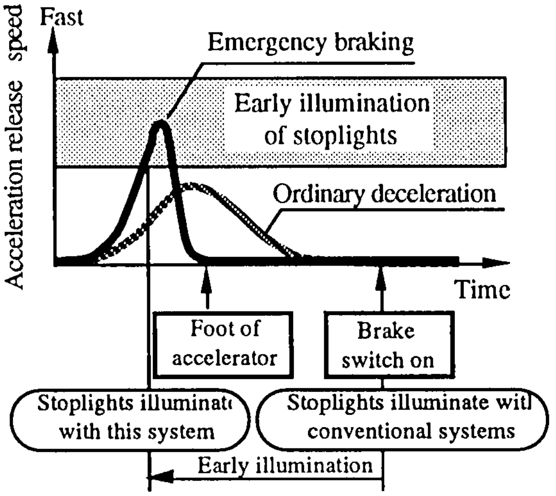

Fig.16 Method of Inferring Emergency Braking

CONCLUSION

Many technological possibilities have been identified as a result of participating in the ASV Project and developing Nissan's ASV, which embodies various advanced safety systems. At this point, there are still numerous issues that must be resolved before these advanced safety technologies can be implemented on production vehicles.

Besides working to resolve the issues peculiar to each individual system, research will also be advanced on human-machine interfaces to address an aspect common to all ASV systems. That aspect can be summed up in the question of how to minimize the driver's workload so as to augment the benefits of advanced safety technologies. The aim of these efforts is to achieve even safer vehicles as early as possible.

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[1] Y.Seko, et al., "Driving Behavior Under a State of Reduced Alertness," Preprint of JSAE Scientific Lecture Series, Vol. 841, 1984 (in Japanese).

[2] "Analysis of Traffic Accidents by Type," Research Report by the Traffic Accident Analysis Center, 1994 (in Japanese).

DEVELOPMENT OF WARNING STRATEGIES AND DRIVER-VEHICLE INTERFACES

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Paper Number 96-S2-O-06

ABSTRACT

It was the aim of this research to develop a warning strategy for a driving assistance system which gives information to the driver about a pending critical driving situation relating to the friction potential.
Driver behaviour was analyzed in order to determine the driving strategy for different drivers under various conditions related to safety limits. The knowledge achieved was used for the development of warning strategies and driver-vehicle information interfaces.
To estimate driver behaviour the ergonomic concept of stress and strain was applied. Stress (task and environment) and performance (e.g. vehicles acceleration and speed) are measured in order to identify relations between task parameters and the behaviour of different subjects.
The behaviour of the test persons was evaluated on different levels depending on parameters of the task (route) and the human being. It can be assumed that driver performance is influenced by certain route properties.
The safety margin, respectively the use of the friction potential in the lateral direction, was primarily used as the most important safety parameter for the development of information, support and warning strategies.
The human information processing and control mechanisms used during car driving were considered in the search and for evaluation of possibilities to inform the driver. The visual, auditory and haptic (kinesthetic and tactile) channels have been considered.
An active accelerator pedal yielded the best result to transmit an information to the driver. The most detectable signals were the combination of haptic and acoustic signals, which gave the best support for the driver.
For driver groups of different experience, different information levels, which were based on the individual use of lateral acceleration potential, were defined and used for warnings.
This research project has been funded under the EUREKA programme PROMETHEUS by the German Federal Minister for Research and Technology and contributed to the ROSES project of the EU-funded DRIVE II programme. The work was conducted within a cooperation of two institutions of Darmstadt University (Technische Hochschule Darmstadt THD): Department of Automotive Engineering (*fzd,* Fachgebiet Fahrzeugtechnik, Prof. Dr.-Ing. B. Breuer) and Institute of Ergonomics (IAD, Institut für Arbeitswissenschaft, Prof. Dr.-Ing. W. Rohmert) with Porsche AG, as industrial partner.

INTRODUCTION

The German development of road-traffic safety is shown in (Figure 1).

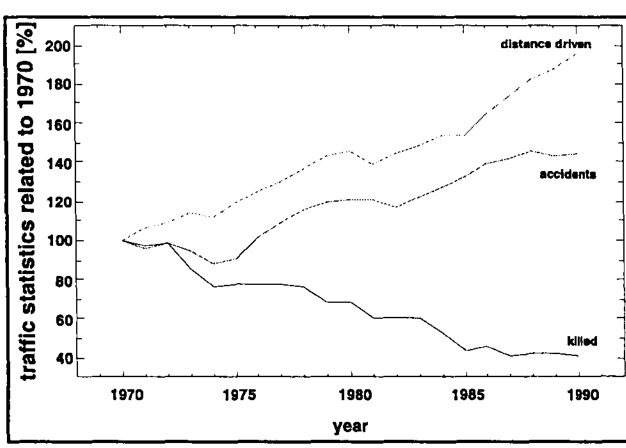

Figure 1. Traffic accidents and distance driven i Germany related to 1970 (modified from [1])

Though the total driving distance rose steadily during the passed years the number of those killed by traffic has been noticeably reduced. The reason for this trend is the improvement of vehicle safety and roads. So far measures to increase active safety which means accident avoidance

could not yet prevent the rise of road traffic accidents depicted in figure 1, [2]. By chassis and body improvements modern cars can easily and comfortably be steered up to almost the physical extreme, the limits of tyre/road friction. On the other hand, the driver receives increasingly less information while approaching the friction limit. This information deficit often causes accidents.
An active intervention in the steering and braking systems of the vehicle was not intended; instead active safety was to be enhanced through acceleration of driver´s reaction by means of information resp. warning.
Earlier driver actions bear a big potential in thus respect, Figure 2.

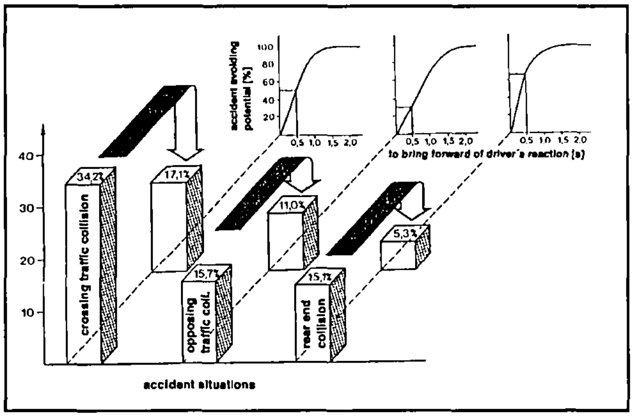

Figure 2. Accident avoiding potential by acceleration of driver´s reaction (modified from [3])

METHODES TO ANALYZE DRIVER´S BEHAVIOUR

The analysis of driver behaviour to determine the driving strategy of the driver related to the tyre/road adhesion was placed in the ergonomic part of this investigation based on the concept of stress and strain [4]. The theory of this concept ensures that the measurement, evaluation and the design of human labour takes into consideration man's individuality. Stresses derive from the task and the environment, they are external measures and can be found in an objective analysis. The possibly regulated or reactive activities supply measurable performances. Within the person, stresses relevant to behaviour lead to strain, dependent also on the individual's characteristics, abilities, skills and needs. Strain was evaluated in order to determine the feedback which influences man while working and its contribution to the systems reliability. Strain could be experienced and rated subjectively by the person; it could also be quantified by objective measures from e.g. physiological data. The measuring concept was verified in more than 60 preliminary driving tests.
In order to develop warning strategies that consider the actual driver behind the steering wheel with her/his driving strategy and driver strain ("driver status"), it was intended to find on the one hand measurable parameters of driver´s behaviour that serve to identify driving style and on the other hand those that indicate driver´s strain. Therefore relations between certain task parameters (vehicle and route properties), driver characteristics, abilities, skills and needs, driving behaviour and driver strain had to be determined.
The test vehicle was a front wheel driven compact class car with a 60 kW 4-cyl-si-engine. The car was equipped with ABS and devices for vehicle dynamics and physiological measurements, Figure 3. This equipment was hidden as well as possible to minimize its influence on test persons.

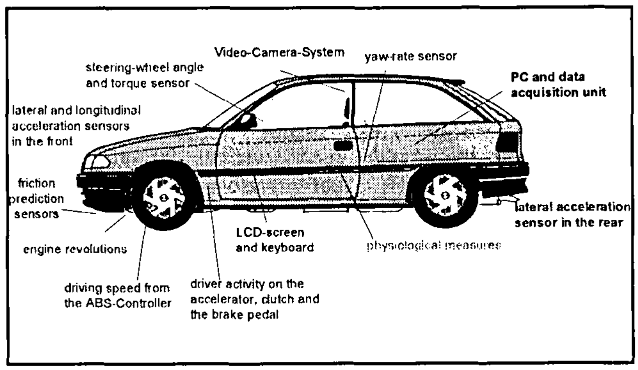

Figure 3. Test car equipment

The THD test route - which was composed in such a way that a broad variety of demands was imposed on the system vehicle/driver - consists of different modules and covers a distance of 80 kilometers. It was analyzed in terms of traffic situations (type of road, legal regulations, vertical and horizontal shape), road geometry and surroundings (determination of curvature, width of lanes).
To classify subjects, those human characteristics, abilities, skills and needs, that have an influence on driver behaviour and strain, were determined by evaluating literature and results of preliminary tests.
Some authors assumed sex-related differences in attitudes towards car driving in general, rules and own driving skills, driver behaviour and strain. There seems to be a relation between age and risk taking behaviour, physiological capacities such as visual perception and reaction time and accident risk. Driving skills improve with driving experience quantified by the number of kilometers driven: e.g. higher efficiency of information processing, reduction

of strain. Driver personality seems to play an important role in regard to certain dimensions of driver behaviour such as risk taken, aggressive interactions, emotional driving, self control, dominance. Three personality measures are taken to identify "normal" and "conspicuous" driver personalities.
During the test drives, test persons were instructed to drive as they normally would (normal driving behaviour).

DATA ANALYSIS

The tyre/road-friction related safety margin [5, 6] determines the potential for the manoeuverability of the car beyond the amount of friction necessary for the actual driving situation, Figure 4.

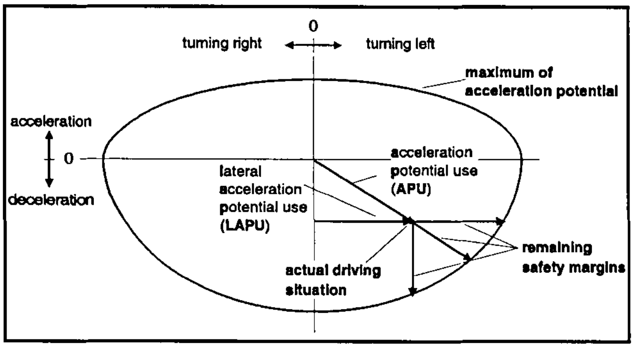

Figure 4. Definition of acceleration potential use

Under vehicle dynamics point of view, the maximum and the utilization of friction can be regarded as the most important safety parameters for the individual vehicle/driver system. Driver behaviour can be described by the use of the vehicle's friction potential. The superimposed body related accelerations (lateral and longitudinal) can serve to approximately quantify the use of acceleration potential by correlating them to the maximum possible values of lateral and longitudinal accelerations, which were measured for the test vehicle in separate experiments (elliptical curve in figure 4) and used as steady state curves for the quantification of the drivers behaviour.

Behaviour of the Drivers Collective

For the assessment and the differentiation of driver behaviour the actual superimposed acceleration related to the maximum possible accelerations for this angle was determined. Therefore the distance to acceleration maximum, figure 5, was calculated for every measured point and given in percent.

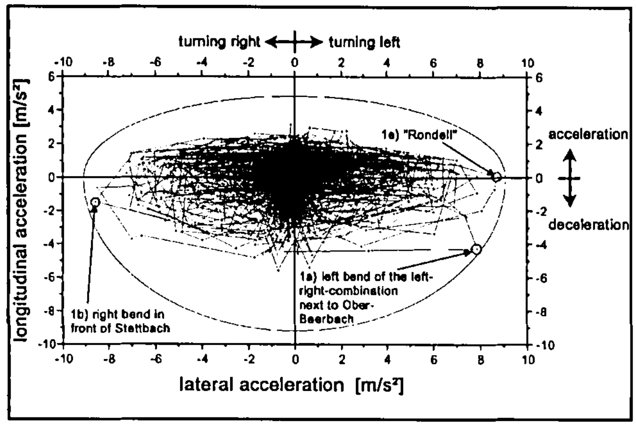

Figure 5. Approximated curve of possible body accelerations together with measured data of a complete test for a "sporty" driver, test drive 104 (TD 104)

With these values the arithmetic mean value and the 95th percentile of the test drive duration were worked out, Table 1. Within the drivers collective, females seem to drive with smaller use of acceleration potential, and drivers with higher driving experience are driving with higher use of acceleration potential.
The higher the driving velocity, the larger become the reaction distances within a given human reaction time. Furthermore higher reactions ask for larger circumferential tyre forces thereby reducing possible lateral forces. Figures 6 and 7 are showing the very different behaviour of two drivers regarding lateral acceleration versus driving speed.

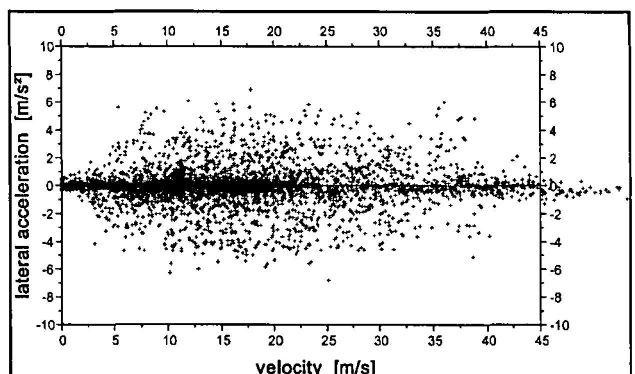

Figure 6. Lateral acceleration as function of velocity TD 89

Regarding the outer shape of accelerations measured in TD 104, figure 7, this typical "lemon contour" with relative and absolute high lateral acceleration potential use at

Table 1.
Minimum, maximum and mean values of 95th percentile acceleration potential used by the test persons (first in cells: minimum-maximum in [%], in brackets: number of available vehicle data)

	age 18 - 27		age 28 - 50		age 51 -		
driving experience	male	female	male	female	male	female	mean
0 - 70.000 km	26,8-42,5 33,9 (10)	29,5-31,1 30,1 (4)				31,1 (1)	31,7
70.001 - 750.000 km	28,6-42,8 37,9 (3)	35,8 (1)	28,9-50,2 42,6 (5)	33,5-36,0 34,8 (2)	31,6 (1)	27,8-33,3 29,7 (3)	35,4
> 750.000 km			36,3-38,6 37,5 (4)		30,3-35,4 33,6 (5)		35,6
arithmetic mean	35,9	33	40,1	34,8	32,6	30,4	
mean all: 34,5%; mean female: 32,7%; mean male: 36,2 %							

lower speeds is defined as "normal". In contrast, the test person of TD 89 drove with nearly similar lateral accelerations over a large range of velocity.

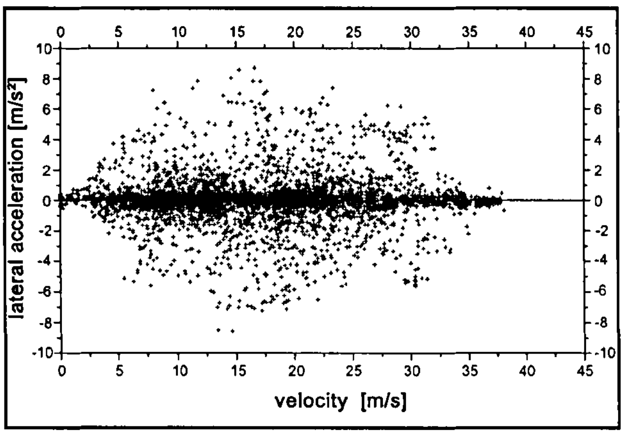

Figure 7. Lateral acceleration as function of velocity, TD 104

<u>Critical Driving Situations</u>-Critical situations caused by absolutely high acceleration potential use for the single driving vehicle/driver system are considered as well as conspicuous single events, figure 5, <u>Figure 8</u> and <u>Figure 9</u>.
Corresponding video recordings are identified and related to an incident. Strikingly often the winding parts of rural roads are the relevant route parts.

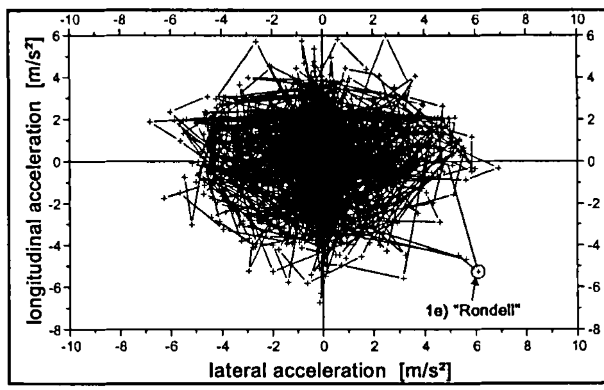

Figure 8. Longitudinal vs lateral acceleration, TD 89

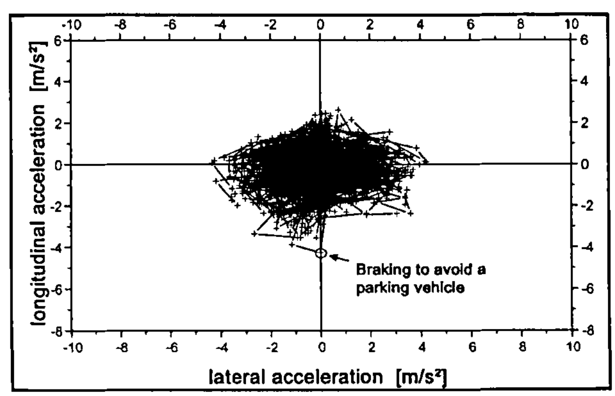

Figure 9. Longitudinal vs lateral acceleration, TD 82

Experimental results like in figures 5, 8 and 9, are only precisely characterised if an acceleration vector angle is added. For better separation of lateral and longitudinal dynamics these angles are transferred into one quadrant. The angle position "0°" means pure lateral dynamic, the angle position "90°" pure longitudinal acceleration. Figure 10 contains the angle positions of 103 measurements of high acceleration potential use.

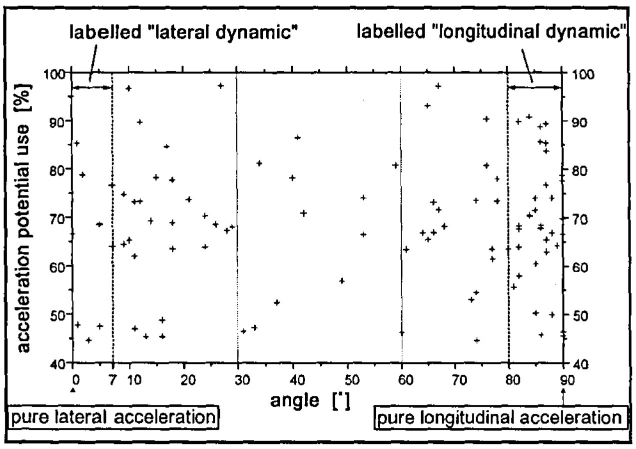

Figure 10. High acceleration potential use (103 points) of 13 test drivers as function of acceleration angle

Three ranges can be defined:

0-30°: Range of high acceleration potential use, lateral accelerations dominating, medium density

30-60°: less densely occupied range, lower maximum use

60-90°: high density of situations, especially in the area of pure longitudinal acceleration

Considering the manifold possibilities of acceleration combinations, there is a striking increase of friction use in longitudinal dynamic situations in contrast to pure lateral dynamic. In most cases there is a clear superposition with longitudinal accceleration. The sum of pure longitudinal and pure lateral incidents amounts to about half of all incidents considered.

The amount of longitudinal dynamic situations with high use of the acceleration potential supports the assumption that these are situations not unknown to the test persons and normally do not end in a accident. Situations which could become critical (curve driving on free roads with high speed) are much less frequent. Junctions are normally crossed with low speed and are therefore not critical even at high lateral acceleration use.

Personality characteristics like readiness for taking risks, degree of driving experience and motivation of the test driver are playing an important part [7].

The difficulty of the test route section is another most important influence. The **L**ateral **A**cceleration **P**otential **U**se **LAPU** characteristic, Figure 11, reflects strong road influences independent of driver types.

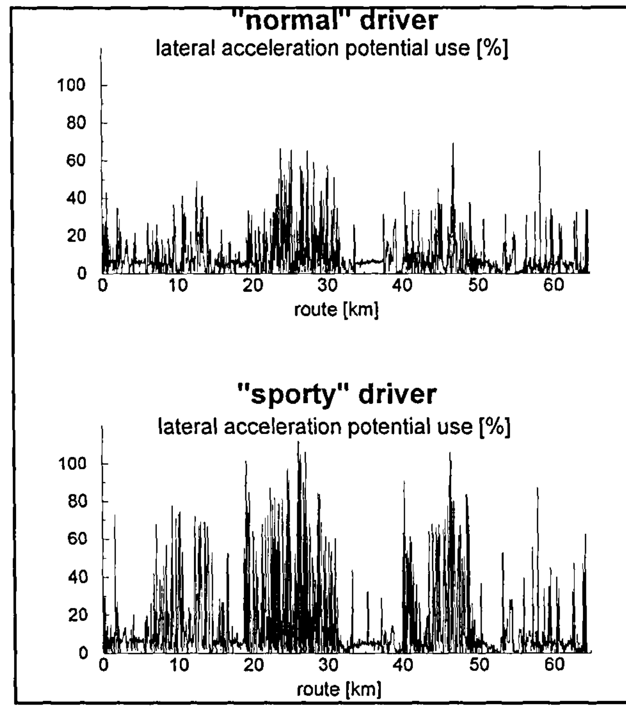

Figure 11. Friction potential demand over the test route for different drivers

The high use of the LAPU for the "sporty" driver could be the result of higher motivation and degree of experience. Finally it can be stated that the type of the test route is responsible for the pattern of the LAPU. On the other hand the absolute friction orientated safety margins are very driver dependent [8].

DEVELOPMENT OF WARNING STRATEGIES

The aim of warning strategies is to warn the driver of the arising of a critical driving situation. In this case the warning must take place early enough to enable the driver to react and induce an accident avoiding driving manouver.

On the other hand such a warning must not be induced too early. Receiving a warning before normally reacting to a perceived critical situation is most inconvenient for the driver and even confusing and thus reduces the acceptance of the warning system [9].

The warning system must be effective independent of the driver´s capacities, e.g. a "sporty" driver should not be permanently warned even though he is still in control. Accordingly an unexperienced driver must not be warned before overexpanding his driving capacities to induce an accident avoiding driving manouver.

As the introduction of the ABS has shown the development of such warning systems has to account for psychological effects. Thus cars which have been equipped with ABS, contrary to the expectation, did not demonstrate reduced accident frequencies [10, 11], the reason being that the driver compensated such safety advantage with higher readiness for risks. In psychology this conduct is being explained by means of risk compensation theories (as well called Risikohomöostase). According to this "model idea" the amount of accidents of a person respectively of a country depend solely on which risks the individual is prepared to take respectively which amount of accidents a society is willing to tolerate.

This results in the fact that the individual respectively the society after improvement of the technical safety increases the readiness for risks to the extent that the danger potential is as high as it was before introduction of the measure [10,12].

Pfafferott [13] summarises the following experiences:

- Adaptation of the driver presupposes that he is aware of the effect of the measure resp. is informed about it.
- Adaptations to the safety measures are especially to be expected if the driver can acquire definite experiences on the effects of this measure.
- Unfavourable effects on safety are furthermore to be expected in cases where safety sensations of the driver are increased.
- The better improvements in performance and adventurous driving style can be put into action, the greater the chance that the usage is far away from the originally intended purposes.

Färber [9, 14] therefore developed a concept for the display of technical assistance systems. Thus the reliability of a warning should cover less than 100%, while its validity should amount to 100%. This means that the driver should not rely 100% upon the fact that he would be warned in a critical situation releasing him totally of his careful attention. It the system delivers a warning, he must be sure that he is faced with a critical situation.

As the evaluation of the "normal" driving behaviour already shows, the most important parameter as a reference quantity for triggering an assistance is the use of the given friction potential, respectively the friction potential reserve. Friction potential dynamically changes depending on the driver, the vehicle and the environment (e.g. route, road conditions, vehicles speed and traffic load).

For the consideration of these dynamic changes a software model was installed into the vehicle's computer-system containing basic parameters of the test vehicle that together with vehicle dynamics measured on-line are needed to predict friction potential [6].

Figure 12 schematically presents the generation process of different warning strategies.

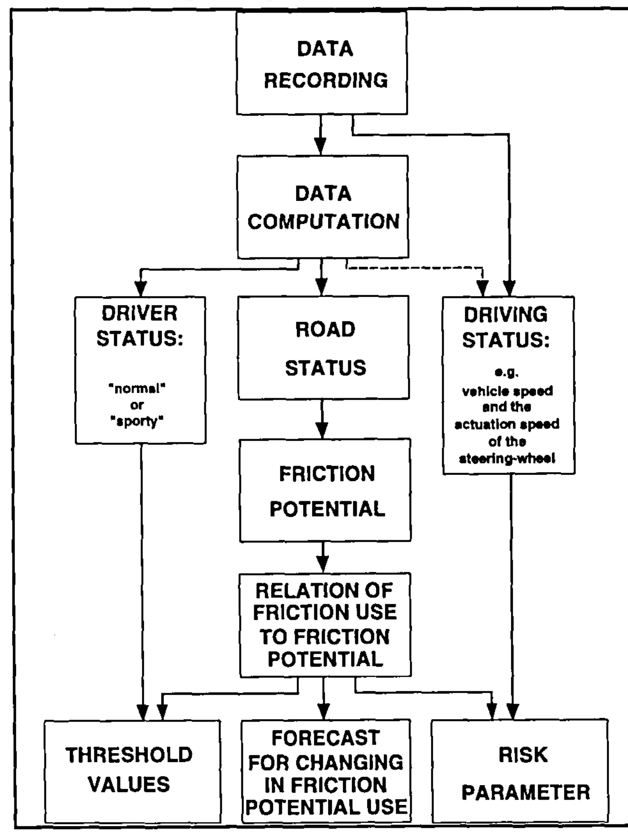

Figure 12. Warning strategies system

During a test drive the data measured on-line will be recorded and evaluated to quantify the actual friction potential (elliptical curve) and the friction potential reserve for the actual driving state.

Adaptation to Driving Status

From the actual road conditions the friction potential can be predicted [6, 8]. It is possible to differentiate between dry and wet road conditions and this will be evaluated in the software model in the vehicle´s computer system [6, 8].

With increasing speed the time span for changing the driving state decreases while the human reaction times remain constant so that warning signals must be triggered earlier by the evaluation of indicators such as for example the actuation speed of the steering-wheel.

For the detection of a prospective critical driving situation not only the actual friction potential use, but also ist further development must be considered and evaluated.

As an example Figure 13 shows two different time courses of friction potential use.

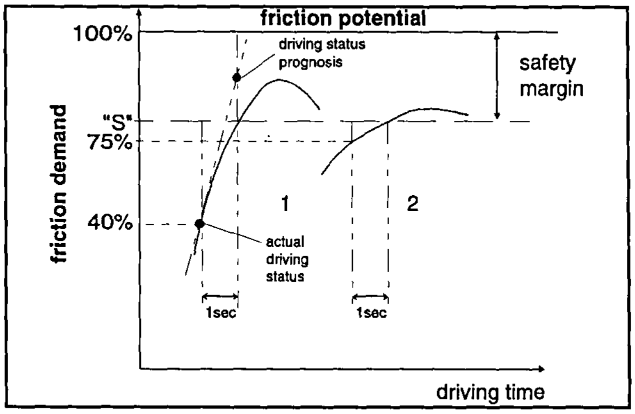

Figure 13. Forecast of the friction demand as function of the time gradient

The safety border "s" can be the absolute friction potential maximum but also an individual driving limit of the driver. The safety margin to the maximum of friction potential is given for the drivers correct action to prevent the accident. If for example the drivers reaction time is one second, the warning must be triggered one second before the safety border "s" will be crossed. So for the first curve the warning must be already triggered at 40% of the friction potential use. For the same safety margin and reaction time the warning in driving situation "2" must be triggered at 75% of the friction potential. For detecting the future time course of the friction demand the actual time gradient can be applicable as also shown in figure 13. When the friction potential use as prognosis for the coming second is higher than the safety border "s", a warning must be given for the driver.

Figure 14 shows the actual friction demand of a test drive over a section of the test route and the calculated friction demand prognosis.

Further evaluation of data indicated that the actuation speed of the steering wheel could be used as a parameter for the detection of a possible critical driving situation. The highest actuation speed value of the steering-wheel angle is already at the beginning of curves when friction demand and its prognosis are very low.

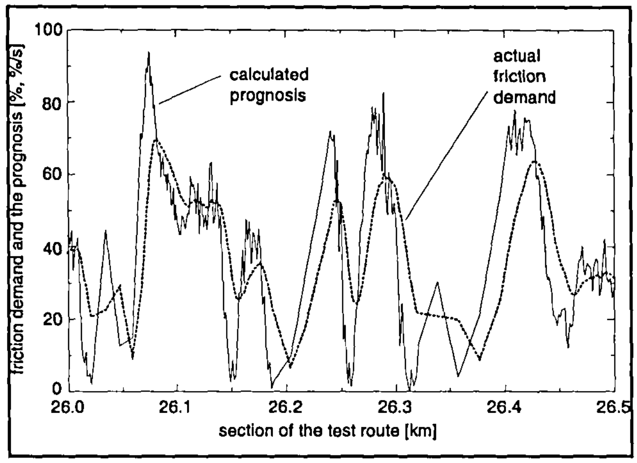

Figure 14. Actual friction demand and the calculated prognosis

In all curves, in which the maximum of the friction demand was higher than 85%, the actuation speed of the steering wheel at the beginning of the curve was uncommonly high. The time range between the maximum of friction demand and the highest value of actuation speed of the steering wheel was 1,5-2 seconds.

With the actuation speed of the steering wheel and two further safety relevant values such as driving speed and the acceleration potential use a **Risk Parameter (RP)** was constituted by experimental evaluation of data thus predicting too high friction demand in the vertex of a bend already at the entrance to it. Figure 15 depicts an example of the RP-value together with the value of the acceleration potential use.

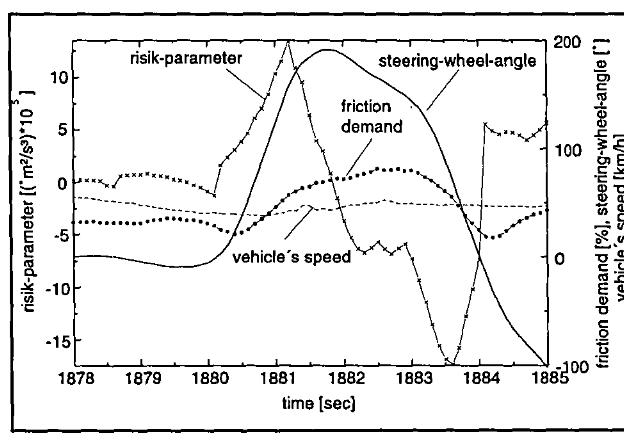

Figure 15. Early detection of high friction demand by the risk parameter (RP)

Adaptation to Driver Status/Driving Style

The boundaries of the working system are not only determined by the dynamic friction potential and vehicles speed but also by the human operator's characteristics of performance, skills and needs, which must be considered for the right driver dependent information-, resp. warning-level. It is possible that an unexperienced driver is driving beyond his individual skill limits while being far away from the boundaries of the subsystem vehicle-environment, the maximum of friction potential.
The examination of correlations between the driver's performance and driver strain serve to make the consideration of human performance limits possible.
It was however not possible to value driver´s strain online and use this as input for driving tests and warning strategies even though by offline evaluations certain correlations could be found. Nevertheless to realize a warning value in accordance with the driver an evaluation of the "driver-performance" was formed.
Further evaluations show that the driving input and its result, the measured Acceleration Potential Use APU, can be also indirectly derived as a parameter for driver performance. The very different use of the friction potential by "sporty" and "normal" drivers was already clear from early evaluations. These led to the conlusion that the time gradient of the APU is a better and faster parameter for the detection of different driver behaviour (and performance). The detection of driver performance is further improved by consideration of the driving speed (v) within the Driver Performance Parameter DPP.
In contrast to the detection of critical driving situations where the time gradient of LAPU is used, the detection of the driver performance is achieved by using the time gradient of the APU. In this case, the longitudinal acceleration potential can alšo be observed, which maybe used in situations where a high use of the braking acceleration serves as an early indicator for fast driving behaviour.
In Figure 16 the behaviour of two different drivers ("normal" and "sporty") is shown.
This diagram presents the mean and maximum values of the time gradient of the APU as function of driving speed. Both dAPU/dt are much higher over the whole speed range for fast driving behaviour than for normal driving behaviour.
For an exact functionality of this DPP during a test drive, filtering, averaging and specific weighting of the raw parameter is necessary to get a specific time-dependent function. With this specific weighting of the parameters, different time ranges could be realized for the rise time and fall time of the function.

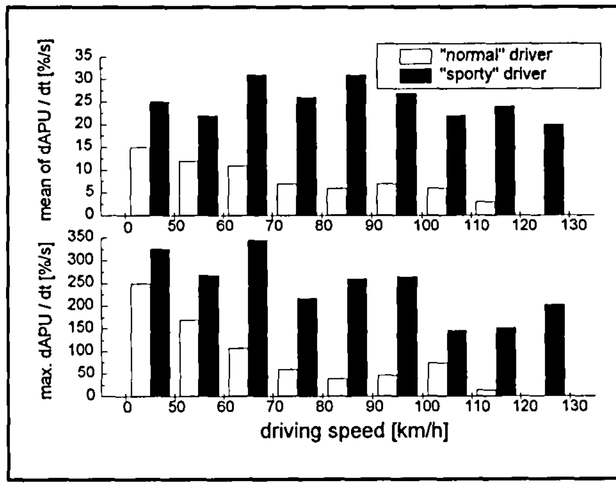

Figure 16. Different driver behaviour depending on the gradient of APU

Figure 17 shows two different DPP plots, one for fast driving behaviour and the second for normal driving behaviour.

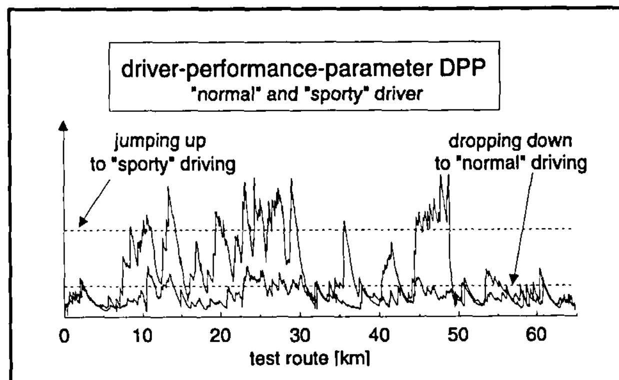

Figure 17. DPP for two different drivers

They were evaluated off-line and then weighted and averaged over the driving time. Additionaly in this figure the threshold for jumping up to information level no.3 and dropping down again to information level no.2 can be seen (see Figure 18 for information level). The thresholds for an up and down change in the information levels (warning thresholds) are determined as a result of off-line evaluation of the DPP from the test period data-pool with fixed warning thresholds. Those off line evaluations have shown that a dynamic adaptation of the warning thresholds (depending on the DPP) would reduce the total number of warnings by up to 60-70%. The dynamics of warning thresholds have no effect on system behaviour in normal driving situations.

The adaptation of warning levels to the driver is mainly based on lateral friction potential of the driver-vehicle system. Three information-levels were realized as shown in Figure 18.

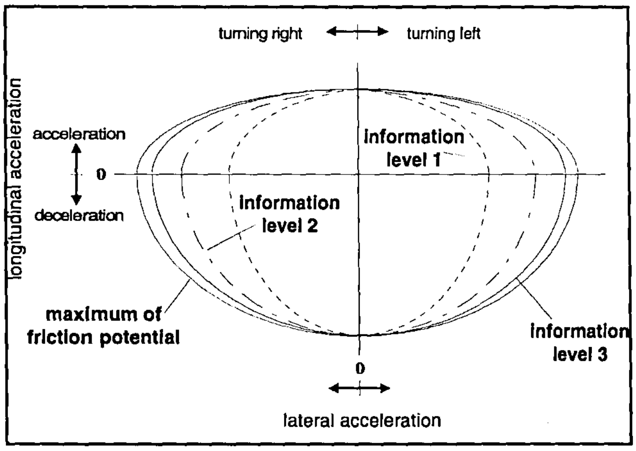

Figure 18. Different information levels as function of friction potential and lateral vehicle dynamics

Purely longitudinal acceleration and deceleration will not be restricted so that the maximum of a longitudinal acceleration will in fact be used by the driver when necessary. In a front wheel driven vehicle the driver can respond to a critical situation during the acceleration of the vehicle, when the driving wheels lose adhesion to the road by simply removing his foot from the accelerator pedal and thus stabilize driving conditions. For a rear wheel driven vehicle such a reaction can make things worse as the vehicle normally tends to swerve. It is obvious that the warning strategy for a front wheel drive vehicle can not be simply used for a rear wheel vehicle without restrictions.
The first threshold level assistance stage remains fixed and only the second level can be moved to the third level depending e.g. on the driver performance (DPP).

DEVELOPMENT OF DRIVER-VEHICLE INTERFACES

The human information processing and control mechanisms used during car driving were considered in the search for the best way to inform the driver [2,15,16]. The visual, auditory and haptic channels have been considered. A flashing red lamp with a general cautionary symbol is installed next to a permanently illuminated (when activated) yellow lamp with a symbol of a skidding vehicle in the dashboard.
Auditory alerts are recorded (sounds like e.g. squealing tyres) and generated by a noise generator and reproduced by a buffered RAM-based recorder via one speaker of the vehicle´s radio system.
All signals can be triggered by the data acquisition unit. Time and modality of the signals presented are stored in the data acquisition unit.

Tactile signals were transmitted as vibrations and as variable characteristics of actuation forces at the basic actuators for car driving:

- acceleration-pedal used mainly to control longitudinal vehicle dynamics,
- steering-wheel used mainly to control lateral vehicle dynamics.

Vibrations were generated by a reversible-stroke-magnet which is very light and small so that it could be mounted without major modifications. A further advantage of this actuator is its low energy consumption and the automotive compatible electrical power supply requirements (12V).
The excitation frequency could be varied within a range of 1 to 30 Hz.
In the first phase of the tests with warning strategies, warnings were manually triggered. The excitation frequency with the best result in driver's perception was examined in the same phase of these tests too.
Examination of relevant literature proved that the best human perception for hands and feet vibrations is in a frequency-range from 8 to 16 Hz (Figure 19).

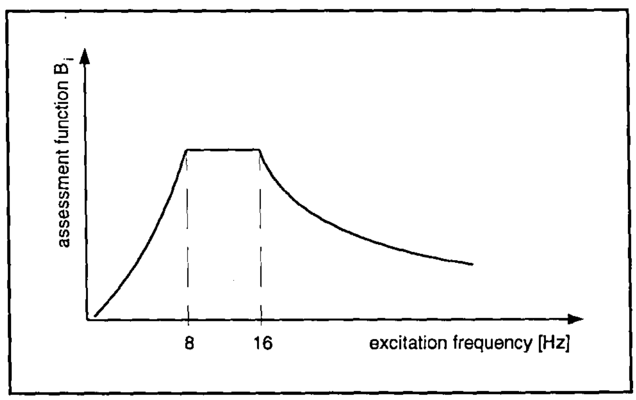

Figure 19. Range of excitation frequency for a goo result in human perception [17]

On the steering-wheel the vibrations are generated in the direction of steering-wheel actuation in order not to influence the vehicle steering-system. The reversible-stroke-magnet is mounted on the bottom of the steering-wheel-rim and transmits the vibrations through the steering-wheel cover to the driver's hands (Figure 20).

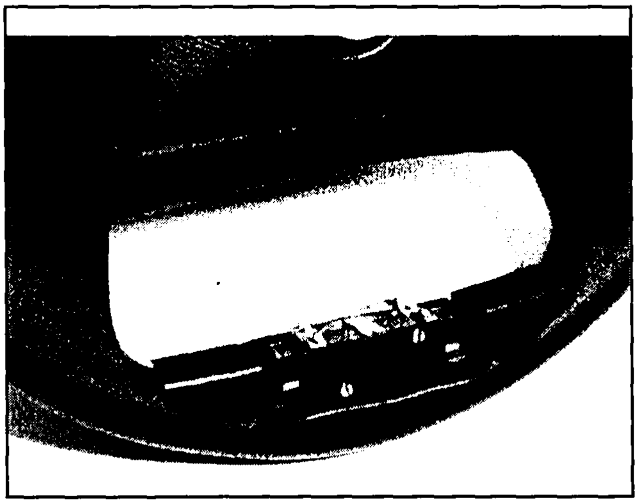

Figure 20. Configuration on steering wheel

In order to avoid excitations of the throttle valve, vibrations at the acceleration-pedal are generated laterally to the actuation direction of the pedal. According to our own and other examinations [17] the effect is similar to vibrations parallel to the actuation direction.
The basic design of the original acceleration-pedal is not changed for the reversible-stroke-magnet which is mounted on top of the pedal. A pivoted body is fixed on top of both and it transmits the vibrations to the driver's foot (Figure 21).

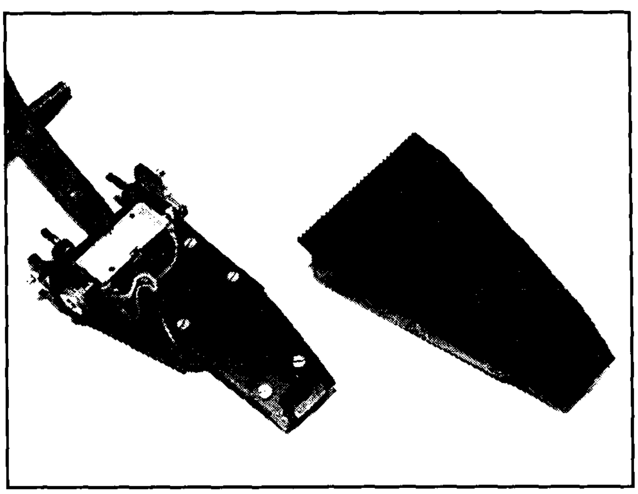

Figure 21. Configuration on driving pedal

Variation of actuation forces is a another possibility to transmit information to the driver at the interfaces steering-wheel and acceleration pedal. At the acceleration-pedal actuation forces are raised by a step-motor effecting the return force of the acceleration-pedal in such a way that the variation is clearly detectable by the driver but does not necessarily mean a system take over Figure 22.

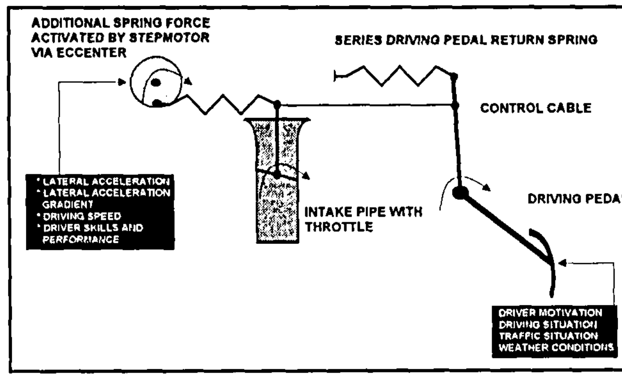

Figure 22. Variation of the actuation forces of the driving pedal system

At the steering-wheel actuation forces could be influenced by a modification of the standard steering-system servo (Figure 23).

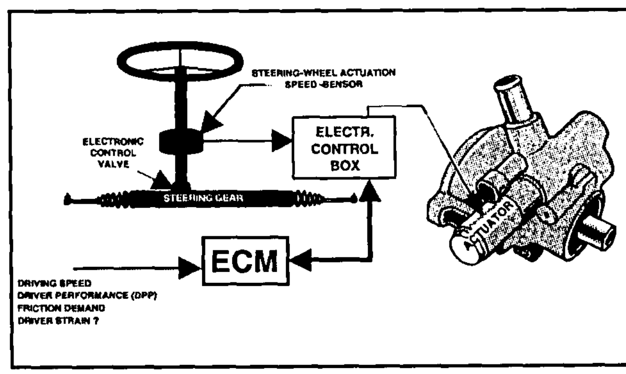

Figure 23. Variation of the actuation force effort o the steering-system servo

As this effects the safety of the steering-system servo this possibility was not yet realized in this stage of the project.

FINAL DEFINITION OF THE IN-VEHICLE DRIVER SUPPORT SYSTEM

The alerting signals were tested in field-studies: in a first phase the support devices were checked for their reliability and safety, in a second phase they were evaluated in driving tests on the test route with subjects who had taken part in the first phase of driver behaviour studies.
The following evaluation criteria were utilized:
- frequency and conditions of information request
- reaction time to cautionary and warning signals
- changes in driver behaviour due to warning strategies

Table 2.
Man-machine-interfaces used for the test phase with warning strategies, classified by the support level

support level	receptor system	warning limit	modality
caution level 1	visual, auditory	LAPU > 50%	symbols (lamps) + tone
warning level 2	visual, auditory, haptic	LAPU prediction > 70%	symbols (lamps) + tone intermittant + increased actuation force at the accelerator pedal
warning level 2	visual, haptic	RP > 12000	symbols (lamps) + vibrations at the steering wheel
warning level 3	visual, auditory, haptic	LAPU > 80% only"sporty" drivers	symbols (lamps) + tone intermittant + increased actuation force at the accelerator pedal
warning level 3	visual, haptic	RP > 14000	symbols (lamps) + vibrations at the steering wheel

(objective measures: e.g. use of friction potential, number of critical situations, subjective measures: expert rating by accompanying test manager)

- strain reactions to caution and warning signals (physiological and subjective measures)
- acceptance of assistance (subjective measures: questionnaires).

These tests showed that the best way to transmit an information to the driver was the active accelerator pedal. In order to prevent a critical driving situation, the increased accelerator return force was the correct method of warning the driver which induced the right reaction at the moment, namely the closing of the throttle.
The most detectable signals were the combinations of haptic and acoustic signals, which had the best support for the driver.
For driver groups of different experience, different information resp. warning levels, which were based on the individual use of lateral acceleration potential (figure 18), were defined and will be used for future driving tests with warning strategies.
The final definition of the assistance system based on the results from the preliminary tests is shown in Table 2
The warnings consisted always of a combination of visual, acoustic and haptic signals, which were given to the driver at the basic actuators for car driving. A warning resulting from the Risk Parameter (RP) was given at the steering wheel by making it vibrate, and a warning resulting from the LAPU prediction was given at the accelerator pedal by increasing the actuation force.

Whenever "sporty" driving behaviour was detected (DPP), the caution level 1 and the warning level 2 were disvegarded, only the level 3 was active. The warnings resulting from RP and LAPU prediction were given to the driver independently from each other as well as in combination.

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USABILITY OF IN-CAR EMERGENCY WARNINGS ACCORDING TO AGE AND CAPABILITIES OF DRIVERS

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ABSTRACT

The response times to collision warning signals of 114 French drivers have been collected, using the INRETS driving simulator. The age of participants ranges from 18 to 80 years. Different scenarios were designed on the driving simulator in order to compare the response times of drivers according to the attentional sharing during driving. We observed that the following variables have an effect on response times while driving : age, gender, level of attention sharing (i.e. task complexity), familiarity with situation (learning).

INTRODUCTION

During the design of advanced longitudinal car controls, two levels of driver aids are considered : informative and control ones. In the informative mode, perceptual signals (visual, auditory, haptic) are given to drivers for alerting them they are approaching a dangerous headway ; in the control mode, the aiding systems automatically slow down or stop the car in case of risk of imminent collision. Both modes are studied in several collision avoidance or intelligent cruise control systems of the European Programmes such as Prometheus and Drive. The switch from one mode to the other is computed according to time-to-collision or risk level criteria. In the informative mode, the driver's reaction times must be considered in order to deliver the information according to an appropriate schedule.

Many studies have already been conducted in order to evaluate the response time of drivers in risky situations (reviewed in Zakosek, 1994 and Mcgehee and al. 1994). These studies were not quite fitted with the objective of designing in-car auditory warnings, because they must consider the ambiguity of real road situations. With the implementation of an intelligent anti-collision system, it is assumed that this ambiguity will be clarified . Consequently, the thinking time before avoiding an obstacle should be drastically shortened.

In the whole process of driver Perception / Decision / Reaction time, the residual response-time of drivers will mainly lie in the perception and motor phases of the response time. Both are typically age-related (Sivak (1995). If Ling (1991) conducted an important experiment in order to evaluate response times according to age and capabilites, he did not consider that drivers are normally in divided attention situations. The aim of the present survey is to collect response times taking in account the age factor and the driving task complexity.

DRIVERS SAMPLE

The drivers involved in the experiments were selected taking in account age (from 20 to 80) and gender (same number of men and women). All the subjects were experienced drivers. All have been driving during the preceding 12 months.

At the beginning, we planned to recruit 20 drivers in six age classes. But, it was not possible to find 20 drivers in the age class from 70 to 80. Only 10 persons (5 male and 5 female) were recruited in this class. This fact was taken in consideration in the statistical analysis.

Among the 144 French drivers involved in the studies, 20 were involved in the setting up of the methodology (Zakosek, 1994). Regarding the other ones, only data for 114 drivers were usable.

METHODOLOGY

Environment

The observations were conducted while driving on the LESCO driving simulator (Durand & Letisserand, 1994). This simulator is a static base one, images are generated by a Silicon Graphics Reality engine, the projector is a Barco and a realistic noise is generated in relation with the virtual car speed. The mock up is a Renault 21 car with efficient commands ; accelerator and brake pedals are equipped with sensors in order to record the pedal status.

Tasks descriptions.

<u>**Itinerary**</u> - Each trial was made on a highway of 8 kilometres.

Auditory signal - In order to evaluate the response time to collision warning, the car was equipped with a so-call "display demonstrator" (Miguel P. de& al. (1993), which delivers auditory warning signals. The sound was a mix of two broad band frequencies; one centred around 1200 Hz, and the other one centred around 800 Hz.; the overall acoustical energy was 80 LAeq (namely dBA in Leq). The average signal-to noise ratio was 10 dBA above in-car noise. The acoustical definition was made according to auditory capabilities in ageing (HARDIE, 1996). Audibility by every subject was checked before the trials.

The signals announced various risk collisions such as stopped or crossing cars, but they were delivered before the obstacles were visible by the drivers. In fact, the conflicting road users were masked either by curves, or any visual element of the road environment

Response task - Drivers were asked to brake as fast as possible every time they hear the warning signal.

The response time was decomposed according to the rules of Ling (1991)

primary reaction : time between the beginning of the auditory signal and the beginning of action off the accelerator pedal.

secondary reaction : time between the beginning of action off the accelerator pedal and the beginning of action on the brake pedal.

total response time : primary response time plus secondary response time.

Attention sharing and task complexity - To modulate the complexity of attentional sharing, three kinds of trials were designed with one, two or three associated tasks :

One task : the car was stopped, and drivers had to brake as fast as possible at each signal. This trial was similar to Ling's experiments.

Two tasks : drivers had to drive the car on the itinerary, and brake at each signal.

Three tasks : drivers had to drive the car on the same itinerary, to brake at each signal (randomly delivered) and they had to perform a conversation with an observer. The auditory signals happened during the conversation, at an unexpected time for drivers.

Learning effect - The learning effect was studied thanks to the repetition of trials.

RESULTS

Effects of age and other variables on sensory -cognitive part of response time.

The influence of ageing during the sensory cognitive part of the process is evaluated during the primary reaction time, which involves time necessary for hearing the signal, for recognising it, for making the response decision.

Table 1 gives the results of statistical variance analysis regarding the primary reaction. A part of these results is consistent with previous studies which pointed out that gender, age, training (coded ("trial") have significant effect on primary reaction time.

The expected action of task complexity on primary reaction speed is not observed, neither by itself nor by interaction with age. It is only observed in the learning process : more complex are the associated tasks, less fast the reaction at a new signal is.

Table 1.
Variance analysis of primary reactions
(432 cases processed)

sources of variation	sums square	DF	Mean square	F	Signif .*
gender	0.144	1	0.144	2.9	SIG
age	1.158	5	0.232	4.793	SIG.
task	0.014	2	0.007	0.142	n.s.
trials	0.539	1	0.539	11.16 2	SIG.
gender*age	0.607	5	0.121	2.511	n.s.
gender*task	0.018	2	0.009	0.182	n.s.
gender*trial	0.044	1	0.044	0.903	n.s.
age*task	0.219	10	0.022	0.453	n.s.
age*trial	0.332	5	0.066	1.372	n.s.
trial*task	0.436	2	0.218	4.513	SIG.

SIG : significant , n.s. : non significant

Effects of age and other variables on motor part of response time

The secondary reaction is the time to move the foot from one pedal to the other one.

Table 2 shows the statistical analysis of sources of variation on the speed of secondary reaction. The significance of effect of age and gender are consistent with previous results which demonstrated that women

slower motor reaction than men, and that ageing is slowing every aspect of motor behaviour.
The unexpected fact is that the task complexity has also a significant effect of slowing motor reaction, by itself and in interaction with age.

Table 2
Variance analysis of secondary reactions
(432 cases processed)

sources of variation	sums square	DF	Mean square	F	Signif .*
gender	0.469	1	0.469	21.913	SIG.
age	0.325	5	0.065	3.037	SIG.
task	0.125	2	0.062	2.910	SIG.
trial	0.040	1	0.040	1.866	n.s.
gender*age	0.101	5	0.020	0.948	n.s.
gender*trial	0.052	1	0.052	2.451	n.s.
gender*task	0.017	2	0.008	0.391	n.s.
age*trial	0.069	5	0.014	0.644	n.s.
age*task	0.369	10	0.037	1.725	SIG.
trial*task	0.007	2	0.004	0.167	n.s.

SIG : significant n.s. : non significant

A precise observation of data shows that the more aged drivers (class 70-80) have better average secondary reaction, and less variability than the preceding class (class 60-70). That is a supplementary argument consistent with surveys which demonstrated that the eldest drivers are a selection of the most heathy people in their class.of age. (STAMMI 1989).

Effect of age on total response time

In total, if we consider the whole response time, we observed that the following variables have an effect on speed on response time while driving :

age,
gender,
level of attention sharing (i.e. task complexity),
familiarity with situation (learning).

Figure 1 shows the average total response time according to age and attention sharing, for the second trial. The slowing effect of attention sharing is clearly demonstrated by the three curves ranking from the bottom (faster reaction when drivers have to cope with a single task) to the top (slower reaction when drivers have to cope with three tasks at the same time).

The effect of age is clearly slowing reactions when drivers have to direct their attention on one task or two tasks. For more complex situations (attention sharing between 3 tasks) the variability is more important for intermediate age classes than for the eldest, and the average is about the same for the age classes from forty to eighty.

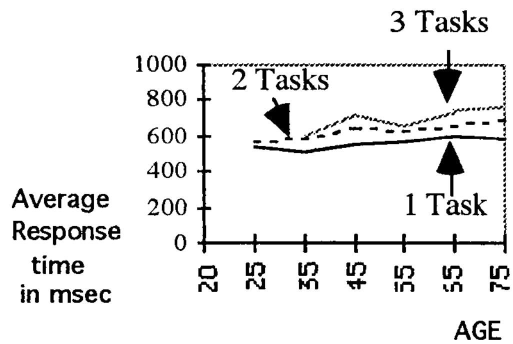

Figure 1. Effect of age and task complexity on total response time.- Second trial.

DISCUSSION

This study was conducted in order to give some information about drivers' response time range to auditory warning. A population of 114 French drivers with an age distribution from 20 years to 80 years was selected. They were observed in a driving simulator while using a collision warning signal. The response time before the beginning of braking was clearly influenced by the complexity of associated tasks and by age. Nevertheless, the eldest drivers (70-80) have the same average response time than younger ones, and less variability. The variability of response time begins to increase from the age of 40, and reaches a maximum between 60 and 70. The slowing of reaction with age is more important during sensory-cognitive part than the motor part of the response time. We observed that for 114 drivers from 20 to 80 years old, performing simple, double or triple tasks, the distribution of data is as following :

86 % of data are under 860 msec,

99 % of data are under 1360 msec.

These data are slightly shorter than those reported by Olson and Sivak (1986) who estimate the 95th percentile drivers at 1.6 seconds ; this can be explained by the difference in situations : we observed the reaction at a warning signal, and not at an external driving situation, so, there is no need of judgement situation before launching the response behaviour. That demonstrates the interest of a non ambiguous signal to speed the driver reactions. Our data meet the proposal to include two time delays in the scheduling of collision warning:

a short delay of 1 sec
a conservative delay of 1.5 sec.

ACKNOWLEDGEMENTS

This study was supported by the French Ministry of Transport - DRAST

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APPENDIX

Table 3
Learning effect : comparison of first and second trial on the total response times according to age (Task complexity : one task)

age	First trial		Second trial	
	mean	st. dev.	mean	st. dev.
20-30	0.55	0.08	0.53	0.10
30-40	0.62	0.14	0.51	0.08
40-50	0.62	0.09	0.55	0.17
50-60	0.90	0.32	0.56	0.12
60-70	0.86	0.62	0.6	0.11
70-90	0.81	0.10	0.58	0.02

Table 4
Sensory-cognitive and motor reaction times according to age (all task complexity levels and trials aggregated)

age	Primary reaction time *Sensory-cognitive*		Secondary reaction time *Motor*	
	mean	st. dev.	mean	st. dev.
20-30	0.31	0.14	0.30	0.12
30-40	0.32	0.12	0.28	0.09
40-50	0.37	0.20	0.28	0.18
50-60	0.40	0.20	0.30	0.12
60-70	0.43	0.36	0.35	0.19
70-90	0.46	0.18	0.27	0.13

Table 5
Effect of age and task complexity on total response time.- Second trial.

age	1 Task		2 Tasks		3 Tasks	
	mean	st.dev	mean	st. dev	mean	st. dev
20-30	0.53	0.10	0.56	0.11		
30-40	0.51	0.08	0.58	0.10	0.59	0.06
40-50	0.55	0.17	0.64	0.30	0.71	0.28
50-60	0.56	0.12	0.62	0.14	0.66	0.16
60-70	0.60	0.11	0.66	0.15	0.76	0.21
70-90	0.58	0.02	0.68	0.12	0.75	0.17

STATUS UPDATE OF NHTSA'S ITS COLLISION AVOIDANCE RESEARCH PROGRAM

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Paper Number 96-S2-O-09

ABSTRACT

This paper provides a status update on the National Highway Traffic Safety Administration's (NHTSA) program to facilitate development and early deployment of cost-effective, user-friendly collision avoidance systems. The program includes an expanding crash avoidance knowledge base, development of a vital set of research tools, identification of crash avoidance opportunities, examination of key human factors and system design issues, and development of performance specifications for crash avoidance products and systems. The status of each element of the program is discussed.

INTRODUCTION

Since 1991 the NHTSA has had a concentrated program to facilitate the development and deployment of effective safety-related systems as part of the Department of Transportation Intelligent Vehicle Highway Systems (IVHS, recently renamed the Intelligent Transportation Systems, or ITS) program. The details of the program and progress have been reported at the 13th and the 14th Enhancement of Safety of Vehicles (ESV) Conferences [Leasure 1991 and Leasure 1994].[1] At the 14th ESV conference, a second paper was presented which discussed the need for, and approaches to, the development of methodologies for estimating safety benefits [Burgett 1994]. The paper is divided into three sections. The first discusses the status of specific projects in NHTSA, the second discuses safety-benefits activities, and the third section discusses new initiatives within the program. The paper concludes with a summary of what the future of the program looks like.

[1]Notation in brackets denote entries in the List of References.

STATUS OF ONGOING WORK

In Leasure 1994, the list of projects and activities was divided into the five thrusts of the program. These thrusts were identified and explained in more detail in the NHTSA ITS Strategic Plan [NHTSA 1992]. This part of the present paper is organized in the same way and the list of projects is the same, to simplify comparisons.

Research Tools, Assessment Methodologies, And Other Considerations

This section updates the status for several research tools and basic research.

<u>National Advanced Driver Simulator (NADS)</u> - Driving simulators are an essential tool for developing an improved understanding of driver behavior since they provide a means for carrying out controlled experiments in crash imminent situations without putting subjects at risk. While many levels of simulation sophistication are possible, NHTSA has focused on the development of a high-fidelity, moving base simulator. This state-of-the-art simulator will replicate, with impressive fidelity, a wide range of highway driving scenarios.

Since the 14th ESV, the NHTSA has moved ahead toward construction of the NADS. To accomplish this, it has awarded a contract to the University of Iowa for design construction of the NADS building. It has also awarded a contract to TRW's Integrated Engineering Division - Transportation Systems, Sunnyvale, California for design and construction of the NADS. It is expected that operation will begin in the spring of 1999.

Portable Human Factors Data Acquisition System for Crash Avoidance Research (DASCAR) - Research in crash avoidance requires a body of knowledge about how drivers drive, how they avoid collisions and how they respond to crash avoidance countermeasures and other in-vehicle advanced technologies. To address the need for such data, NHTSA is developing a family of state-of-the-art, easy to install, portable data acquisition systems. These systems will be unobtrusive to drivers and inconspicuous to other drivers and hence will support in-situ "naturalistic" studies of driver behavior and performance. A prototype system has been completed and validated, and additional systems are now being constructed. The prototype system is currently being used to support a variety of human factors research.

System for Assessing the Vehicle Motion Environment (SAVME) - This project is developing and validating a measurement system that can quantify the motions of vehicles as they move in traffic. The SAVME system will establish the locations of all vehicles within the field of view relative to roadway boundaries and other features. The motions will be described by location, relative velocities, and angles. In operation, the SAVME will gather information on successful collision avoidance maneuvers, such as reaction to other drivers cutting in front, normal following distance, typical lane change trajectories, and response to inclement weather. This information will provide a database which can be used to design ITS countermeasures that intervene and/or provide collision avoidance warning to drivers.

The initial sensor concept for the SAVME was a ranging laser which could be scanned in two directions. Unfortunately, it has not been possible to obtain equipment that can provide this capability. Thus, the project has been redirected to base the sensing on a charge-coupled camera that directly reduces the image to a digital format. This redirection has extended the time for completion of a prototype. A fully functioning prototype should be available by mid-1997.

Variable Dynamics Test Vehicle (VDTV) - The VDTV is a research vehicle with computer control of throttle, brake, and steering as well as changeable handling and stability characteristics. This research tool will be used to establish performance boundaries for systems that directly control vehicle motion, i.e., determine the vehicle-related limitation that should be placed on collision avoidance algorithms. It will also allow determination of how drivers react to various proposed ITS crash avoidance concepts, including the effect of vehicle characteristics on device effectiveness. The VDTV will also be used to validate NADS control algorithms and as a research vehicle for the safety evaluation of automated highway system (AHS) concepts.

An agreement has been signed with the Jet Propulsion Laboratory of the California Institute of Technology for design and construction. It is expected that the VDTV will be available for research in mid 1998.

Driver Workload Assessment - This project has developed a methodology for evaluating workload imposed upon drivers using in-vehicle technologies, including crash avoidance systems, route guidance/navigation systems, and other advanced technologies. The unique feature of the methodology is its use of demonstrated safety relevant measures. The methodology will be used to develop standardized workload evaluation protocols involving comparisons between baseline data and that obtained under conditions associated with use of high technology systems either individually or in concert. The assessments will be used to identify systems and system design and implementation attributes that may compromise safety. Future applications will apply the protocols to support development of human factors guidelines for the driver-vehicle interfaces for these systems.

Crash Avoidance and the Older Driver - One segment of the driving population that may benefit from ITS is older drivers. NHTSA has completed a study to examine their vehicle crash experience, assess their capabilities and limitations that may influence these crashes, and identify vehicle crash avoidance technologies that may aide them [Harowski 1995]. The study found that while they are under-represented in crashes and fatalities relative to their numbers in the U.S. population, their per-mile crash involvement and fatality rates are higher than other driver groups. For example, the analysis revealed that crash types with increased involvement of older drivers included backing and lane change/merge crashes. These findings suggest where ITS crash avoidance technologies may be particularly helpful to older drivers. However, it is also recognized that this technology can be a "double-edged sword" if the systems that are designed to assist them also increase their workload and are confusing to understand and operate. As NHTSA research continues, consideration of the needs and capabilities of older drivers will be included.

In-Vehicle Crash Avoidance Warning Systems - Human Factors Considerations - This project has developed guidelines for the interface requirements for collision warning systems [NHTSA 1996]. The research is addressing the following human factors questions:

How do false alarm rates affect driver annoyance? What are the characteristics of an effective auditory warning? What type of information should be presented to the driver for back-up alarms? When should it be presented to provide the driver with the optimum amount of time to take action?

Evaluation of Potential Health Hazards from Wide-Spread Usage of Collision Avoidance Systems - Widespread introduction of collision avoidance systems which utilize active sensors such as radar or laser could result in an increase in the level of electromagnetic radiation in the highway environment. Two studies have been completed to address potential health or safety hazards which could arise from the use of active sensors. One reviewed the physiological basis for current standards which address health effects of non-ionizing electromagnetic radiation. The second study developed a computer model which can be used to estimate field strength in the vicinity of one or more radiation sources. The Federal Communications Commission for vehicular radar has also approved the following power-density levels for vehicular radars [FCC 1995]:

> The radiated emission limits within the bands 46.7 - 46.9 GHz and 76.0 - 77.0 GHz are as follows: (1) If the vehicle is not in motion, the power density of any emission within the bands specified in this section shall not exceed 200nW/cm^2 at a distance of 3 meters from the exterior surface of the radiating structure. (2) For forward-looking vehicle-mounted field disturbance sensors, if the vehicle is in motion, the power density of any emission within the bands specified in this section shall not exceed 60 μW/cm^2 at a distance of 3 meters from the exterior surface of the radiating structure. (3) For side-looking or rear-looking vehicle-mounted field disturbance sensors, if the vehicle is in motion, the power density of any emission within the bands specified in this section shall not exceed 30 μW/cm^2 at a distance of 3 meters from the exterior surface of the radiating structure.

Part of the basis for these levels is the need to keep power-density exposure for highway users at an acceptably low level of 10 mW/cm^2.

Vehicle-Induced Feedback Cues and Their Relationship to Driver Performance and Safety - NHTSA has recently completed a project to study the importance to drivers of cues and feedback from within the vehicle, such as kinesthetic, vestibular, and cues associated with certain vehicle response characteristics, e.g., body roll, tire screech, apparent oversteer/understeer. The purpose of the project is to develop a better understanding of the role of these vehicle cues relative to driving performance.

Identify Promising Crash Avoidance Opportunities

Figure 1 shows the distribution of crash types that provide the maximum opportunity for significant safety improvement through the introduction of safety-effective collision avoidance products/systems. From this figure, it can be seen that single vehicle road departure, rear end, and crossing path (intersection) crashes comprise nearly three-fourths of all crashes. The remaining one-fourth is comprised of blind-spot, head-on and other crash types. Contributing factors such as reduced visibility, e.g., at night or in degraded weather conditions, and driver drowsiness, occur across the spectrum of crash types shown in this figure. A summary of the causal factors for these various types of collision is shown in Table 1 [Najm, et al 1994].

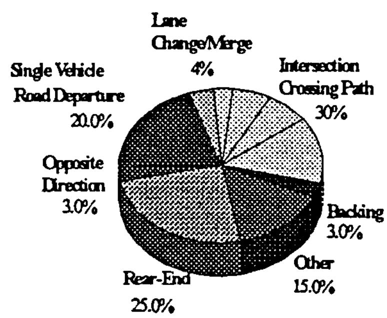

Figure 1. Distribution of Crash Types

Table 1. Causal Factor Distribution of Target Crashes

Crash Type	Driving Task Errors			Driver Physiological State			Vehicle Defects	Road Surface	Atmosp. Visib.	Total
	Rec. Er.	Dec. Er.	Err. Ac.	Drunk	Asleep	ILL				
RE	56.7	26.9	1.1	2.1	0.0	9.6	1.2	2.3	0.1	100.0
BK	60.8	26.6	2.0	3.0	1.9	0.0	5.7	0.0	0.0	100.0
LCM	65.0	32.1	2.6	0.0	0.0	0.0	0.3	0.0	0.0	100.0
SVRD	15.5	17.8	15.9	10.1	11.8	3.5	5.3	20.2	0.0	100.1^
OD	17.8	7.0	19.6	31.7	0.0	1.1	4.5	18.3	0.0	100.0
SI/SCP	40.7	16.2	29.1	12.6	0.0	0.0	1.6	0.0	0.0	100.2^
UI/SCP	73.6	12.2	3.4	2.7	0.0	0.0	0.0	7.0	1.1	100.0
LTAP	49.0	41.2	9.1	0.4	0.0	0.0	0.0	0.0	0.1	99.8^
%*	43.6	23.3	8.5	6.0	3.5	4.5	2.5	8.0	0.1	100.0

* Percentage of all target crashes (71% of 1993 GES) ^ Rounding error

These results are from an extensive study of NHTSA accident files by staff and support contractors at the Volpe National Transportation Systems Center (VNTSC) (see Burgett 1995 for a listing of the specific report titles). They provided the initial basis for the performance specification work that is described later in this paper. In addition to the work at VNTSC, each contractor that is working on collision avoidance performance specifications did additional analysis of accident data fields and specific cases. This work was done to develop a clearer description of the dynamics of the events that preceded specific types of crashes. Table 2 is an example of the type of engineering insight that was obtained from these additional studies. A different perspective on pre-crash dynamics is shown in Figure 2. [Campbell 1995]. These data, which show the distribution of rear-end collision conditions on interstate highways in the state of North Carolina, reflect the same three predominant types of crashes that are shown in Table 2.

Another example of engineering insight is shown in Figure 3. This figure shows that there is a great variety of conditions that precede a collision, in this case road departure collisions, but that they can be organized into a form that provides insight into potential countermeasures.

Demonstrate Proof of Concepts for Crash Avoidance and Mitigation

A key responsibility of NHTSA is to demonstrate that advanced technology can practicably enhance the crash avoidance performance of motor vehicles. NHTSA's program includes the development of performance guidelines for crash avoidance technologies and testing of hardware systems.

Table 2. Rear-End Collisions Dynamic Situations Matrix (Numbers in table are % of Total)

Lead Vehicle	Following Vehicle			SUM
	Accelerating	Constant Velocity	Decelerating	
Stationary	0.54	23.72	0.69	24.95
Constant Velocity	0.74	2.80	0	3.54
Decelerating	0	14.71	0	14.71
Accelerating	0	2.07	0	2.07
Decel & Stationary	0.11	50.05	4.57	54.73
Sum	1.39	93.35	5.26	100

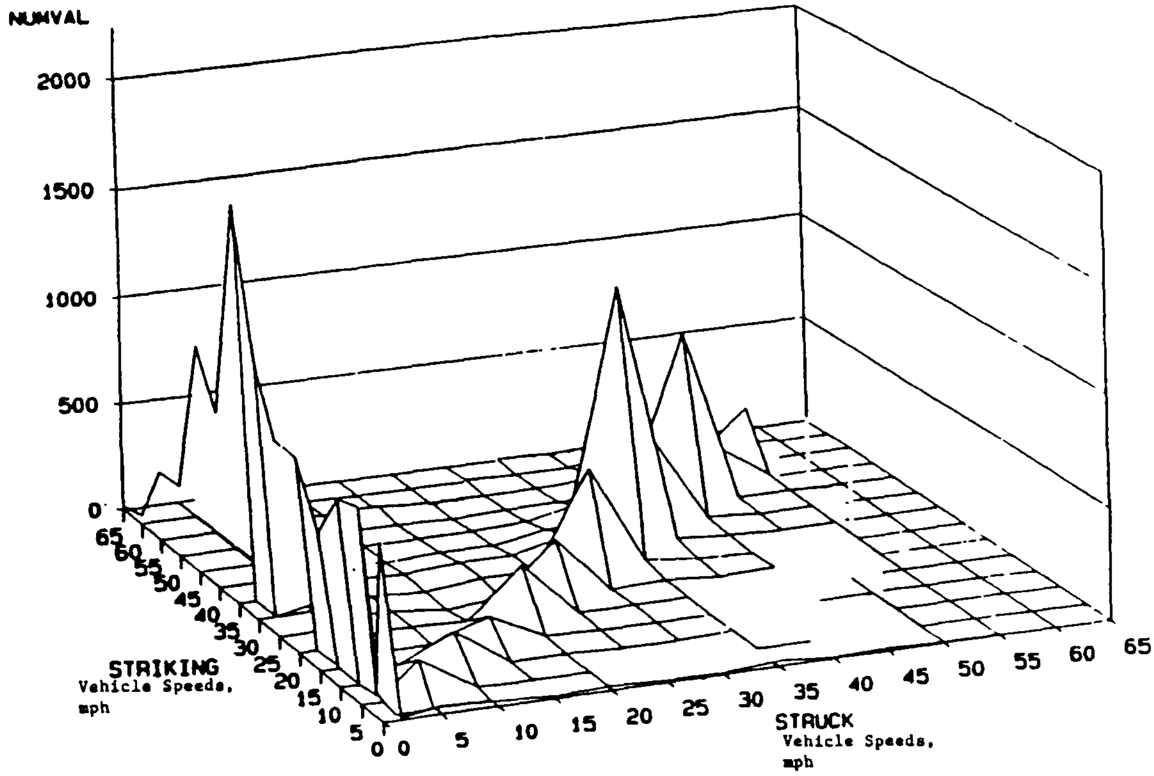

Figure 2A- Pre-event travelling speeds for rear-end collision in which the lead vehicle was stopped at the time of impact

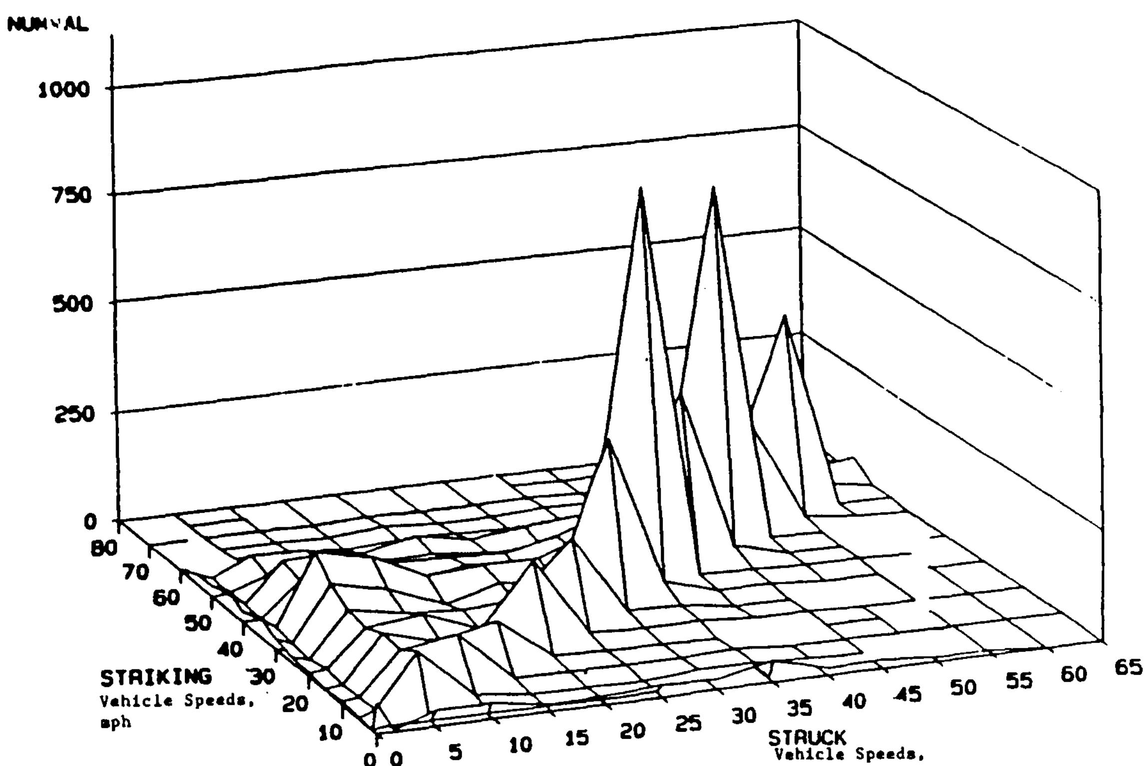

Figure 2B- Pre-event travelling speeds for rear-end collision in which the lead vehicle was moving at the time of impact

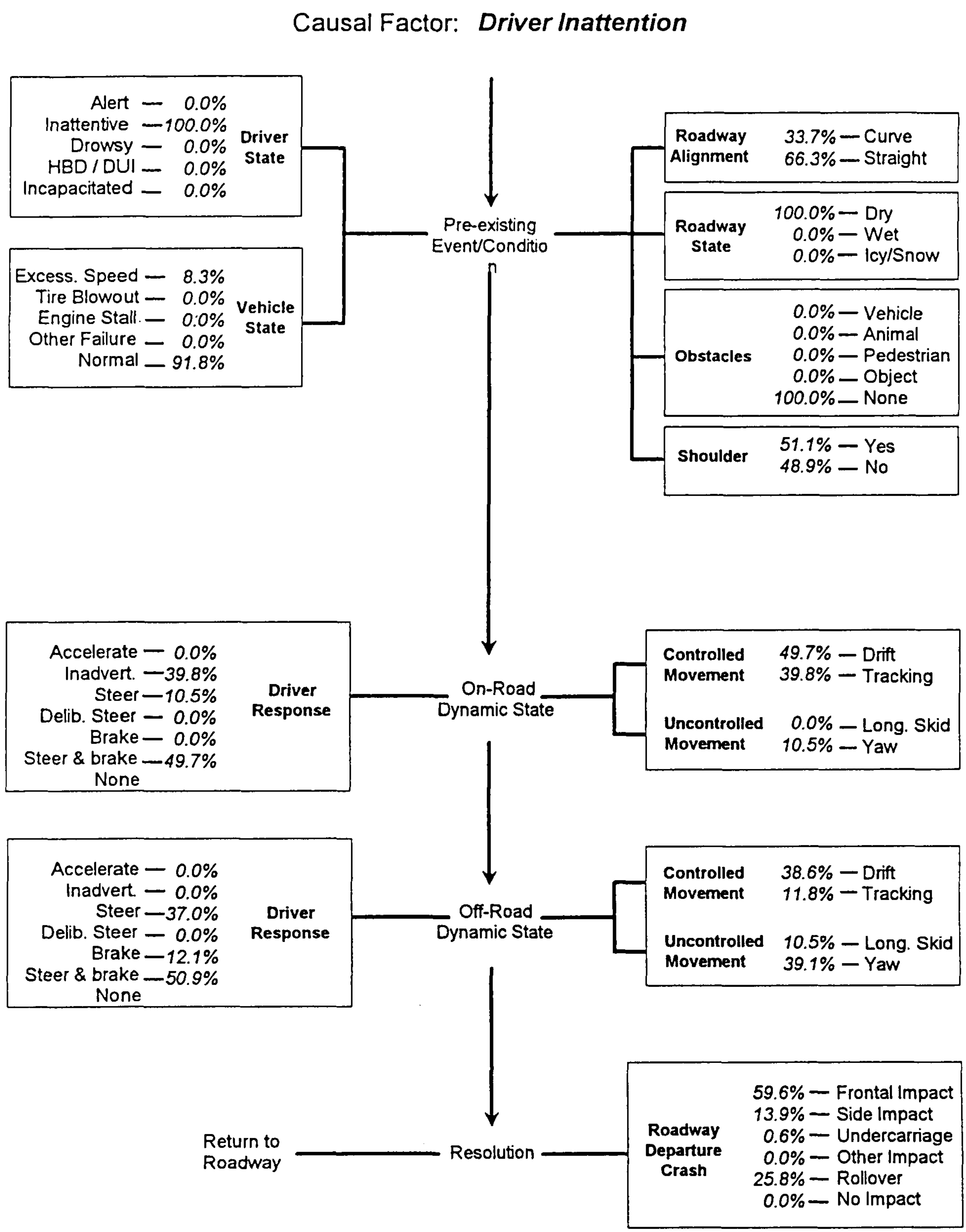

Figure 3. Vehicle Dynamic Scenario Analysis - for Road Departure Collision Related to Driver Inattention

<u>Performance Specifications for Collision Avoidance Systems</u> - The heart of the ITS collision avoidance program is the development of performance specifications for systems that can assist drivers in avoiding collisions. There are seven problem areas for which NHTSA has projects. They are rear-end collisions, road-departure collisions, intersection collisions, lane change and merge collisions, backing collisions, collisions involving drowsy drivers, and collisions associated with reduced vision.

The projects for development of the first five of these types of collisions have a common organization. Each project is divided into three phases. The phases and individual tasks are shown in Table 3. The first phase of these projects has been completed and each project is now in the middle of the second phase. A major part of the first phase of each of these projects was the development of a set of preliminary performance specifications. These performance specifications cover the sensing, processing and driver interface functional elements of the complete system.

Table 3. Phases and Tasks

Phase I: **Laying The Foundation**	Task 1: Crash Problem Analysis Task 2: Establish Functional Goals Task 3: Hardware Testing of Existing Systems* Task 4: Develop Preliminary Performance Specification
Phase II: **Understanding the State-of-the-Art**	Task 5: State-of-the-Art Review Task 6: Design
Phase III: **Test and Report**	Task 7: Building Task 8: Testing Task 9: Final Report

* Not Included in Intersection Project due to lack of available systems for testing

A summary of a subset of the preliminary performance specifications for sensor features is shown in Table 4 [ITS America 1995 - 3]. This summary is incomplete, but will be refined during the remainder of each of the projects. In addition to the sensing functional element, each project also included preliminary performance specifications for the algorithms which process the sensor data into meaningful instructions to the driver, and for the driver interface. These specifications are based on the analysis of data from NHTSA accident data files that was described in the earlier section on Identify Promising Crash Avoidance Opportunities, data from the use of driving simulators, and other analysis. For example, two of the projects utilized the Iowa Driving Simulator at the University of Iowa. This is a moving-base simulator which can be programmed to provide subjects with a realistic driving experience. The simulated routes used in an experiment to test rear-end collision avoidance systems and an experiment to test road departure collision avoidance systems are shown in Figure 4 and 5, respectively. An example of a rear-end collision warning algorithm and graduated visual display that could be used are shown in Figure 6 and 7, respectively. An example of an algorithm for a lane change collision avoidance system is shown in Figure 8.

The vision enhancement systems for nighttime and inclement weather project is investigating the feasibility of equipping motor vehicles with vision enhancement systems to help drivers avoid collision at night and in inclement weather because of reduced visibility. It will address the visual information requirements for successful crash avoidance, as well as driver useability requirements, to ensure that supplementary vision enhancement systems do not distract drivers or otherwise degrade their overall driving performance. Two approaches are being used in this ongoing study. One is to analytically model the performance of passive infrared sensors under a variety of climatic conditions. The other is to expose drivers to reduced vision conditions with, and without, the assistance of a prototype vision enhancement system. The results of these studies will form the basis for determine the next steps in the program.

The driver status and performance monitoring research is addressing the concept of a vehicle-based device to unobtrusively monitor driver performance and, potentially, psychophysiological status. The device will monitor driver status/performance, detect degraded performance and provide an appropriate warning signal or other countermeasure to prevent its continuance.

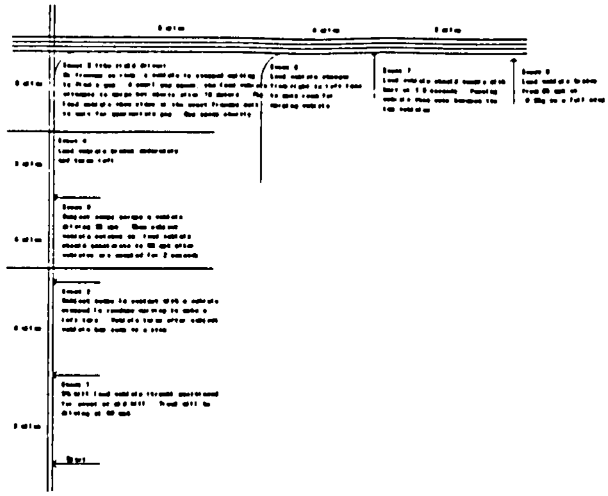

Figure 4 - Rural Highway Used for Rear-End Collision Tests in the Iowa Driving Simulator

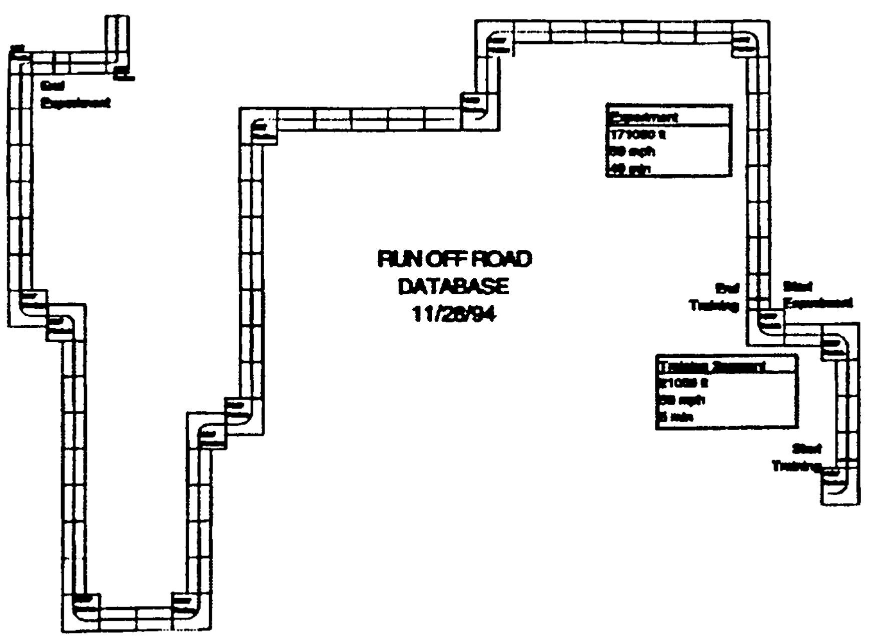

Figure 5 - Scenario used for Driving Simulator Road Departure Tests in Iowa

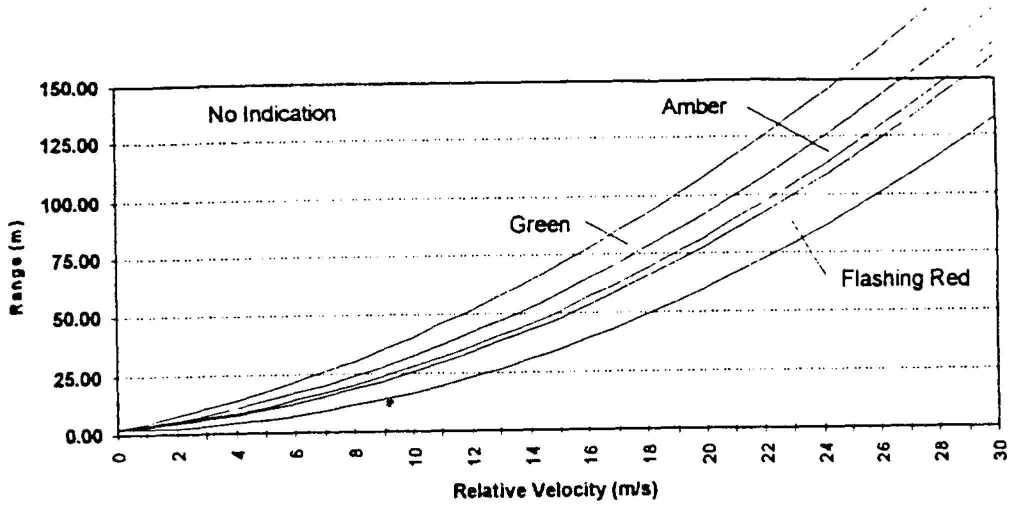

Figure 6 - Graphical Description of Driver Warning Algorithm for Rear-End Collisions

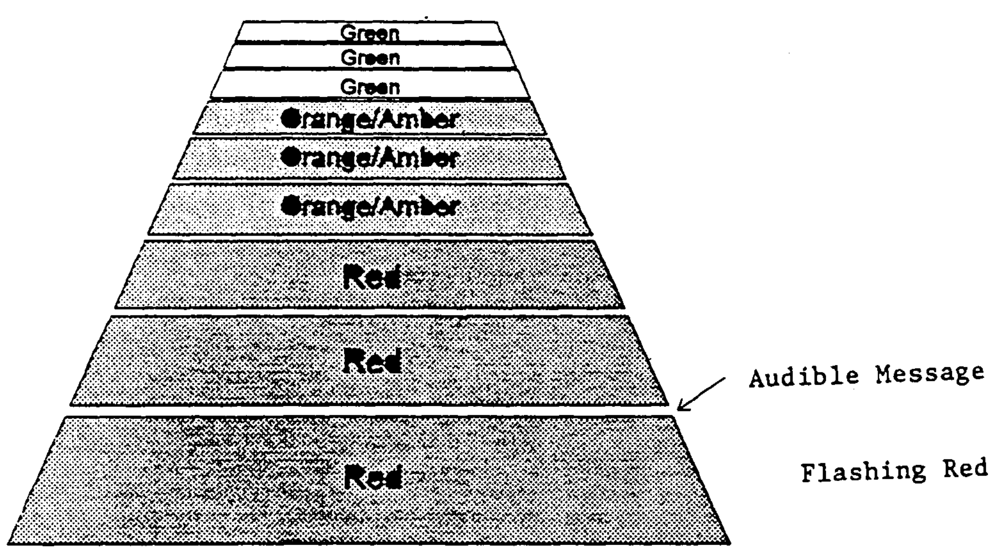

Figure 7 - Depiction of a driver interface for rear-end collison warning

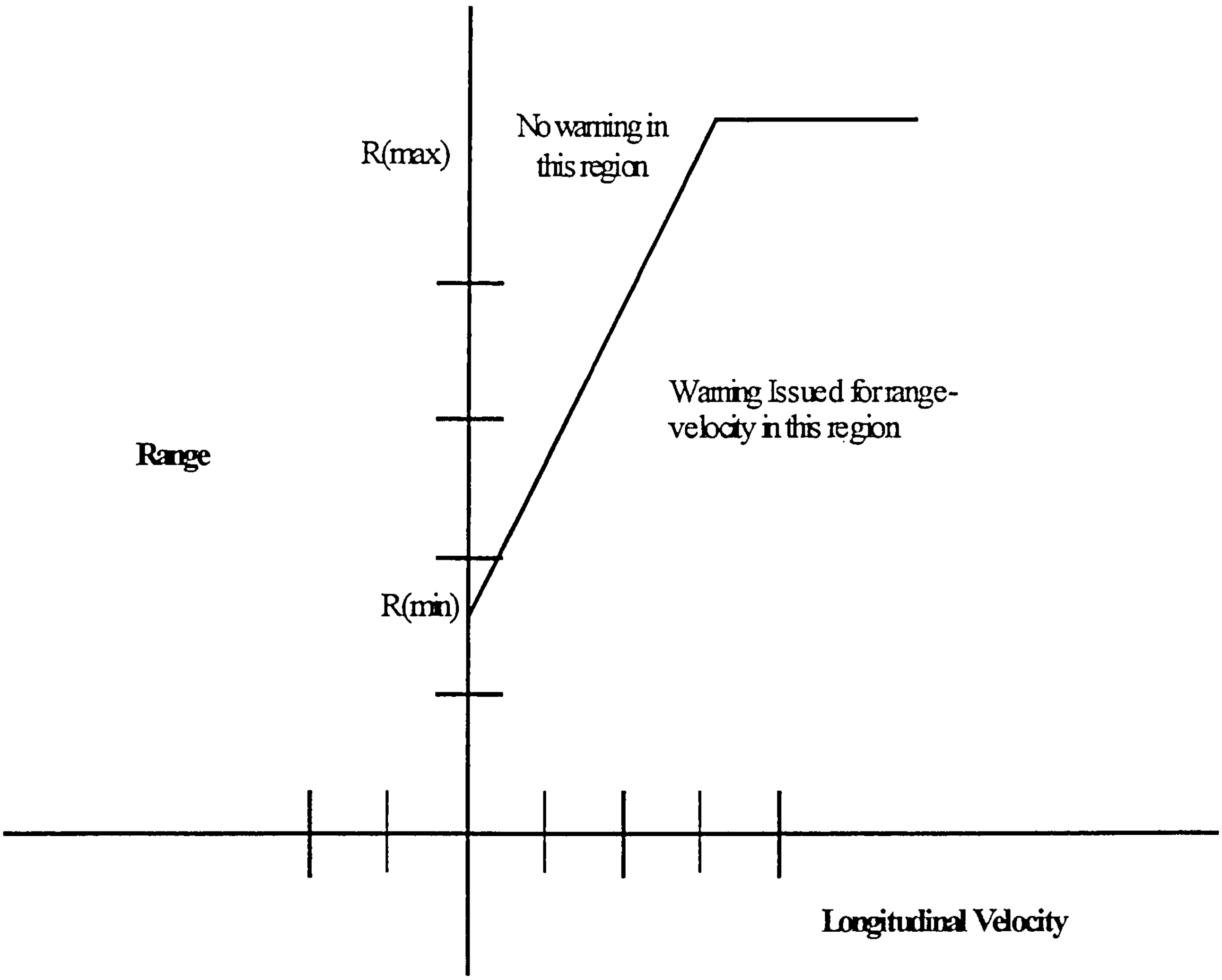

Figure 8 - Conceptual Warning Criteria in Range-Relative Velocity Spce for a Class of Lane Change Collision Avoidance Systems

Table 4. Preliminary Specifications for Sensing Functional Element

Specification	Rearend	Lane change	Backing	Intersection	Road Departure
Type of Sensor Beam	Multi-beam or scanned	TBD	TBD	TBD	TBD
Horizontal Field of Regard	+/- 8 deg	90 deg	width of vehicle	+/- 110 deg	NA
Horizontal Angular resolution	1.6 deg	TBD	TBD	TBD	NA
Acquisition range	>400 ft	(TBR) 80 ft	(TBR) 13 ft	300 ft	TBD
Range Accuracy	+/-2 ft	+/- 2 ft	TBD	TBD	TBD
Range rate accuracy	+/- 1 ft/sec	(TBR) +/-5 ft/sec	TBD	+/-1 ft/sec	TBD
Subject vehicle speed accuracy	+/-1 ft/sec	TBD	TBD	+/-1 ft/sec	+/-4 ft/sec
Accuracy of lateral position	NA	+/-2 ft	TBD	TBD	+/-0.1 ft
Minimum radius of curvature that can be accommodated	TBD	TBD	TBD	TBD	200 ft

TBR = to be reviewed
TBD = to be determined
NA = not applicable

A prototype of this monitor is being prepared for testing in a fleet of heavy trucks.

In addition to the projects listed above, the NHTSA is investigating the feasibility of equipping motor vehicles with high-technology sensing and communications systems for automatically informing emergency medial service (EMS) dispatchers of the occurrence and location of a collision. These systems should speed EMS response by providing exact crash location, especially in rural areas. This will reduce the time it takes to get medical attention and thus reduce the injury consequences of the crash. The goal of this work is to provide improved notification and delivery capability that will help provide hospital-level medical care as early as possible following onset of the trauma. An operational test has been initiated in the area around Buffalo, New York. This is discussed in more detail in the section on new initiatives.

Working with the motor vehicle industry to facilitate deployment of effective safety-related systems

The ultimate goal of NHTSA's overall involvement in the ITS program is safety-effective commercialization by industry. To be effective, vehicle-based safety devices must be available to, and purchased by, the motoring public either as standard or optional equipment on new vehicles or in the after market. To ensure that NHTSA's activities do complement industry, the agency is utilizing a variety of cooperative activities that will help in the successful deployment of safety-effective products.

Human Factors Aspects of Autonomous Intelligent Cruise Control - This project is addressing the range of human factors/driver acceptance issues associated with implementation of an intelligent cruise control system. The project is at the stage of using an instrumented vehicle with variable driver interface features to gather data on driver acceptance and performance.

Forward Crash Avoidance Systems based on Intelligent Cruise Control - This project is evaluating varying levels of deceleration through throttle control, transmission down-shifting, and utilization of service braking as critical components of either ICC or crash avoidance systems.

An example of results is shown in Figures 9 and 10. Figure 9 shows the distribution of following distances (range) and closing speed (-range-rate) for drivers who drove a circuit of specified public roads without an ICC. Figure 10 shows the same distribution when an ICC was used. The obvious difference is that drivers with an ICC maintain a much tighter range/range-rate pattern than do drivers without an ICC. These results will be part of the foundation for exploring how ICC systems can be extended to collision avoidance systems.

Forward Looking Automotive Radar Sensors - This project is gathering data on the radar cross-section (at 94 GHz) of a variety of vehicles and other roadway objects. These data will assist developers of forward looking collision avoidance systems which utilize radar sensors. An example of strength of the return signal as a function of observation angle is shown in Figure 11.

Vehicle Lateral Position Data Collection and Analysis - The objective of this project is to collect and analyze dynamic "vehicle lane position data" to support the feasibility study of using continuous monitoring of vehicle lateral lane position as a means of determine the safety status for the driver and vehicle. The project objective is to be achieved by developing, testing, refining and calibrating a lane position measurement system both in the laboratory and in full-scale on-the-road vehicle tests. The project is scheduled to be completed by October 1996. Present activity is concentrating on the analysis of field test data to determine the sources of vehicle lane position variablilty for refining driver warning thresholds. Future work will involve the testing and analysis of a two camera system for comparison with the present single camera system.

Automatic Braking for Heavy Vehicles - This project studied the feasibility of the concept of automatic braking for heavy vehicles. One part of the project was to identify design requirements necessary to accomplish assisted braking through modification of existing antilock brake/traction control system components including associated costs and benefits for potential accident reductions. Another part studied how to provide an early indication of driver reaction to assisted braking under controlled conditions.

TravTek - TravTek was an operational test of an advanced motorist information system that took place in Orlando, Florida in 1992 and 1993. The test vehicles combined vehicle navigation and tourist information with up-to-the-minute traffic data to improve driver efficiency. TravTek was a joint venture of General Motors, the American Automobile Association, the State of Florida, the City of Orlando, and the U.S. Department of Transportation. The primary objectives of the project were to determine the technical feasibility of such a system, user acceptance, and reduction in travel times. As part of the project, a thorough study of the impact on safety was performed [SAIC 1996]. The safety study included an evaluation of TravTek incidents and accidents as well as modeling the potential safety impacts of TravTek. This study indicated that the TravTek system did not impose an added safety risk and has the potential to improve safety, by such means as helping drivers avoid wrong-turns.

Advanced Driver and Vehicle Advisory Navigation Concept (ADVANCE) and Faster and Safer Travel Through Traffic Routing and Advanced Controls (FAST-TRAC) - Both of these projects are also operational tests of route guidance/navigation systems. ADVANCE is in the northwestern suburbs of Chicago, Illinois and involves vehicles which serve as probes for providing travel time data to a traffic information center. A general user acceptance evaluation of the ADVANCE system, using 30 vehicles and volunteer drivers, was recently completed and an evaluation of the safety impacts of the ADVANCE system is currently in progress. FAST-TRAC uses infrared beacons at critical locations in the network to provide a continuous exchange of real-time traffic and route guidance information. Vehicles are equipped with a route guidance and driver information system that interacts with the infrared beacons. The tests are expected to be completed by the end of 1996 and each technical report on topics such as natural use and yoked driving will include a discussion of the safety impact.

TravelAid - This project will evaluate the effectiveness of variable message signs and in-vehicle displays in improving safety along a 40-mile stretch of heavily traveled Interstate highway that is prone to snow, ice and poor visibility. Electronic sensing and equipment will be installed to monitor traffic, speeds, and road/weather conditions. This information will be the basis for determining appropriate speeds and other advice to drivers. Variable message signs will broadcast warnings about road conditions, accidents, or slow-moving equipment, as well as appropriate speeds.
In addition, an in-vehicle device will display to the driver a text message similar to that displayed by the variable

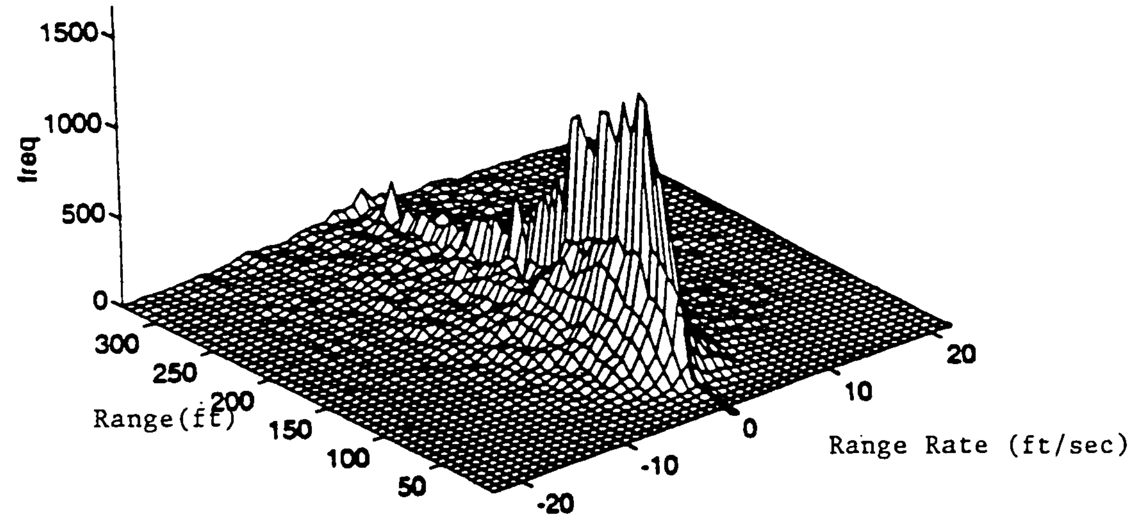

Figure 9 - Data from Intelligent Cruise Control Experiment - Headway Control Performed Manually By 36 Drivers

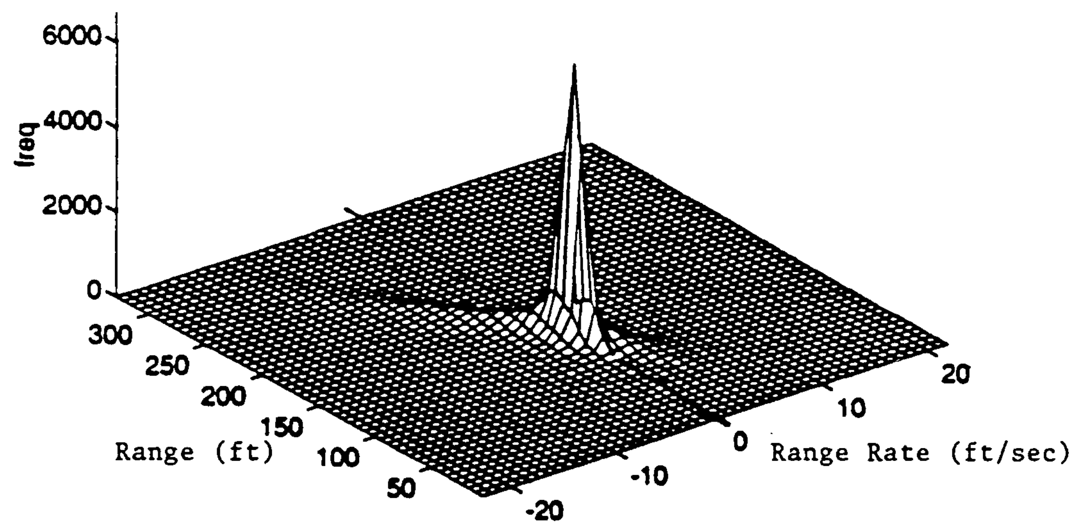

Figure 10 - Data from Intelligent Cruise Control Experiment - Headway Control Performed By An ICC System

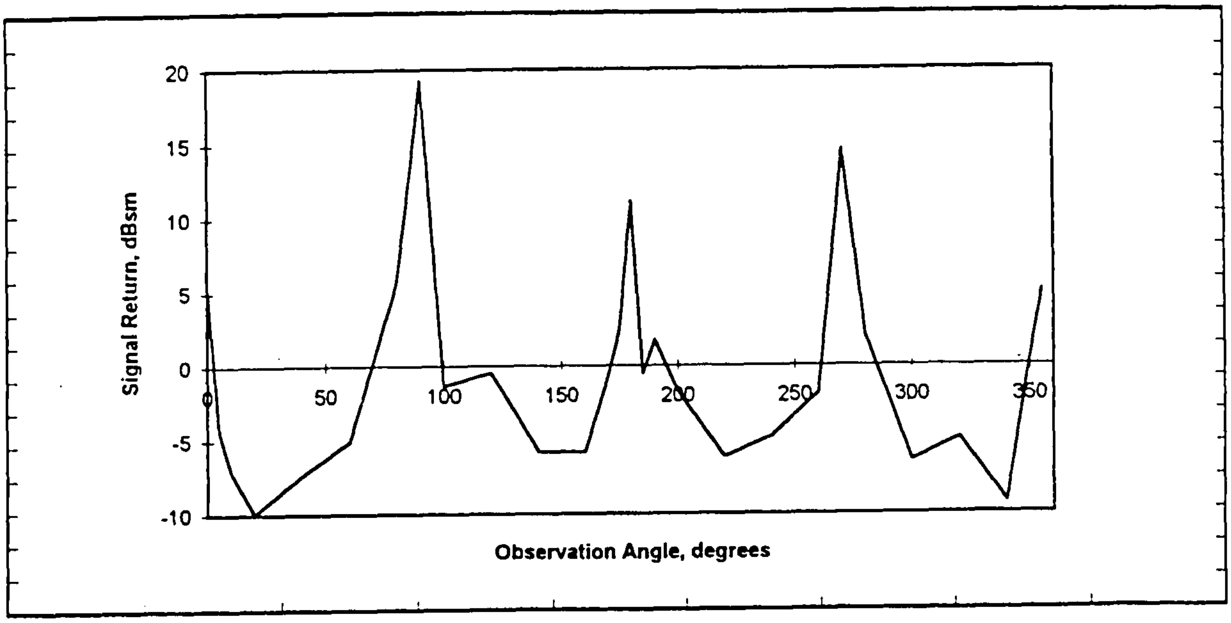

Figure : 11 Signal Return vs. Observation Angle for Corvette
(Data collected at 94 Ghz)

message signs. Up to 200 vehicles will be equipped with the in-vehicle devices.

It is expected that most of the data collection will occur during the winter of 1996 - 1997.

EVALUATION OF BENEFITS

The second NHTSA paper at the 14th ESV, Burgett 1994, described a methodology for estimating the safety impact of Intelligent Transportation Systems. The evaluation of the safety impact of the TravTek route guidance/navigation system that was being done at that time was used as a points of departure for the discussion. That TravTek evaluation is now complete and is described elsewhere in this paper. The paper also described how the methodology could be generalized to estimate the safety benefit of collision avoidance and other types of systems.

Subsequently, a workshop was held to explore the full spectrum of issues associated with estimating the benefits of safety-related intelligent transportation systems. This workshop was jointly sponsored by ITS America and the NHTSA [ITS America 1995-2]. The workshop included presentations by a number of experts in the field as well as presentation of case studies for systems such as air bag restraint systems and the automated highway system. Some of the points made at this workshop were:

- Research and evaluation should be proactive, beginning with evaluative thinking at the outset of system planning and continuing through design, testing, and deployment of a technology.
- Safety evaluations should employ a variety of methods and measures, including statistical approaches to crash data and hazard analysis.
- An important element of safety evaluation is the task of identifying appropriate empirical measurements and the means for extrapolating them to measures of safety effectiveness. In this regard, empirical studies should connect our knowledge of "the measurable" to that which is ultimately important future safety.
- The effort of safety evaluation must bridge the space between driving simulator studies, test track experiments, limited testing of prototypes and the long range effectiveness of systems in the "real world".
- Use or near-misses as an empirically-observable measure may be one way of spanning this gap.

The most recent step in the process of benefits estimation has been establishment of a task force within NHTSA for the purpose of developing "best-estimates" of benefits for selected collision avoidance systems. The task force reviewed all available data from experiments where drivers drove with and without collision avoidance systems. The empirical data were combined with dynamic and statistical analysis to produce "best-estimates" for several collision avoidance systems. The first step was to define the relevant crashes. Each countermeasure is capable of addressing only certain crash scenarios under each of the various crash conditions. These subsets are defined as relevant crashes. The crashes that could be prevented with an ITS system are calculated as a product of the number of relevant crashes in a crash condition and the effectiveness of the countermeasure system under consideration, and then summing them for all crash scenarios. To estimate each effectiveness value, a mathematical model was constructed to describe the crash condition and analyze individual crash scenarios within that condition, their likelihood of the occurrence, and then estimating the number of crashes reduced, i.e., an effectiveness value of the countermeasure for each crash scenario. The basic data for estimating effectiveness come from studies such as those using driving simulators where subjects drove through a simulated roadway course that contained curves and other features, which are often associated with collisions. Similarly, data on driver reaction times and other features, such as braking capability were obtained and used as input to a computer model. Additionally, data from experiments in which drivers were asked to follow a specified route on sections of public roads were also used. Expert judgements were also occasionally used by the task force when experimental or simulator data were not available. It should be noted that in estimating effectiveness certain assumptions were necessary. For example, it is assumed that the market penetration is 100 percent for these systems and they are assumed to be in continuous use. Further, it is assumed that the drivers respond to the warnings and alerts given by the vehicle and take the necessary evasive actions, and that system reliability is 100 percent. Finally, it is assumed that the effects of false alarms and risk compensation (i.e., driving in a more risky manner in an ITS equipped vehicle) was negligible. A summary of the conclusions of this group is shown in Table 5. These results were presented at the recent 1996 Annual Meeting of ITS America and will be explained in detail in an upcoming report from the task force [Recht 1996].

Table 5: Crashes Prevented By Selected Cras Avoidance Systems

Crash Condition (1)	Total No. of Crashes (2)	Relevant Crashes Addressed by Countermeasures (3)	Effectiveness Estimates (4)	Number of Crashes Reduced (3) x (4)
Roadway Departure	1.2 million	458,000	0.65	296,000
Lane Change/Merge	0.2 million	192,000	0.20	39,000
Rear-End Crashes, Driver Warning	1.7 million	1,547,000	0.49	759,000

NEW INITIATIVES

Intelligent Cruise Control Field Operational Test - In September 1995 NHTSA signed a cooperative agreement for an Intelligent Cruise Control Field Operational Test (ICC FOT) with the University of Michigan Transportation Research Institute (UMTRI). The members of the ICC FOT team are UMTRI, Leica AG, Haugen Associates and the Michigan Department of Transportation. The independent evaluation for this test will be conducted by the Volpe National Transportation Research Center. This two year test will evaluate the improvements offered by ICC systems in a naturalistic setting and serve as a bridge between research and deployment of ICC systems. The test and evaluation will be conducted with a fleet of ten identically equipped passenger cars. The test plans to provide 324 weeks of ICC exposure to up to 260 participating drivers while accruing between 130,00 to 200,00 vehicle-miles of operation.

The results from this test will be used to assess basic safety issues by determining how well the ICC system provides separation control with respect to a preceding vehicle. Additionally, the evaluation will also assess the potential for decreasing the number and severity of rear-end collisions through the use of the ICC system.

Automated Collision Notification Operational Test - Also in September 1995 NHTSA signed a cooperative agreement for the Automated Collision Notification Operational Field Test (ACN FOT) with the Calspan Corporation. The members of the ACN FOT are the Calspan Advanced Technology Center, the New York State Department of Transportation, General Motors, Cellular One, Rockwell, Erie County Emergency Management Service, Erie County Community College, and Datumtech. An ACN System is an advanced in-vehicle system that will automatically alert Emergency Medical Services (EMS) in the event of a serious automobile crash. The independent evaluation for this test will be conducted by the John Hopkins University Applied Physics Laboratory. This two year test will evaluate the improvements offered by ACN systems and serve as a bridge between research and deployment of ACN systems. The team will design, build, and deploy an ACN System, using 1000 privately owned cars in a large area covering the western portion of New York State. The test plans call for each of these vehicles to be driven by the owner/operator for one year. It is expected, based on an analysis of accidents in the western New York area, that between 10-50 injury level accidents will occur during the test.

The field operational test seeks to determine if the ACN system can provide a reduction in the notification time for collisions. Additionally, the evaluation will assess the impact on EMS operations and trauma care facilities of deployment of an ACN system.

Consortium for Advancing the State-of-the-Art of Collision Avoidance Technologies - This is a cooperative research agreement with a consortium of automotive industry research partners to advance the state-of-the-art of collision avoidance technologies toward commercialization and facilitate introduction of products into the automotive marketplace. This effort is an integral part of NHTSA's focus on facilitating development and production of effective safety systems. The project is supported by the Technology Reinvestment Project of the Advance Research Projects Agency of the Department of Defense.

Activities are focused on achieving the goal of improving performance through use of the existing technologies and increasing system manufacturability by developing methods to reduce the cost of system components. Throughout the program, prototypes of collision avoidance systems will be incorporated into test vehicles as part of the test and evaluation program and demonstration of system operation. The project will be completed early in 1997.

Heavy Vehicle Electronic Braking/Brake Performance Systems - This project is in response to a 1991 study that found brake system performance could play a contributing role in approximately one-third of all medium/heavy truck crashes. This cooperative agreement is assessing the current knowledge/technology and performing limited laboratory testing related to electronic braking systems. This will be followed by pilot test and analysis of cost, reliability and maintainability. The results will form the basis for development of a functional specification for an optimal system.

SUMMARY

In 1991, the National Highway Traffic Safety Administration embarked on a major new collision avoidance initiative to improve the level of injury prevention on the nation's highways. This work is a part of the larger initiative within the Department of Transportation, and the entire surface transportation community, to develop Intelligent Transportation Systems to provide a wide range of services to all users of the nation's highways. [ITS America 1995-1]. The ultimate goal of the DOT program and related industrial programs is to have effective ITS-based collision avoidance capability in motor vehicles early in the 21st century.

This paper describes the progress that has been made in the NHTSA program. Four significant new research tools are being developed. For example, a prototype of the Data Acquisition System for Crash Avoidance Research (DASCAR) has been completed and the design and construction of the National Advanced Driving Simulator (NADS) has begun. A thorough and extensive analysis of accident data has been completed and the results are being used as the basis for development of countermeasure concepts. Preliminary performance specifications have been formulated for five types of collisions: rear-end, lane change and merge, backing, intersection, and road-departure. These preliminary specifications will be refined through extensive testing in remaining phases of the projects. Also, a number of significant projects have been completed through joint efforts with motor vehicle industry partners. Among these are the completion of a ground-breaking analysis of the safety impact of the TravTek route guidance and navigation system, demonstration that automatic braking is compatible with current antilock braking systems on heavy trucks, acquisition of data on use of an intelligent cruise control system, and data on radar cross-sections for vehicles and roadway features are a few of these advances. As a total contribution, this work promises to provide a solid and balanced basis for moving forward toward improved collision avoidance capability for the motoring public.

ACKNOWLEDGMENTS

The work described in this paper is the product of efforts by a large number of contractors and other partners who have worked with the staff of the NHTSA Office of Crash Avoidance Research and significant help from the staffs of the Volpe National Transportation Systems Center and NHTSA Vehicle Research Test Center.

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PSA EXPERIMENTAL SAFETY SUBSYSTEMS IN VSR

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Paper Number 96-S2-O-10

ABSTRACT

The Experimental Safety SubSystems (ESSS) proposed by PSA Peugeot Citroën in the Véhicle and Safety on Road - VSR - program are based on the information coming from accidentological activities; they are the result of synthesis studies performed in the frame of accident avoidance, primary safety and occupant protection:

- ESSS 1 are devoted to electronics and intelligent vehicle developments for crash avoidance;
- ESSS 2 present new methods for dynamic behavior improvement in order to prevent the driver to be in critical driving condition;
- ESSS 3 given up to enhanced passive safety conditions aim at increase occupant protection in case of accident.

The ESSS have been developed using project management between the various Research and Development Departments of PSA Peugeot Citroën.

The goal of SubSystem of the 1st type is to study information assistance devices that can alert the driver on hazardous driving conditions. They include specific studies related to vehicle driver interface and specially ergonomics.

A great part of the work pay attention to the driver behavior evaluation through the vehicle control devices.

Dynamic behavior SubSystems of 2nd type want to develop the concept of global dynamic control of the vehicle platform to ground relation, that is to say, adjust, at any time, in the optimal way, the vehicle dynamic functions taking into account the intentions of the driver and the characteristics of road surface.

This part of the program present principally now :

- a static driving simulator as a special tool for study, simulation and evaluation of global suspension control strategy;
- the basis of a running experimental vehicle for a final evaluation of the concepts.

SubSystems of 3rd type, related to passive safety and occupant protection are built on a 3 level approach :

- better knowledge of materials behavior and failure prediction during high speed deformation;
- improvement of mathematical modeling of crashworthiness in order to reduce feedback delay during car design with the use of simplified models;
- realization of practical automobile applications for frontal and lateral impacts.

ESSS Type 1 Integrated Electronic Safety System

Previous studies as PROMETHEUS had made possible the constitution of electronic warning and assistance device collection. Individually, each of those devices must be considered as running well and being effective.

The integration of one single electronic device into a car does not induce any problem in terms of technology or driver ability to understand correctly all the messages delivered by his new environment into the car. The problem arises when a great number of that kind of new function have to be introduced and used simultaneously.

The aim of ESSS1 project is to reach a complete control of the technology in order to obtain a simultaneous working of the various functions with a complete reliability. But also, it is important, at the same time, to check the acceptability and conviviality of the whole system and his ability to deliver convenient messages easy to understand. This can lead to develop completely new driving post.

For that purpose, it has been judged necessary to integrate physically in the same vehicle devices like on-board video camera, RDS-TMC receiver, LIDAR anti collision sensor, infra red receiver and screen, AICC system .....

This action have been made in four steps :

• enhance the visibility of the display;

• imagine a new Information System Instrument Panel;

• analyze the driver behavior;

• integrate all the elements on a the synthesis vehicle « ALTO »

A basic study on Technologies for Display

The main technologies have been studied by Magnetti-Marelli in connection with GIE PSA Peugeot Citroën, mainly Liquid Crystal Display (LCD), Active Matrix Thin Film Transistor(AM TFT) and Vacuum Fluorescent display.

The table 1 give the advantage vs. disadvantage balance between those display technologies.

Passive LCD can be used with 3 technologies - reflective, transflective, transmissive - but the first one cannot be used in automobile due to the fact that display must be seen lighted naturally during day and from the rear during the night.

Active Matrix Thin Film used presently on portable computers is probably a good solution for automobile display screen as multi-function screens.

Table 1

Comparison of different types of displays

	AVDANTAGE	DISADVANTAGE
Passive LCD	• Low cost • Well known technology • Low consumption & over-heating	• Slow display • Angle vision & contrast
AM TFT	• Angle vision & contrast • Fast commutation • Wide color choice	• High cost • Complex technology
VFD	• Works at extreme temperature • Angle vision & contrast • Strong luminosity	• High cost • Complex technology • Bulky

The Vacuum Fluorescent Display has the advantage of a very high luminosity and brings a good solution for automobile Head Up Display.

On this basis, two mock up have been developed. Each of those realization was fitted with a central display between driver and front passenger and a secondary display located just under the windshield.

Information System Instrument Panel

In front of the increase of new functions proposed to the motorist, the information on-board system aim to make the use of these functions simple, clear and fast. It can command functions for audio center and air-conditioning, inform on travel conditions and technical state of the vehicle through on-board computer. The main device is constituted by the display located on the central support and in rest the driver can use a monochromatic display, just in front of him, at the basis of the windshield who indicate instantaneous speed and the gasoline volume remaining in the tank.

The central display is controlled by a collection of six specialized menus for air-conditioning, audio control, on board computer, diagnostic, telecommunication, navigation.

Driver behavior

The purpose of this study is to detect failures in the driver's behavior that can be caused by lack of attention, drowsiness, nervous excess, ... It is driven as an extension of previous studies which have been conducted as, by example, the steering wheel movements analysis (RENAULT,NISSAN) but without real reproducible results. The present study has been conducted with Heudiasyc - Compiegne University and Research Laboratory Ecole Centrale Paris. The principle stays in driver observance in order to detect any way of driving modification whatever the cause (fatigue, drowsiness, lack of attention ...). The system must be non invasive and able to predict. It changes by itself without any predetermined knowledge. The driver initial state is « learned » by a neural network progressively on line through the several sensors installed on the main commands of the vehicle : accelerator, steering wheel, video camera for positioning, ... On these basic information, the system establishes an adaptative diagnostic through a fuzzy pattern recognition process.

Figure 1 shows a class example corresponding to one state of a driver. This reference state has been learned between 0 and 4 minutes and appears again successively at 12, 17, 28 and 33 minutes.

PSA has conducted in relation with EPAP[1] Laboratory two series of experimentation.

One aimed to detect evolution in driving activity. It is operated during a 300 km trip on highway with an assistant observer. It has been demonstrated that the system can work in real time on-board. and that evolution can be interpreted during the experiment and a posteriori. It was confirmed the capability of the system to isolate various states of driving as it is shown on Figure 2.

[1] « Electronique Physique Appliquée et Productique » de l'Ecole Centrale Paris

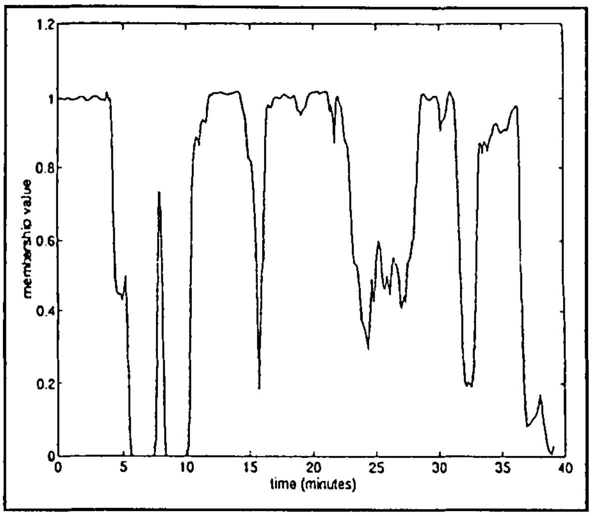

Figure 1 Example of one class corresponding to one state of the driver and one neuronal network.

The membership to one class is detected when the value delivered by the associated network is near to 1. The continuous and dotted lines are corresponding to two different classes where the up step means that the system goes to a class i and the down step that the system is leaving a class i.

The other series of tests were oriented to drowsiness detection in connection with LAA - Pr Coblentz. The subjects drove on highway under favorable conditions for drowsiness. It is more difficult to establish a strong correlation between the driving state measured by the system and drowsiness evaluated by physiological measures. Nevertheless, one can establish that usually a class change occurs just before the beginning of a drowsiness period. This can be seen on Figure 3 where the horizontal lines in the upper part of the diagram are corresponding to drowsiness periods.

The present balance of the program show that the proposed system is relevant to detect changes in driving activities. It can self adapt easily to a given driver or road environment and works pretty good in real time. But, at the moment, it must be confirmed on the basis of a larger number of different drivers and on common road journeys. This issue must be solved before launching a full scale demonstration on a fleet of passenger cars. It will be the topic of the second half of the VSR project.

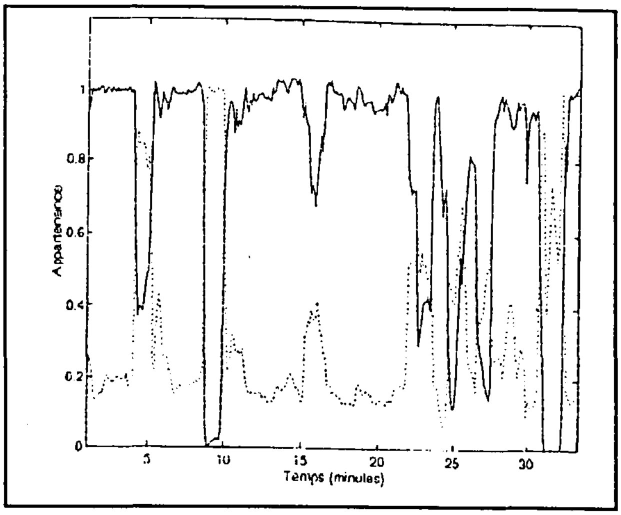

Figure 2 Record showing that driver goes from class i to class j.

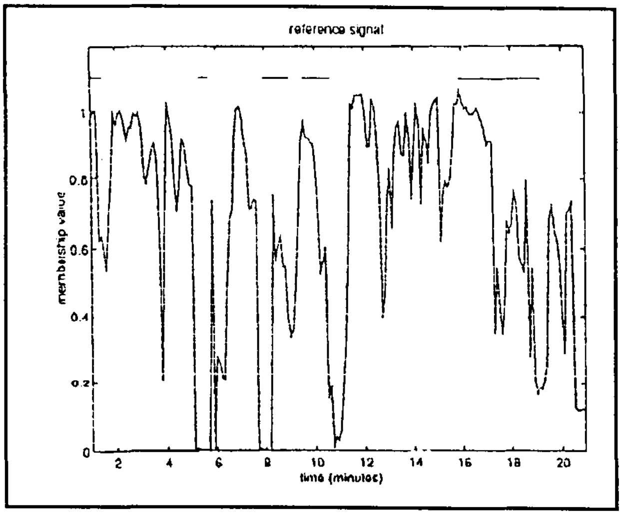

Figure 3 Correspondence between drowsiness and neuronal network analysis

A synthesis vehicle -ALTO -

An experimental synthesis vehicle - ALTO - has been realized by PSA PEUGEOT CITROEN based on several computers connected with multiplexed BUS. It contains guidance, lane keeping, Intelligent Cruise Control, vision enhancement and digital cartography.

Vision enhancement improve vision by night thanks to infrared head lamps, camera CCD and a screen on the dashboard.

Through vehicle motoring the system makes a diagnostic of the suspension and brakes. The driver is alerted that the suspension is wrong by an emergency red light.

A lane keeping function is achieved by a camera, a computer and an active command of the steering wheel.

The obstacle detection is achieved using a laser telemeter, video cameras and computer.

By the way of a microwave system, the experimental vehicle can perform cooperative driving and establish communication between infrastructure and cars and between cars.

Autonomous Intelligent Cruise Control (AICC) uses a LACA telemeter (Lidar infrared), a classical control command, a display for information. The telemeter checks the distance and relative speed with the vehicle ahead. The car is hold at the safety distance of that vehicle. If the follower car changes of lane, it comes back to the initially programmed speed.

Moreover, ALTO is equipped with EUROSCOUT as an interactive infrared guidance system, connected to on-board information display. Information is exchanged through infrared beacons working as traffic sensors.

The autonomous guidance is obtained through a system using a numerical map on board on which the driver gives the destination and the system answer the best route.

The traffic information is supported by the lateral band (RDS - TMC) of the FM network with data coming from sensors integrated into the road. The driver gives its destination and the system propose several different routes with, for each of them, the journey duration times.

ESSS Type 2 Active Safety System

The dynamic behavior demonstrators intend to help the development of generalized control of the relation between vehicle and ground. We want, at any time, insure that we have an optimized adaptation of the dynamic functions of the car to the driver's orders and to the characteristics of the vehicles and the road.

The work made in the area of primary safety has combined the development of a new study tool, consisting in a steady state economic driving simulator - static demonstrator-, and the elaboration of a new suspension control concept - road demonstrator -.

A static demonstrator : mini driving simulator.

The program intend to develop a technical, methodological and software real time tool able to lead research aiming to improve primary safety. This tool, where we have integrated the association driver/car, must be predictive at the beginning of new car design and allow to do easily parameter studies.

At the present state of evolution, it allows to support cooperation on closely related subjects as :

- test and assessment of driving assistance device;
- analysis of accident causation from Fatal Accident Statements and Detailed Accident Studies;
- more in depth analysis of driver behavior as explained in above mentioned ESSS Type 1 program.

Moreover, it is an appropriate platform in order to elaborate a methodological approach for the future use of French National Great Simulator SARA..

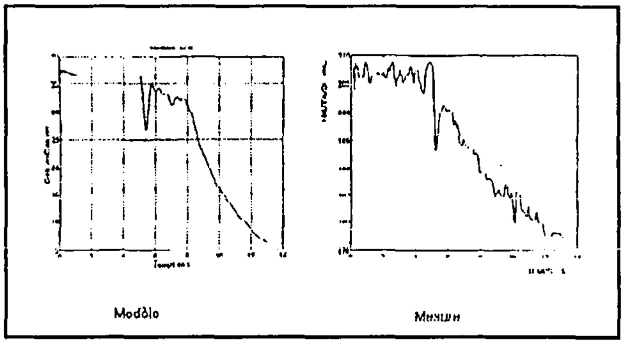

Figure 4 Validation of the vehicle dynamic model

The installation is based upon Silicon Graphic station equipped with a validated vehicle model - Figure 4 - and a real time simulation software who brings back to the test driver realistic image, sound and reaction forces into the steering wheel.

The next steps of the study will contain :

- complementary developments on the already existing ground data bases, aiming at a better fit to realistic experimentation need;
- the set up of an « intelligent » traffic which can describe, by position, speed and acceleration all interactions between the driver, the road and the other vehicles;
- the improvement of the vehicle in terms of response, especially related to ground surface parameters;
- visual cues enhancement, including field aperture and possibly stereoscopy.

Road holding demonstrator

It is today a wide challenge for car manufacturers to provide their customers with vehicle who add excellent qualities of comfort and a great safety level. One of the major way to succeed is the active control of the suspension where the board computer can soften the vehicle ground relation when possible, but stiffen it shortly and with a small reaction time when the behavior of the car make it necessary.

Through the vehicle computer, it is possible to act separately function of time on rigidity and damping of the 4 wheels of the car inducing a great difficulty for the adjustment of the suspension parameter laws through a pure experimental procedure.

Thanks to the « bond graph » methodology a physical model of the suspension has been built and used to predict, from parameter laws of suspension elements, the displacement, speed and acceleration laws vs. time of vehicle platform. These results introduced into the mini driving simulator could deliver a preliminary subjective evaluation of the demonstrator tuning before running the car itself.

In comparison with commercial Citroën XM Hydractive and XANTIA ACTIVA, the vehicule demonstrated that it was possible to obtain a full control of load transfers initiated by the driver in turn, braking or acceleration phases associated to an enhanced comfort level.

ESSS Type 3 Passive Safety System

The Experimental Safety SubSystems (ESSS 3) proposed by PSA Peugeot Citroën for passive safety is sustained by three main axis : enhanced tools to improve passive safety, car behavior during impact, effectiveness of restraint systems. It leads up to frontal and lateral impact demonstrators.

Tools to improve passive safety

The strategy of PSA Peugeot Citroën is based on the use of mathematical models of car and occupant able to be operated quickly and easily as soon as possible at the beginning of a new design. In this frame, ESSS 3 Program had to take into account the accuracy of the crash programs and enhanced methodology to operate them.

Our knowledge on material laws has to be improved especially for the effect of the deformation speed on the constraint and for the capability to compute breaking prevision of mechanical parts.

A large campaign of experimentation on material law has been performed from November 1992 to June 1995; some of the results for XES steel are presented here. Tests have been executed on traction machine into the speed range from 10^{-4}/s to 100/s and with Hopkinson bar installation from 100/s to 1000/s.

Figure 5 show a significant effect of deformation speed on the elastic limit with an increase of approximately 25% when the deformation speed goes from 50/s to 250/s.

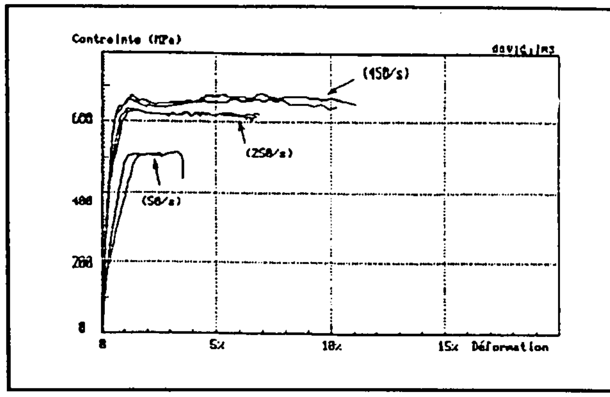

Figure 5 XES Steel - Effect of deformation speed on elastic limit

From another point of view, the mathematical formulation of the different empirical laws proposed presently has been compared with the experimental result. The main conclusions are represented on Figure 6 where are expressed exponential, logarithmic and Malvern representation associated with the experimental result.

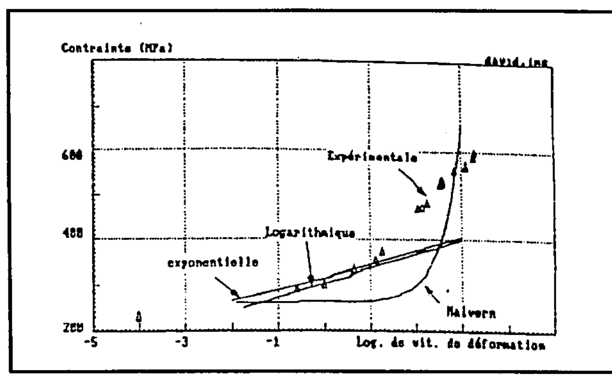

Figure 6 Representativeness of various material laws compared to test.

It appears that none of the proposed laws can fulfill the test result on the complete deformation speed kingdom. So one intend to proceed in two steps :

- for the short range, determine a set of realistic parameters for Ludwik law presently used in RADIOSS

$$= \left(\sigma_{0\,moy} + K_{moy}\, \varepsilon_p^{\,n_{moy}} \right)\left(1 + C_{moy} \ln \frac{\dot{\varepsilon}}{\dot{\varepsilon}_{0\,moy}} \right)$$

- for medium range, define then integrate in former release of RADIOSS a new material law resulting from comparison on Figure 6 and taking into account the material viscosity.

Another difficulty encountered in crash is due to the fact that some mechanical parts can break or not during the collision. The influence can be drastic for the quality of the result. So, engineers aim to have at their disposal a representative model capable to predict the rupture of the part.

The example chosen was the part insuring the connection between the engine and the engine compartment shown on Figure 7 and realized with the aluminum alloy AS7G03.

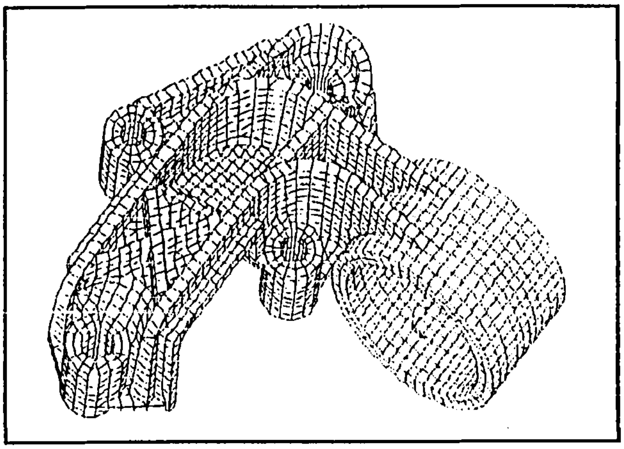

Figure 7 Engine to car body connecting support - Aluminum alloy AS7G03 -

The material law of AS7G03 has been studied through 300 experiments under the following aspects :

• influence of deformation speed on smooth test-bar;

• influence of chemical composition by test with different samples of material;

• influence of cooling speed using parts molded by sand and chill casting.

At the present time, these test have shown an influence of chemical composition on unit elongation and practically no influence of deformation and cooling speed on rupture characteristics.

Car behavior

Increase occupant protection during a crash means to obtain the best deceleration law associated to limited intrusion and optimized restraint system. Seen from car, it is necessary that one must reach three main performances leading to increase the energy absorption capability of the front structure, enhance passenger compartment resistance and control inside the various equipment displacement, especially for the steering wheel in order to prevent contact with occupant head.

Enhance energy absorption

One of the way to increase the capability of structure is to improve the connection between car body parts. A demonstrator have been built using continuous laser welding in areas well known to be heavily loaded during impact, as front sill connection with floor, roof and side body, ...

This demonstrator has been tested against rigid 30° ASD barrier at 57 km/h and compared with a production car tested in the same conditions.

The good behavior of the laser welded connections allowed to reduce considerably the backward movement of the upper part of the structure. A lower windshield pillar displacement was reduced by 35 % (50 mm) despite a slight increase of lower part intrusion (+15 mm at the dash panel level). An optimization is on the way particularly based on cost analysis.

Steering column displacement control

The backward and upward steering column displacement can induce head and thorax injuries. Moreover displacement exceeding 50 to 60 mm can lead to a dramatically reduced air bag effectiveness. In addition, manufacturers have to fulfill regulations on this point.

These considerations lead to study in priority the steering wheel displacement which is conducted mainly by the mechanical environment of the steering column and the engine compartment architectural organization.

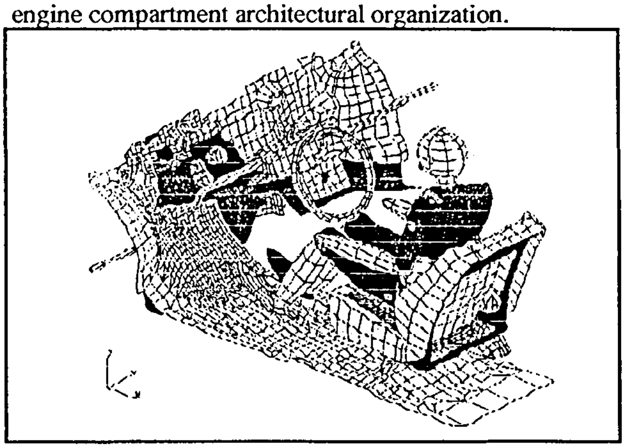

Figure 8 FEM dummy model in driving position

The aim of the study is to make product engineering capable to predict and understand steering wheel displacements from the earliest phase of the design.

To succeed in this way, we need to have a computer model of driver capable to evaluate the head trajectory (as shown on Figure 8) associated to a simplified vehicle representation.

The results obtained from this simplified model are compared on Figure 9 for the backward steering wheel displacement with those coming from full size FEM model and full size crash test. It has been demonstrated that the representativeness for X displacement was 1% and for Z displacement under 5%.

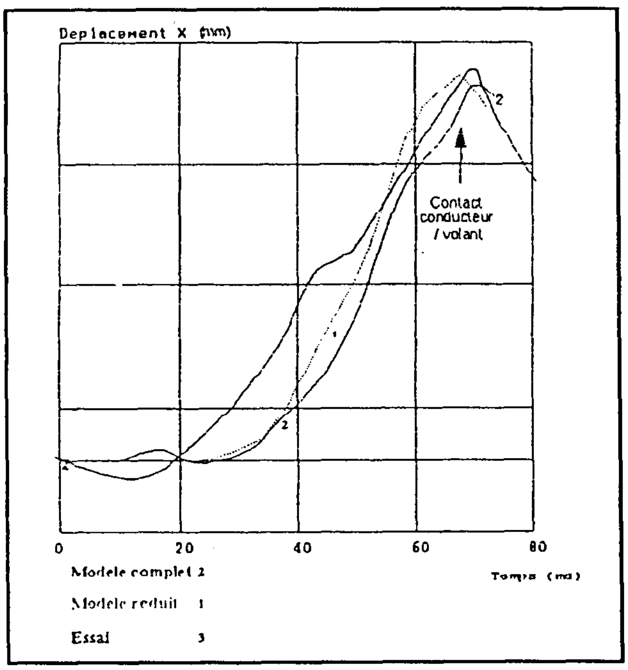

Figure 9 Backward movement of the steering wheel - Comparison simplified and full size models with test

Car deceleration law control

Through the deceleration laws control, PSA PEUGEOT CITROËN studies experimentally the process to obtain a « good time history deceleration curve » for the vehicle. Six demonstrators have been impacted at 57 km/h perpendicularly against a rigid fixed barrier in one configuration chosen only for his aptitude to provide high level energy pulse.

Many parameters concerning the architecture of the car have been modified and crossed together as :

- engine angle and position of exhaust and various accessories;
- stiffness and shape of the engine holder;
- pedalier and steering column holder;
- energy absorbing devices in front part of the sills ...

The peak deceleration in time history curve was reduced from 50 g to 35 g. In the same time, the mass of the car body structure was reduced by 20 kg with an increased level of passive safety.

Restraint systems parametric study with MADYMO

The parametric study on restraint systems wanted to define qualitatively and quantitatively the relative influences of various parameters conditioning the restraint quality during a frontal impact. This study aimed to establish recommendations for inner car fitting devices as seat, instrument panel, belt anchorage, time history acceleration curve, ...

For that purpose, MADYMO dummy model from TNO has been used after we had, first, calibrated the bounding between dummy and car through sled tests on accelerator.

In addition to the classical biomechanic criteria, this work tried to define complementary design criteria easy and fast to evaluate with simplified models. From this point of view, the main tendencies are gathered in table 2 below.

The parametric study of occupant restraint issued on a better understanding of mechanism and coupling during the crash. By example, it was demonstrated that the contact of the knees against the instrument panel increases the probability of head impact with the steering wheel but reduces the tendency of submarining.

This parametric study can help designers to reach better compromise between different aspects of passive safety.

Submarining control

Submarining is a phenomenon difficult to diagnose in a very reliable way. Hybrid II can be fitted with a submarining sensor located just above the pelvis and recording the force generated by the lap belt applying in this area. The force threshold adopted up to now is adjusted at 800N. But the observations made with high speed camera have shown that pelvis overlap can occur even then the 800N threshold has not been over passed.

It is the reason why PSA PEUGEOT CITROËN has initiated a fundamental study of the parameters who govern submarining. These parameters are summarized on Figure 10

Tests have been executed on sled accelerator with programmed acceleration time history with Hybrid II. Biomechanic criteria were measured for head, thorax and pelvis.

Table 2
Injury design criteria hierarchy

SEAT	Injuries Order of importance	CAUSE	Criteria Order of importance	POSSIBLE SAFETY DEVICE
	1-Head	Contact with steering column or steering wheel With/without intrusion	1 Displacement into R < 500 mm	Air bag or retractable column Reinforced seat Belt with retractor or blocking device
DRIVER	2-Chest	Shoulder belt (rib fracture)	2 $F_{Shoulder\ belt}$ < 5.6 kN	Force limiter
	3- Legs	Contact feet or legs with enclosure wall	2 F_{Femur} < 10 kN	Optimized instrument panel
PASSENGER	1-Abdomen	Submarining	NO submarining (see after)	Reinforced seat Belt with retractor or blocking device *<u>CAUTION</u> on initial angle of legs, buckle position and clearance of the belt*
	2-Chest	Shoulder belt (rib fracture)	1 $F_{Shoulder\ belt}$ < 5.6 kN	Force limiter

The data were completed by high speed camera recording (1000 i/s).

The parameters that varied were devoted to embossing in the horizontal part of the seat (height, longitudinal position, slope) and to the lower part of seat belt (angle and length of the lower buckle belt).

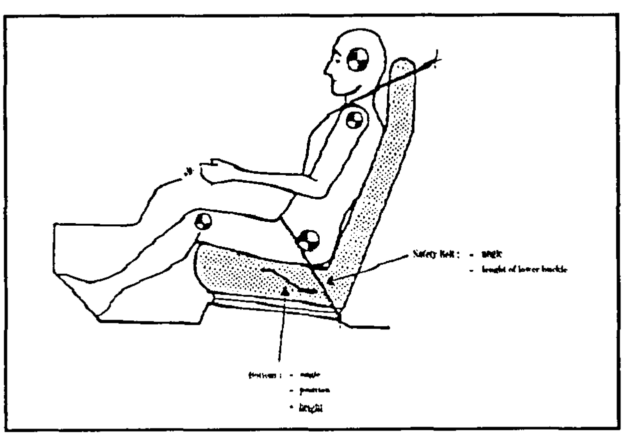

Figure 10 Parameters controling sub-marining

The main conclusions are that it is confirmed that the force level on the submarining sensor is not sufficient to diagnose the defect. This criteria must be completed by a set of other indicators as quick change in horizontal and vertical pelvis deceleration, fall of force on lap belt, rise of the force recorded above pelvis, but not its absolute value.

One can mention that a lot of other studies of less volume has been done as integration of restraint system on the seat, child restraint system, effectiveness of air-bag (bibliographic study) ...

Lateral impact demonstrator

The project of car body structure for lateral impact mix closely design, computation and experiment against a mobile deformable barrier. The impactor is made of an aluminum honeycomb block who intend to represent the mean stiffness of an European medium car.

Due to that requirement, the impactor is a very complex structure who need approximately 15 000 elements to be described correctly from a geometrical point on view. This induces unacceptable calculation time. Before the proceed to performance improvement study, it appeared necessary to develop a physical model of the barrier aiming to reproduce directly the compliance of the force displacement corridor given by the regulation.

The resulting model need only 5 000 elements and give a good correlation as well for static compression than for 50 km/h collision test.

A global model has been developed gathering the above described impactor model, a meshing of the car fitted with its reinforcements and a model of lateral impact EUROSID dummy.

Using a study on sensibility of lateral impact parameters one could obtain between optimized and standard structure definition a reduction from 11% to 43% of the side deformation leading to a correlative reduction of biomechanic criteria.

CONCLUSION

At midway of VSR french safety program, the work done and the results obtained show concrete progress on which it is possible to base the continuation of the PSA Peugeot Citroën program for the next two years.

For type 1 demonstrators devoted to prevention systems, the most significant projections concern sign of drowsiness detection. Our program plan to operate a vehicle equipped with a complete non invasive system. It will be used in a test campaign organized with several drivers using different kind of roads. That research have to be completed by a selection of the most judicious alarm criteria.

Driving assistance will be enhanced through research on multi target radar for environment detection; lane keeping have to be improved thanks to sensitive reactions of the steering wheel able to alert the driver on the fact that he is leaving his traffic lane. At last, we consider as a priority to define ergonomic rules for the driver environment in relation with the high level of information delivered to him.

In relation with Type 2 demonstrators, the « mini » driving simulator have to be made more realistic through the road holding mathematical model and the road and traffic environment data base.

For passive safety, PSA Peugeot Citroën plan to develop in priority 3 actions :

- find the best adaptation between crashworthiness characteristic and performance of restraint system;
- simplify numerical models in order to used them in the early stages of the new car design;
- elaborate an integrated methodology to apply systematically for crash vehicle conception.

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INNOVATIVE VEHICLE LIGHTING FOR ACTIVE SAFETY AND COMFORT

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Paper Number 96-S2-O-11

ABSTRACT

One hundred years of development have turned the motor car into a commodity for today's mobile society. Growing awareness for safety and increased regulation of traffic have led to significant reductions in accident fatalities. The proportion of night-time to day-time accidents however, has not altered. Around 90 percent of all accidents occur at low speeds with lack of visibility or obsured vision. Accident risks double on night drives and rise further in adverse weather conditions. A recent study on night driving in Europe revealed that women often feel uncomfortable due to insufficient visibility and that many men realise they would not be able to avoid an unexpected obstacle.

Vision provides 90 percent of all driving information. Vehicle lighting plays a major role in active safety. The introduction of halogen light sources, the arrival of projector headlamps and the recent move towards high-intensity discharge lighting increase the available light output and thus permitted the light distribution to be tailored to the prevailing driving and road conditions. Fast light sources, such as LED or NEON, in brake lights have taken meters off the braking distance of following vehicles. Quality lighting can prevent accidents.

Good lighting can also increase visual comfort. For example, today's free-form surface headlights can control the beam to illuminate road markings and verges for additional visual guidance. All European headlights with HID are equipped with headlamp cleaning and therefore able to show less glare than halogen headlights. In conjunction with dynamic headlamp levelling they produce a steady and reliable field of vision. The AFS-initiative sets out to increase visual performance for varying road or weather situation by taking into account the comfort requirements of driver and traffic.

Vehicle lighting provides for vision and visibility. A current trial evaluates the significance of marked contours on trucks and heavy-goods vehicles with respect to accidents. In the vehicle interior there is a noticeable move away from point light sources in dashboard or cockpit area. The driver is distracted less, when displays are back-lit, switches marked by their contour and feeling for space created by unobtrusive, ambient light.

DANGERS OF NIGHT DRIVING

Night-time accidents are more severe, fatal accidents twice as likely as in the day-time. In 1993 were counted in Germany 385.384 accidents with personal injuries. The victims were 515.450 people, of whom 125,854 suffered major injuries and 9,949 died [1]. One third (31,7 %) of the accidents occurred at night or during twilight hours. The proportion increases with the severity of injuries: 37 % of source: major injuries and half (49,6 %) of the deaths happen in conjunction with night-time accidents (see fig. 1).

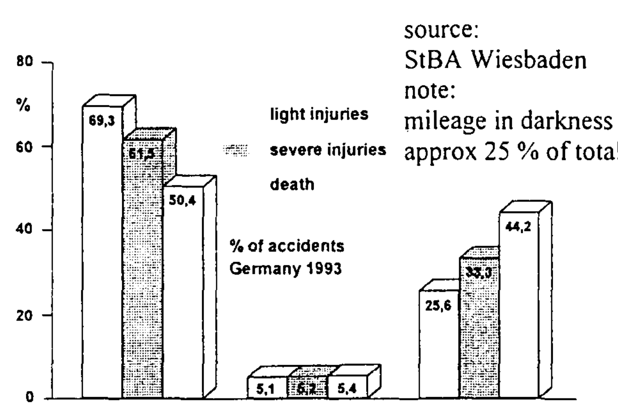


Figure 1. Road safety by night risks of accidents w.r.t light

Mileage driven at night is between 20 - 30 % of total mileage. Hence, the risk of accidents at night is more than twice as high as in the day-time [2].

The introduction of passive safety features such as crash barriers, seat restraints or airbag has reduced the overall number of severe accidents. It has not affected the relative risk of night-time accidents. Since vision provides over 90 % of the information relevant for safe driving [3], vehicle lighting yields the primary active safety device for night driving. Good vehicle lighting can reduce prime risks (see fig. 2) for night-time accidents and further help to improve safety during the day-time.

TOWN	COUNTRY	MOTORWAY
crossroads and junctions	reduced speed sensation	reduced speed sensation and
pedestrians in dark clothing	orientation in bends	false feeling of security
glare lights and dense traffic	differentiation between road and country (esp. when wet)	short seeing distance w.r.t. speed
	turning heavy - goods vehicles	exhaustion and sudden sleep slow vehicles
Note: Risks are raised throught impairment due to alcohol, inexperience and vision-deficiency (esp. with old age)		

Figure 2. Road safety by prime risks of areas

Vehicle lighting can

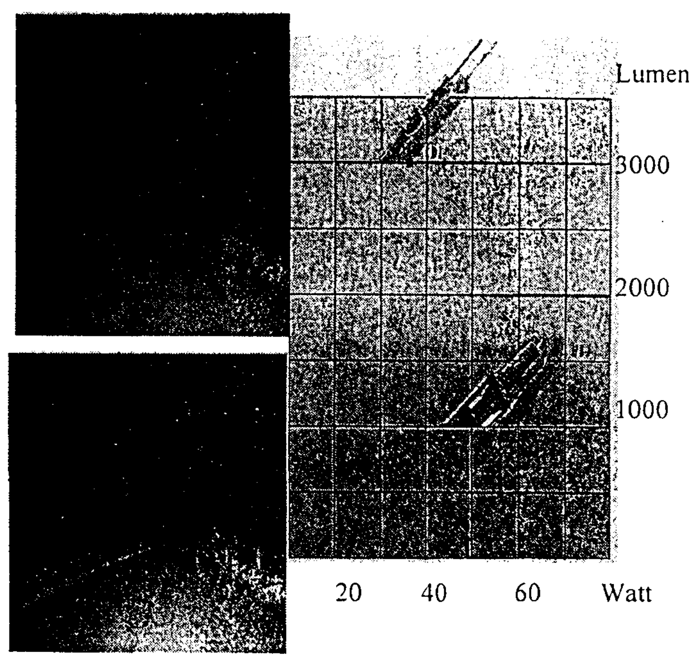
- provide better vision in front of the vehicle,
- create a less stressful work environment for the driver and
- signal information on driving state and intention.

Typical examples of vehicle lighting systems with the above properties are XENON lighting, intelligent front lighting systems and CELIS, which are described below.

XENON-FORWARD LIGHTING SYSTEM INCORPORATING HID

High Intensity Discharge (HID) light sources have been used widely. They complemented incandescent lighting in the early seventies in interior and sports lighting by lending the necessary brightness and colour reproduction for use with colour television. In the eighties first functional prototypes for automotive application were shown. Initially slow start-up and failing restrike when hot seemed to exclude the use of HID as a light source for forward lighting. HELLA, BOSCH, PHILIPS and OSRAM started the european research initiative, EUREKA-project # 273 "VEDILIS" to remove technical and regulatory obstacles. In 1990 HID passed the vehicle proving ground and the stringent safety rules of independent institutes and government bodies. In 1992 Germany approved HID as "sealed units", in 1996 the ECE type-approval was granted for a forward lighting system including HID light source, headlamp cleaning and levelling. HELLA delivers HID or "XENON" light systems, named after the traces of Xenon gas which provides instant light, in high volume for the Mercedes E-class and BMW 5-series.

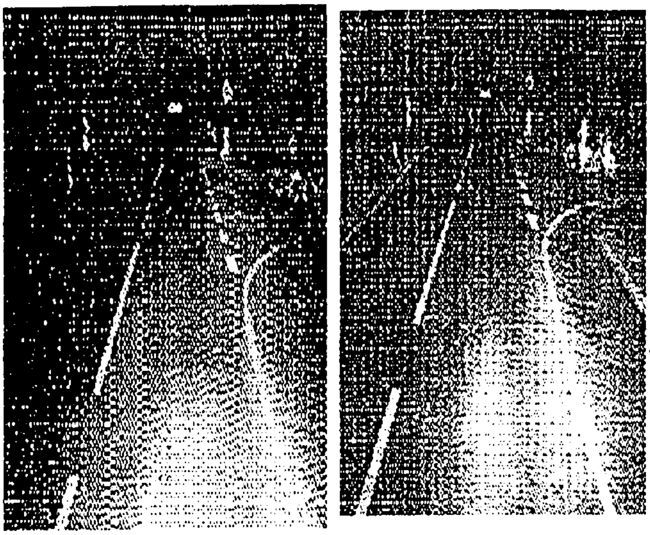
XENON-lighting is a safety enhancement. Nearly three times the useful light output and higher source luminance lends sufficient freedom to optical engineer and designer to combine headlamp appearence with optimal forefield intensity and light spread on the road. The beam pattern can spread wider and illuminate the area more uniformly (see fig. 3 beam pattern comparsion - XENON versus Halogen. See also fig. 4 road scene - HID and Halogen patterns).



Passing beam with halogen (top) HID (bottom)

Higher light output and lower wattage of D1 lamp compared with H1 lamp

Figure 3. XENON performance comparsion



HALOGEN H I D

Figure 4. Road Scene HID and Halogen Beam Patterns

Subtle beam design also takes care to:

- limit near-field illumination (in order not to attract the driver's attention towards a bright spot and in order to avoid reflective glare on wet roads),
- illuminate road sides and verges for better visual guidance and early recognition of dangers,
- provide extra reach with high light values below the cut-off,
- sculpture the cut-off in an effort to minimise glare to drivers ahead, pedestrians, cyclists and opposing traffic particularly in right bends.

XENON-lighting produces a bluish white light more akin to day-light than the reddish-white found in Halogen filaments. The XENON light source unlike the current bulbs do not wear since there is no filament to burn out. Ideally, its life should be equivalent to the life expectancy of the vehicle (2 000 hours is roughly equivalent to 100 000 miles).

The XENON lighting system for ECE comes with headlamp leveling. As incorrectly aimed headlamps are a source of discomfort and possible danger the EG directive 76/756 demands compensating measures.

Headlamp levelling limits disability and discomfort glare by correcting aim relative to vehicle inclination changes due to vehicle loading (see fig. 5).

Correct aim

Loaded: 3° misaim
- excessive glare to oncoming traffic
- reduced vision

Figure 5. Computer Simulation
Correct Headlamp Aim and Mis-Aim

HELLA provides dynamic headlamp levelling and thus limits further glare during acceleration and compensates shortening of the beam during braking (see fig. 6).

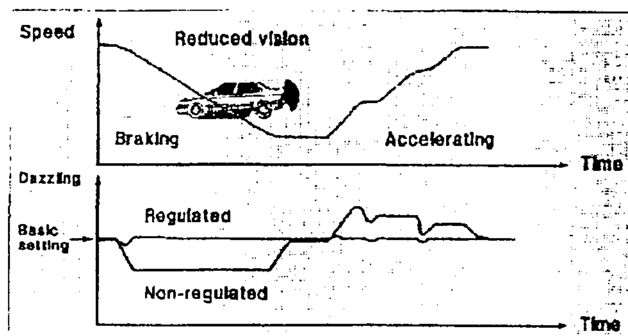


Figure 6. Lighting Condition Comparison Non-regulated vs. Dynamic Headlamp Leveling System

The XENON lighting system for ECE also includes headlamp cleaning. Dirty headlamps spoil good vision. Already a slight layer of dirt will reduce visibility distance to 50 % whilst doubling glare intensities (see fig. 7).

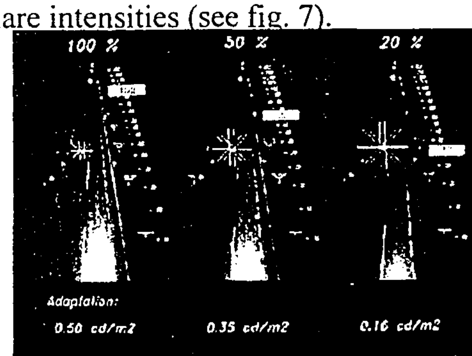


Figure 7. Dirty headlamps spoil good vision. Layers hardly visible reduce the visibility distance to 50 % whilst doubling glare intensities

INTELLIGENT FRONT LIGHTING SYSTEM WITH AFS

XENON lighting is the first frontlighting system always to include modern sensors and actuators. The beam pattern is still restricted to a light distribution which accompanied the technology of the H4 bulb in the sixties. Major lamp, set and vehicle manufacturers in Europe have now joined in a European research project, EUREKA # 1403 on Advanced Frontlighting Systems (AFS). The project examines the feasibility of adaptive front lighting. The work is driven by the rising requirement for safe and stressfree driving.

An intensive study of over 5000 km of road in Europe [4] relates "object scenes" to road geometry (see fig. 8).

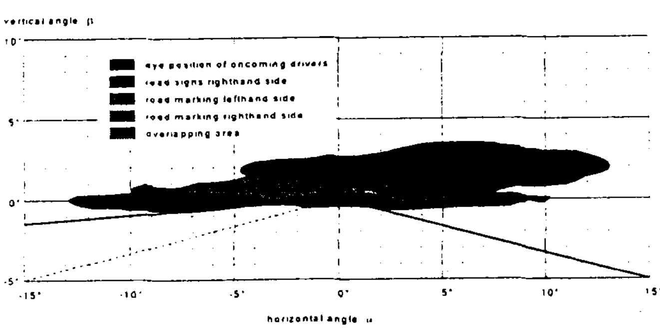


Figure 8. "object scenes" to road geometry

The study concludes with a recommendation to differentiate the lighting requirements for motorway, country roads and city roads and provides suggestions as to their prefered shapes. For example, country-road lighting in comparison with the ECE dipped beam (see fig. 9 and 10) would provide a cut-off with reduced illumination above the 15°-area - similar to what has been specified for XENON headlamps.

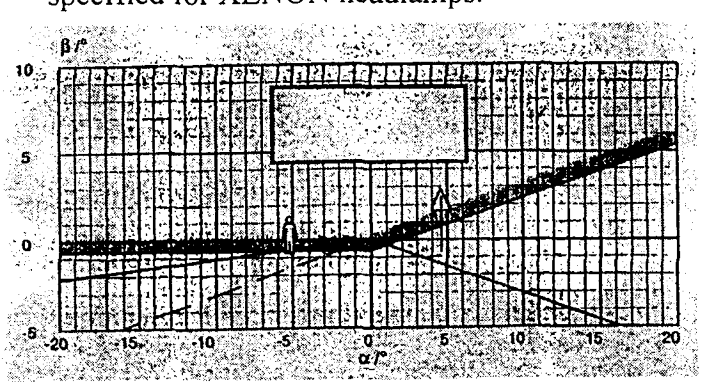


Figure 9. ECE-dipped beam

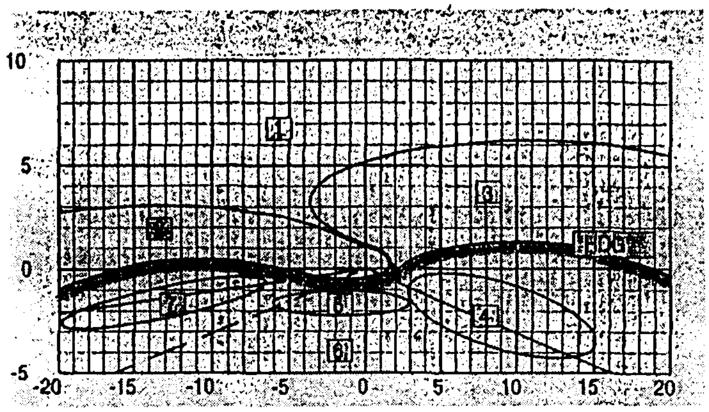


Figure 10. Country-road lighting

The illumination would be wider to accommodate bends and twisty roads. Dynamic headlamp leveling would permit the beams to provide further reach without the risk of extra glare.

Following the road study an extensive study of driver behaviour and expressed lighting requests is underway. First results conducted in Germany indicate a strong correlation between stress and accident risk. Driver stress for driving in the dark is due to general lighting conditions (twilight, night, tunnels), precipitation (rain, fog, snow), road infrastructure (such as motorway, country or town), roadsigns and special manoeuvers. Prime stress scenes and the driver's intuitive reaction are shown in fig. 11 for precipitation and fig. 12 for infrastructure.

RAIN	glare from raindrops spray from other vehicles water on windscreen glare from reflections	look for road markings follow tail lights sudden "blindness" stare forward
FOG	glare from backscatter visual attraction to lights	look for road markings rear fog light for visibility
SNOW	glare from backscatter glare from high fore-field light	look for markings squint

Figure 11. Forward lighting requirements stress due to precipitation (and darkness)

COUNTRY ROAD	hill tops dips bends widening/ narrowing junctions	sudden appearance of on-coming traffic fore-shortened light no light into bend uncertain location of edge vehicle not identifiable in side-view
MOTORWAY	spray glare hight speed	obscured vision sudden "blindness" stationary objects appear suddently
TONW	pedestrians cyclists road signs bright lights	invisible in dark clothing no lights particularly at junctions directions or road names ill lit coloured and obscuring lights

Figure 12. Forward lighting requirements stress due to road infrastructure

Technical solutions for stress and risk reductions through intelligent front lighting are evolving. Computer-aided lighting tools [5] now create tailormade light distributions using free-form surface technology (FF) as shown here for country-road lighting (see fig. 13).

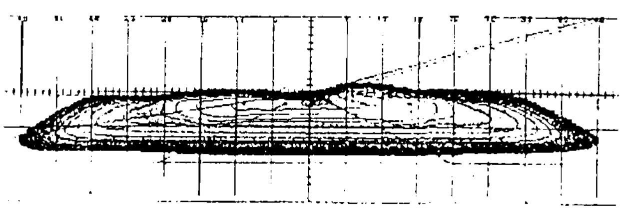


Figure 13. free-form surface technology (FF) as shown here for country-road lighting

Sensors and actuators can choose and direct light according to prevailing situations. By 2000 the technology will have been tried and tested for the new regulations on advanced front lighting.

CELIS-AMBIENT INTERIOR LIGHTING

Improved interior lighting can raise comfort and safety. Instrument lighting within the direct field of the driver's vision will become less distracting as lighting changes from point-marking to contours and lighting of areas. Very little light is necessary to mark objects such as switches, handles or pockets since luminance must be kept low for not interfering with the low adaptation level of outside view from the vehicle. When these are marked by their context (contour or area) they are more easily located and in addition generate a feeling of space.

Control tower or submarine lighting uses the colour red to orange in order to avoid reduction of night vision. Cold colours, such as blue, reduce dark adaptation. Warmer colours and a more direct lighting at low levels enhance the well-being as Kruithoff shows [6] (see fig. 14).

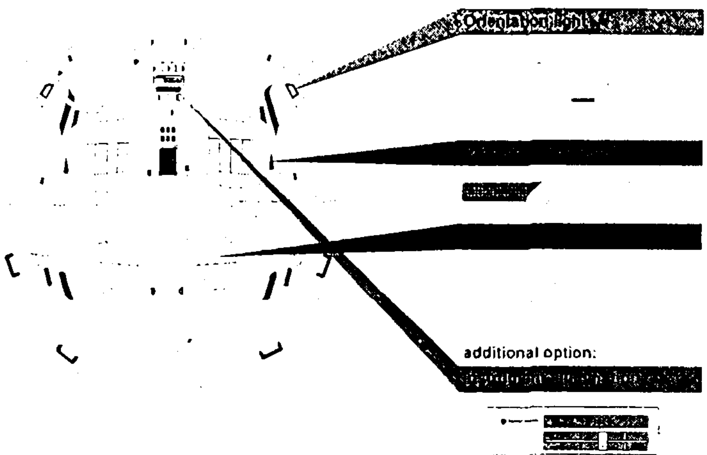


Figure 14. Definition of functiona l groups for the CELIS lighting concept

CELIS provides orientation lighting, lighting for entry/exit and workspace illumination with the new quality [7]. The light complements the functionality and surface quality of the vehicle interior with an ambience of light. HID is subtle enough not to be intrusive but sufficient to provide the visual context for intuitive orientation. Ongoing trials show that drivers value the ambience particularly on long night drivers where it relaxes and maintains their levels of concentration.

IMPROVED REAR-LIGHTING AND SIGNALLING

A current trial in Europe evaluates alternatives for improved conspicuity of heavy-goods vehicle at night [8]. Contour markings and area highlights on the vehicles prove to be more effective than point light-sources for immediate identification. The significant reductions in side and rear-end accidents involving marked heavy-goods vehicles has led to changes in the regulations. Similar effects can be expected for smaller vehicles.

Reliability, low-energy consumption and rapid response are performance features which apply also to signal lighting for vehicles. New light-sources such as LED or NEON provide service lives in excess of vehicle life, reduced power consumption of up to 70 % and shorter ramp-up time (see fig. 15).

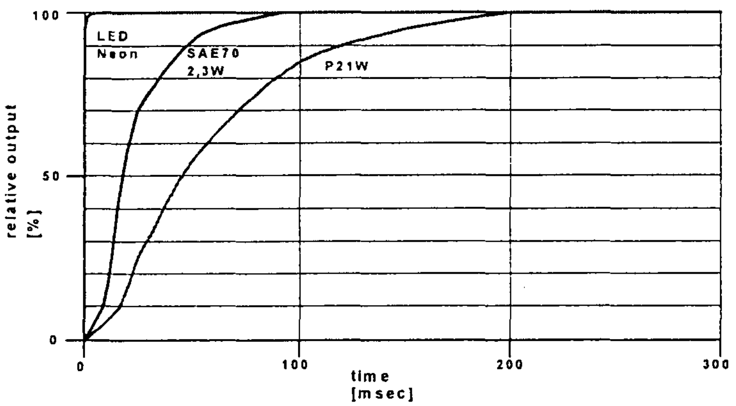


Figure 15. Output vs. Time
LED/Neon and Incandescent

For a brake light, the speed of LED or NEON is significantly better than incandescent and can shorten the breaking distance of following drivers by the equivalent of one vehicle length at 100 kmph [9].

New and improved signalling functions, such as warning signals with double flash [10], failure mode operation, reduced bulbs proliferation and improved position lights will further raise safety on our roads.

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MOTOR VEHICLE SAFETY and the ELECTROMGNETIC ENVIRONMENT: A REVIEW of their RELATIONSHIPS and CONSIDERATIONS

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Paper Number 96-S2-O-15

INTRODUCTION

This paper provides an overview of the history of motor vehicle electromagnetic compatibility, and underscores the need for cooperation between the motor vehicle industry and those who control the electromagnetic environment.

Background

Until near the turn of this century, the only electromagnetic radiation affecting man was natural radiation. This radiation came from various celestial sources as well as from the atmosphere. With Marconi's invention

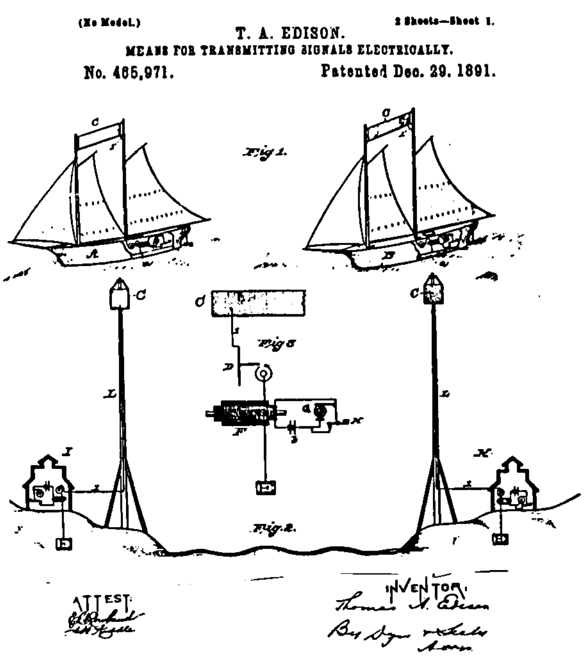

Figure 1. 1891 Thomas Edison radio Transmitter Patent

of the spark coil oscillator, man-made electromagnetic radiation became a part of the environment.

What does that have to do with us? Around this time, other individuals such as Daimler, Ford, and Maxwell were developing mechanical devices, called

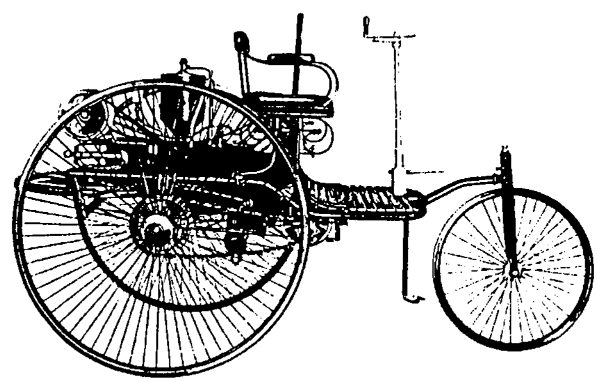

Figure 2. Typical early automobile with spark ignition engine.

quadracycles but sometimes only having three wheels, which transported people. Some of these machines were propelled with gasoline-powered engines. The gasoline internal combustion engine demanded the production of a sufficient spark to arc across the spark plug under high pressure. The ignition source (spark) for the engine was generated by a high voltage transformer and a set of mechanical contact points that opened some period of time after charging the transformer primary circuit. This opening of the mechanical contact points caused the high voltage side of the transformer to generate a high voltage that was carried by a wire to the spark plug, igniting the fuel/air mix in the cylinder. When the breaker points opened, a back current was also set up in the high voltage transformer (coil) that resulted in a spark, or several quick sparks, across the points. The coil and mechanical point breaker system used to generate the spark proved to be an excellent random generator of electromagnetic emissions. These secondary sparks were not wanted and were somewhat controlled with a capacitor across the points.

When the ignition points opened and the sparking occurred across the breaker points, the world was introduced to another new source of man-made electromagnetic radiation. However, when vehicles were first introduced, this was not a major problem because there were few vehicles and fewer intentionally transmitted signals.

Before the introduction of broadcast radios, the spark generated electromagnetic radiation did not bother anyone. Things changed rapidly around 1920, when the world was introduced to commercial broadcast radio. In 1923 automotive radios were introduced as a production option in the family automobile. This was also the first time that vehicle manufacturers had to consider electromagnetic radiation and immunity

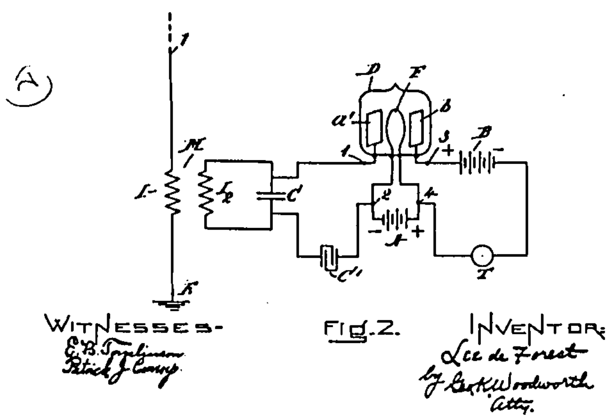

Figure 3. DeForest triode tube patent. Made radio possible in vehicles.

between the vehicle's spark ignition radiation and the vehicle's radio.

The local oscillators of broadcast radios in homes and in vehicles also generated radiation that contributed to the electromagnetic environment. But when radios were operated in the vehicles, or in houses adjacent to streets, there were two sources of man-made electromagnetic radiation, both being operated where there were people trying to listen to broadcast signals.. This raised the issue of electromagnetic compatibility. Unwelcome noise came from the radio when components on the vehicle ignition system such as the spark plug wires malfunctioned.

Science and industry brought more electricity into our lives as time passed. Electric power companies

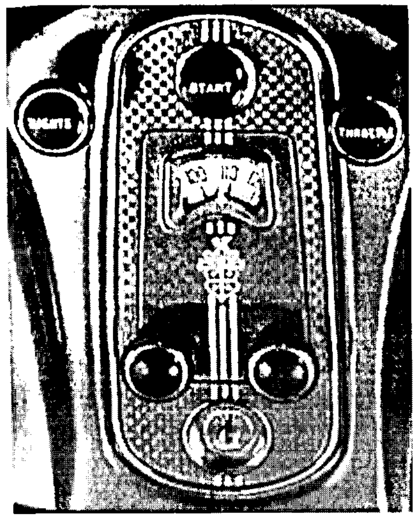

Figure 4. Early automobile entertainment radio installation.

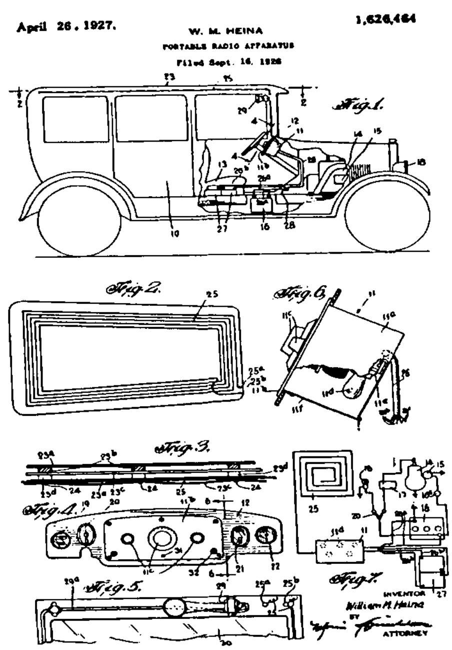

Figure 5. "Simplified" radio installation for 1928 automobile.

developed nation wide power grids. Electric lines and power substations were constructed and this wonderful marvel of modern society pervaded our lives. In 1926, Wabash, Indiana, in the heartland of the United States, was the first city in the United States to have electric power for homes, businesses and street lamps. With the advent of plentiful, inexpensive electric power in the following years, new devices such as electric arc welders, elevators, escalators, and electric motors around the home in appliances such as vacuum cleaners, blenders, etc. were in wide use. Other inventions such as the automatic traffic signal at intersections, railroad grade crossing signals and blinking neon advertising signs were more visible to the general public than items such as electric motors used for elevators, hidden away in a building structure.

One of the side effects of all the electric circuitry powering these devices is that they are not 100% efficient, and part of the energy that goes to power them is radiated into and contributes to the electromagnetic environment. Before the 1960's, the largest compatibility issue facing vehicle manufacturers and the general public was the effect of motor vehicle spark ignition radiation on broadcast radio and television reception. Out of the view of the general public was the compatibility of spark ignition engines with critical land mobile communications systems such as fire, police, and ambulance services. It is very important that the emergency responder (e.g. police, ambulance, fire department) hears the message and reaches the correct address in time. Their eception range could be limited by electromagnetic interference.

The emergency responder issue is a different concern today because many communications are sent to the responding unit by digital codes, and these digital codes are more immune to broadband interference (random spark ignition, motors, relays, switches) than analog modulation. However, with additional on-board microprocessors, there are now digital signals that have to be considered as potential sources of interference before vehicles are introduced into commerce.

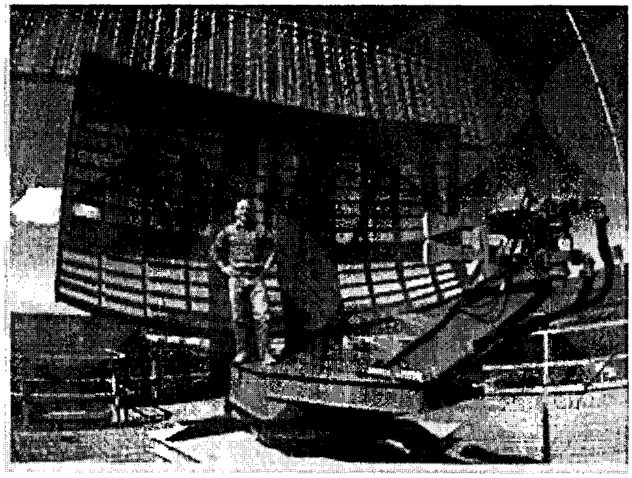

Figure 6. Typical radar transmit/receive antenna that contributes to the electromagnetic environment.

Today, the vehicle must operate and be compatible with electronic equipment unknown to the general public a scant 50 years ago. Radars, broadcast stations, home and business computers, radio frequency control devices, diathermy, ultrasonic welders, pagers, cellular phones, land mobile, amateur and citizen band radios, and FAX machines in vehicles are just a few of the devices that contribute to the electromagnetic environment.

American Automobile Manufacturer's Association (AAMA) members (Chrysler Corporation, Ford Motor Company, and General Motors Corporation) are researching and developing the technology needed to equip vehicles with electronics to control and to activate devices such as anti-lock braking systems, inflatable restraints, exhaust emissions control systems, and transmissions. For proper functioning, these devices need to be compatible with other electronic devices operating on or near public highways. As the number of electronic devices in use proliferates, it becomes important for the motor vehicle industry to have knowledge of the electromagnetic environment in which vehicle electronics will operate so vehicles can be compatible with the ever changing environment and the additional devices contributing to the electromagnetic environment.

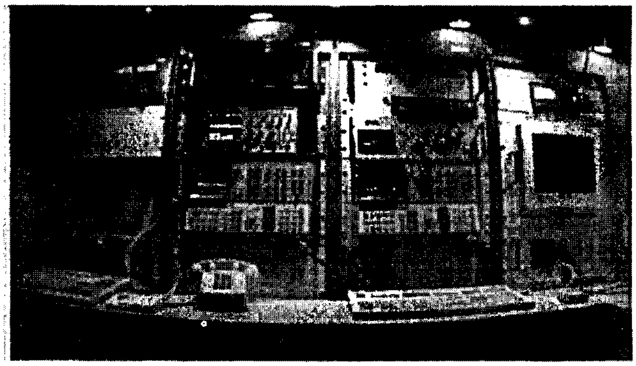

Figure 6. Electromagnetic environment measurement equipment (Courtesy National Telecommunications and Information Administration).

The motor vehicle industry has been involved with potential compatibility problems for many years. Test procedures have been developed to ensure vehicles will be compatible with the electromagnetic environment, and this has turned out to be no small effort. In the United States, the Federal Communications Commission (FCC) issues licenses for communications and broadcast stations to use specific frequencies and power levels. The FCC also specifies the amount of power, (harmonics, etc.) that can be emitted outside the spectrum band allocated for the device. The National Telecommunications and Information Administration (NTIA) allocates frequencies and specifies powers for intentionally radiating devices for government (including military) users.

Vehicle manufacturers must make a best effort to predict what the electromagnetic environment will be not only the date a vehicle is introduced, but also over the life of the vehicle may be used on public roads. About 10 years is the current average vehicle on-highway operating time after new model introduction for the United States, while some vehicles may be operated by their owners for 25 or more years. AAMA members continue to solicit information from the U.S. Federal agencies that allocate frequencies and specify power levels for intentionally radiating devices. The need to know what constitutes the electromagnetic environment is a continuing effort that vehicle manufacturers undertake to ensure vehicles will operate on all public motorways in the United States.[i ii iii]

Growth of Vehicle Electronics

Starting in the 1970's, integrated circuits became more available as a consumer item. Also about this time, motor vehicle manufacturers were acting to develop vehicles with greater fuel economy, lower exhaust emissions, better ride and handling qualities, and digital displays in place of traditional analog instruments. Computer controlled engine temperature devices such as cooling fans, electronic fuel injection for increased fuel economy, computer controlled ignition and exhaust recirculation for lowered emissions, computer controlled transmission shifting, electronic seats, lower lumbar back supports, automatic headlamp dimming, automatic headlamp activation are some of the functions that previously either were not continually monitored and controlled, or were controlled without electronic devices.

Electromagnetic Radiation in the Vehicle Operating Environment

Today motor vehicles encounter many products that emit electromagnetic radiation. A partial listing of some of the sources of radiation include:

- Radar
- Shortwave broadcast stations
- Microwave transmission systems
- Imaging devices in hospitals and doctors offices
- Microprocessor-based systems
- Burglar alarm systems
- Telephone Paging systems
- Cellular telephone systems in vehicles
- Loran navigation transmitters
- Omega navigation transmitters
- VOR aircraft navigation transmitters
- Electric welding devices
- Arc furnaces
- High voltage neon lights used for advertising
- Entertainment (AM/FM/TV) broadcast stations
- Military training facilities

These devices emit electromagnetic radiation as either a direct function of their operation or as a by-product of their operation. Vehicle manufacturers need to check vehicle electronic systems to ensure compatibility with these devices. This requires knowledge of some parameters of the device's radiation:

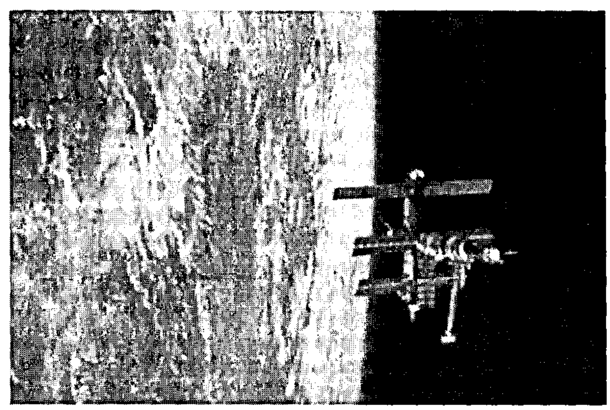

Figure 7. Spacecraft such as Muir and communications satellites contribute to the electromagnetic environment.

Is the signal direct or reflected?
Is the signal modulated? If so, what type of modulation?.
Is the signal horizontally, vertically, or elliptically polarized?
Is the signal transient or steady state?
Is the signal energy coupled directly or indirectly to the module?

As an example of the latter question, external electromagnetic waves may reach on-board electronic modules in two ways. That is instead of being directly **radiated** into the vehicle component, they may be **conducted** into the component through a secondary device on the vehicle. The radiation from an external source may be coupled into the vehicle through a wiring harness on the vehicle. A simple example of this type

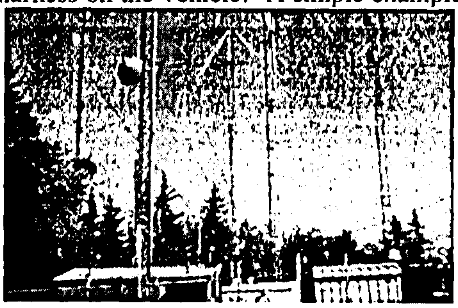

Figure 8. Typical transmitter site emitting high frequency radio energy into the environment.

of potential interference would be a signal received by the vehicle wiring harness at the door-fender slot, entering the power cable for the radio and then being propagated along the power cable to one of the other electronic devices on the vehicle such as the anti-lock brake module. That example is a simple one. Usually conducted radiation that is introduced into a vehicle is more difficult to anticipate. Radiation could be conducted into the vehicle through the stop lamps, for example. This hypothetical example could lead to interference when the vehicle was only driving away from the radiating source.

Vehicle manufacturers must ensure compatibility with the environment with the possibility of any or all combinations of electromagnetic radiation entering the vehicle in all the polarization and other conditions shown above. A number of tests have been developed and two types of tests will be discussed in this paper.

The general term is **compatibility**, which enables electronic equipment to function without degradation from electromagnetic sources (immunity) and without degrading the electromagnetic environment (emissions). It is up to the reader to refer to the specific tests listed below for details.

Vehicle Electromagnetic Compatibility Test

Emissions

Once the environment is known, then compatibility between the vehicle including its components, and the electromagnetic environment can be evaluated with appropriate tests.

One of the first tests developed and adopted was the Society of Automotive Engineers (SAE) J551 (Now referenced as J551/2), Performance Levels and Methods of Measurement of Electromagnetic Radiation from Vehicles and Devices (30 to 1000 MHz). The SAE J551/2 procedure is conceptually identical to the International electromagnetic radiation test developed by the International Special Committee on Radio Interference (C.I.S.P.R.). The ISO procedures are comparable also. The SAE test procedure adopts the vast majority of the C.I.S.P.R. Publication 12 requirements. Both the SAE and C.I.S.P.R. tests provide a method to measure broadband electromagnetic radiation and the control of radio interference. This type of test relates to the impact a spark ignition vehicle would have on radio reception equipment in buildings, such as that used for communications / entertainment systems.

The SAE/C.I.S.P.R. type test provides engineers with the impact a vehicle's broadband emissions have on communications/entertainment and not what impact the communications/entertainment systems have on the vehicle. This SAE/C.I.S.P.R. test has its basis starting in 1934 when radios were the primary form of entertainment and might be impacted by spark ignition radiation.

With the advent of microprocessor controlled functions in vehicles in the late 1970s and early 1980s the focus of electromagnetic compatibility specifications for vehicles had to be broadened. SAE J551/3 MAR 94 (the latest revision) was developed to provide limits and methods of measurement of vehicles for narrowband emissions from 10 kHz to 1000 MHz.

With the increasing number of on-board communications radios being placed on vehicles, SAE J551/4 Mar 94 was developed to provide limits and methods of measurement of narrowband and broadband interference received on an antenna on a vehicle.

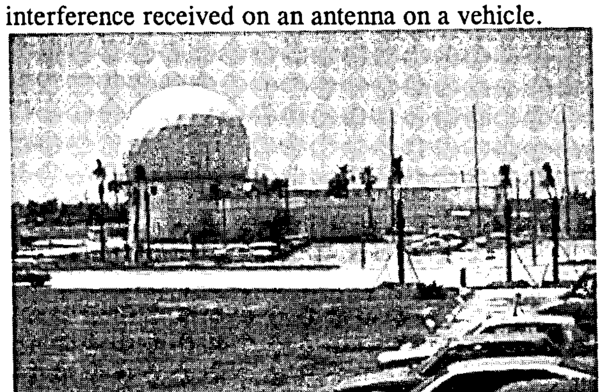

Figure 9. Typical radar station with vehicles operating in close proximity.

The SAE J551/2 / C.I.S.P.R. Publication 12 procedure includes frequencies from 30 to 1000 MHz and a new version of SAE J551/5 MAR 94 includes frequencies from 9 kHz to 1000 MHz. This procedure anticipates the future by including provisions for vehicle-mounted rectifiers used for charging in electric vehicles.

Immunity

In addition to emissions, the following vehicle immunity methods have been developed:

SAE J551/11 Vehicle Electromagnetic Immunity-Off-Vehicle Source

SAE J551/12 Vehicle Electromagnetic Immunity-On-board Transmitter Simulation

SAE J551/13 Vehicle Electromagnetic Immunity-Bulk Current Injection (BCI)

SAE J551/14 (Reserved) Vehicle Electromagnetic Immunity-Reverberation Chamber

SAE J551/15 Vehicle Electromagnetic Immunity-Electrostatic Discharge (ESD)

SAE J551/16 (Reserved) Vehicle Electromagnetic Immunity-Conducted Transients

SAE J551/17 (Reserved) Vehicle Electromagnetic Immunity-Magnetic Fields

Component Electromagnetic Compatibility Tests

Technology improvements in electronics resulted in increased mean time between failure, lower costs, and an associated increase in supply. This translated into more electronics for vehicle applications that were previously controlled by mechanical or analog methods. As electronics became used on vehicles for more applications such as transmission, exhaust emissions, and engine controls, a concurrent need developed to assess the immunity and emissions of these electronic modules with respect to other electronic sources, either on or off the vehicle.

The Society of Automotive Engineers Standard: "Electromagnetic Compatibility Measurement Procedures and Limits for Vehicle Components (Except Aircraft) -- SAE J1113 July 95 establishes the immunity and emissions levels of individual vehicle components. The following test protocols are provided in the Standard:

J1113/2 Conducted Immunity, 30 Hz to 250 kHz, Power Leads

J1113/3 Conducted Immunity, 250 kHz to 500 MHz, Direct Radio Frequency (RF) Power Injection

J1113/4 Conducted Immunity, Bulk Current Injection (BCI) Method

J1113/11 Immunity to Conducted Transients on Power Leads

J1113/12 Electrical Interference by Conduction and Coupling-Coupling Clamp

J1113/13 Immunity to Electrostatic Discharge

J1113/21 Road Vehicles-Electrical Disturbances by Narrowband Radiated Electromagnetic Energy-Component Test Methods-Absorber Lined Chamber

J1113/22 Immunity to Radiated Electromagnetic Fields From Power Lines

J1113/23 Immunity to Radiated Electromagnetic Fields-10 kHz to 200 MHz, Strip Line Method

J1113/24 Immunity to Radiated Electromagnetic Fields-10 kHz to 200 MHz, TEM Cell Method

J1113/25 Immunity to Radiated Electromagnetic Fields-10 kHz to 500 MHz, Tri-plate Line Method

J1113/26 Immunity to AC Power Line Electric Fields

J1113/27 Immunity to Radiated Electromagnetic Fields-Reverberation Chamber Method

J1113/41 Test Limits and Methods of Measurement of Radio Disturbance Characteristics from Vehicle Components and Modules, Narrowband, 150 kHz to 1000 MHz

J1113/42 Conducted Transient Emissions

Electromagnetic Environment and New Vehicle Technology -Intelligent Vehicle Highway Systems

The following list of concerns and suggestions are provided to help convey the need to continue expanding efforts concerning electromagnetic radiation coordination from the industry and government point of view. This continued coordination and harmonization, by necessity, has to include coordinated efforts by governments. Vehicles and equipment can be barred from crossing man-made international boundaries but electromagnetic radiation is exempt from man-made borders.

- Manufacturers of electronic devices and manufacturers of products that use these electronic devices must have a good understanding of what the **current** electromagnetic environment is for those environments in which the product is expected to be used.

- Manufacturers of electronic devices and manufacturers of products that use these electronic devices must have a good understanding of what the **future** electromagnetic environment is expected to be for those environments in which the product is expected to be used.

- Government agencies that allocate electromagnetic spectrum and specify maximum radiated power for electromagnetic devices need to be aware of the potential interactions for **current** products that are already operating in the environment.

- Government agencies that allocate electromagnetic spectrum and specify maximum radiated power for electromagnetic devices need to be aware of the potential interactions for **future** products that are expected to be introduced into the environment.

- Electromagnetic waves are invisible to political borders. It is critical that **international harmonization** of frequencies, power levels, and other operating characteristics take place.

- **International harmonization needs to involve all parties affected**. This includes electronic equipment manufacturers, vehicle manufacturers, users, governmental agencies, test development societies, and professional societies, at a minimum.

SUMMARY

By providing an overview of the history of motor vehicle electromagnetic compatibility, this paper underscores the need for cooperation between the motor vehicle industry and those who control desired emissions the electromagnetic environment.

It is the expectation of all those in the electromagnetic community that as more electronics are introduced into the environment, the previous years of harmonization effort can continue and global test procedures continue to result from this effort.

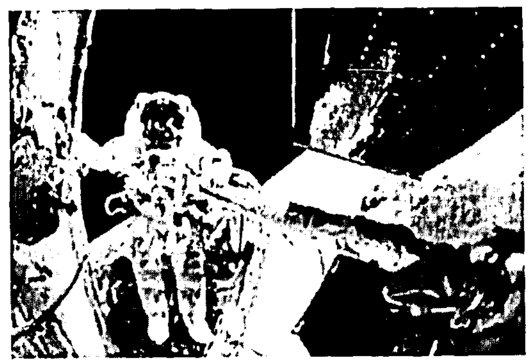

Figure 10. As technology progresses, the electromagnetic environment is expected to become more complex.

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FURTHER IMPROVEMENTS FOR MOTORCAR-HEADLIGHTNING SYSTEMS

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Paper Number 96-S2-O-17

INTRODUCTION

At this moment a world wide harmonised light distribution for low beam headlamps is discussed, based on requirements of the areas of SAE, JSAE, and ECE. At all discussions taking place today it comes out that not the improvement of the light distribution of low beam headlamps of the light distributions existing today is the aim but only the harmonisation.

The 90% Road Scenery

Up till today the basis of all discussions of the light distribution was the straight road with an eye height of the driver of h = 1.25 m and a mounting height of headlamps of h = 0.75 m. Real values of these heights are plotted in Figure 1 in form of frequency distributions (A: for the mounting height of the headlamp, B: for the height of the driver's eyes) measured under real traffic conditions.

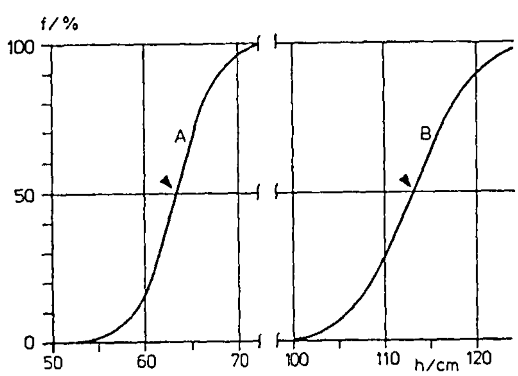

Figure 1: Frequency distribution
A: mounting height of headlamps
B: eye height of drivers

Taking the values for f=50% the Figure 2 can be plotted in a perspective way. As a sample the 90% area for the position of the driver's eyes in an on-coming car (1). The traffic signs (2) and (3) and the pedestrian are positioned at a distance of d = 50 m.

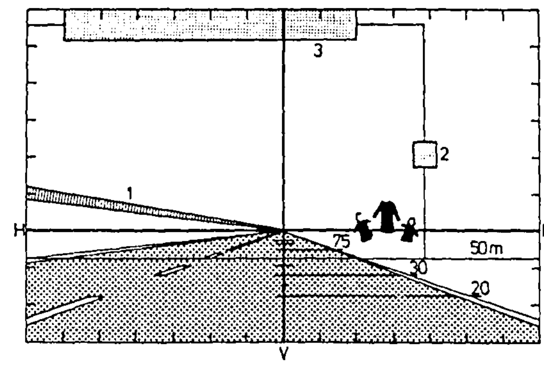

Figure 2: Perspective drawing of a two lane road
1: Eye height of oncoming cabdrivers
2: shoulder mounted sign
3: bridge sign
20..75: distance in front of the car
eye height: h = 0.63 m
mounting height of headlamps:
h = 1.13 m
grid: 1° x 1°
width of lane: b = 3.50 m

In addition to this „improved" geometry some of the parameters.of the 90% road scenery are listed in Table 1.

Table 1
90% Road Scenery

Geometry	Luminances	Reflection
mounting of headlamps	surrounding	pavement
	adaptation	condition of pavement
position of drivers	glare	
		pavement marking
course of the street	pavement	
		traffic signing
		pedestrians
		objects

This list of parameters is not compete, others may be added.

The Object Situation in the Road Scenery

In a large scale test in real traffic situation the positions of pedetrians in the road as seen by a car driver were investigated. The results are shown in Figure 3 in the perspective drawing of a street in the geometry conditions as explained before.
In the figure the 10%-, 50%- and 90%-areas are plotted (measuring height at the pedestrians

h = 0.60 ... 0.80 m) for a distance of recognition of d = 50 m. These results can be chosen as basis for minimum requirements of illumination close to the H-H-line.

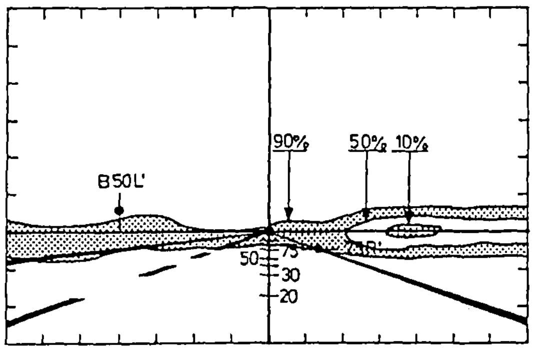

Figure 3: Probability areas of position of pedestrians in the street (recognition distance d = 50m, grid 1° x 1°, reference height h = 0.60...0.80m)

The Glare Situation in the Road Scenery

Similar to the results of Figure 3 in Figure 4 the positions of driver's eyes in an oncoming car are plotted. The meeting distance is d=50m. Comparing Figure 3 and Figure 4 requirements for a cut-off line can be derived.

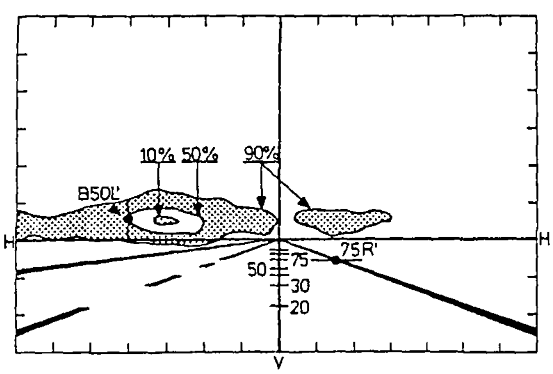

Figure 4: Probability areas of position of the drivers eyes in appearing cars (distance between cars d = 50 m, grid 1°x1°)

The Reflection Factor R' of Pavements

In a large scale experiment the reflection factor R' of pavements of different types of streets are plotted in Figure 5 as frequency distributions.
Taking a 50%- or 90%-value of R' requirements for the illumination below the H-H-line can be derived because the absolute luminance-values for comfort or minimum requirements are known.

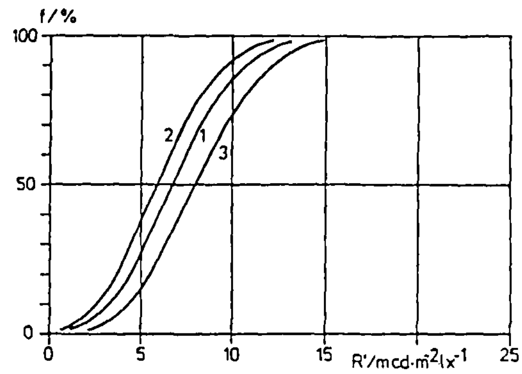

Figure 5: Frequency distribution of reflection factor R' of pavements
1: motorway
2: highway
3: rural road

The Contrast of Pavement-Markings

In different outdoor and indoor experiments the requirements on behalf of pavement-markings were investigated. Some results are shown in Figure 6 for the necessary optimum contrast k of a marking against the pavement under different road conditions, and pavement luminances L_U.

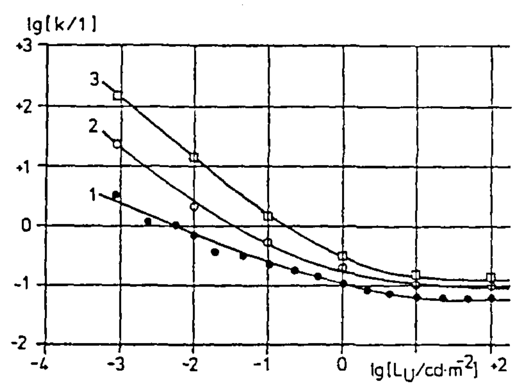

Figure 6: Necessary contrast k for pavement markings for optimal guidance
1: dry pavement
2: dry pavement, glare by oncoming cars
3: wet pavement, glare by oncoming cars

The Luminances of Traffic Signs

The minimum requirements for the luminance of traffic signs were investigated in indoor and outdoor experiments. The results are shown in Figure 7 for high and low ambient luminances.
The rating w=5 represents „optimum luminance L_Z". These kinds of results is the basis for the illumination requirement for low beam headlamps above the H-H-line.

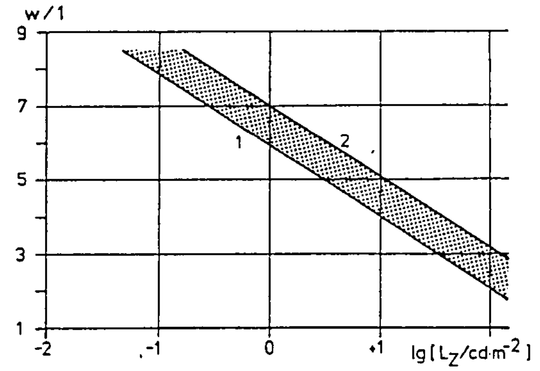

Figure 7: The rating w of luminances of traffic signs
1: low ambient luminance
2: high ambient luminance
w = 5: optimal luminance

The Illumination of Traffic Signs

The „luminance" of a traffic sign depends beside of others on the angles of direction of illumination and direction of observation.
Results of the position of right hand shoulder mounted traffic signs at a distance of 50m in front of a car driver are shown in Figure 8. As a sample the horizontal (S_H) and vertical (S_V) „lines of gravity" are plotted.
For the illumination-situation the results are plotted in Figure 9.
The results show the position of a mean headlamp (positioned in the middle of a car) on behalf of a right hand shoulder mounted sign. Again the „lines of gravity" are plotted.
The combination of the results of Figure 8 and Figure 9 makes it possible to give probability curves similar to those in Figure 3 and Figure 4 on behalf observation angle/illumination angle.
The „points of gravity" (cross points S_H/S_V) for three typical positioned traffic signs are shown in Figure 10 for observation distances between d=50m and d=250m.
These dates can be the basis for the minimum requirements for the illumination above the H-H-line.

Proposal for a New Necessary Screen

Based on the investigations about the „90% road-scenery" a new measuring screen as shown in Figure 11 can be derived.

The illumination requirements are listed up in Table 2.

In addition the requirements for the cut-off-line are the following:

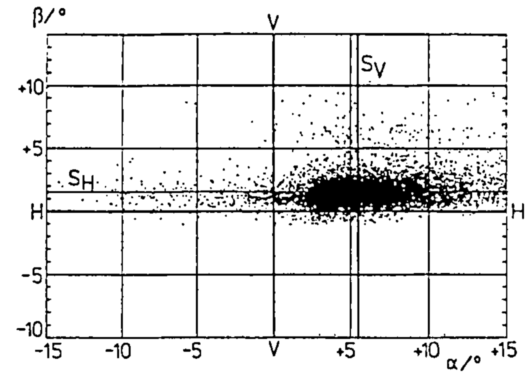

Figure 8: Distribution for right hand shoulder mounted traffic signs as seen by a driver at a distance of d = 50 m
S_V: $\alpha = +5.44°$
S_H: $\beta = +1.52°$

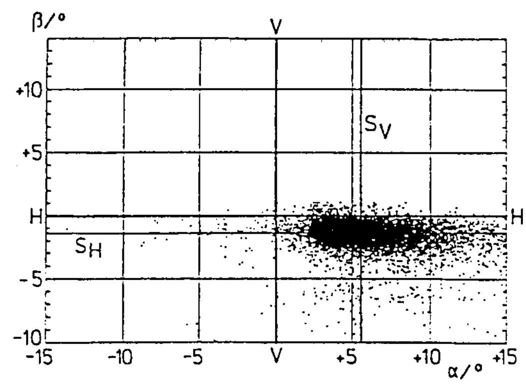

Figure 9: Distribution of the position of headlamps as seen from a shoulder mounted sign
S_V: $\alpha = +5.76°$
S_H: $\beta = -1.45°$

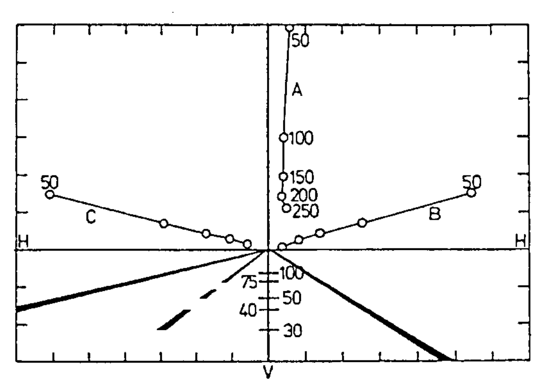

Figure 10: Points of gravity of the position of traffic signs in distances d = 50m ... 250m in front of a driver (grid 1°x1°)
A: bridge sign
B: right hand shoulder mounted sign
C: left hand shoulder mounted sign

Vertical position of the cut-off-line:

$$\left.\frac{d(\lg E)}{d\beta}\right/_{\max} \quad \text{or} \quad \left.\frac{d^2(\lg E)}{d\beta^2}\right/_{0}$$

Sharpness of the cut-off-line:

$$S = \frac{1}{\ln 10} \cdot \frac{1}{\Delta\beta} \cdot \frac{E_{n+1} - E_1}{0.5(E_{n+1} + E_1)}$$

Linearity of the cut-off-line:

$$\alpha = -1.5° \ldots -3.5° \ : \ \Delta\beta \leq 0.05°$$

Horizontal position of the cut-off-line:
a system of 3 single measurement points.

Adaptive Headlamp System

The kind of light distribution as described above is still a compromise. An optimal low beam headlamp system will consist of different headlamp units fulfilling special requirements as described in [1].

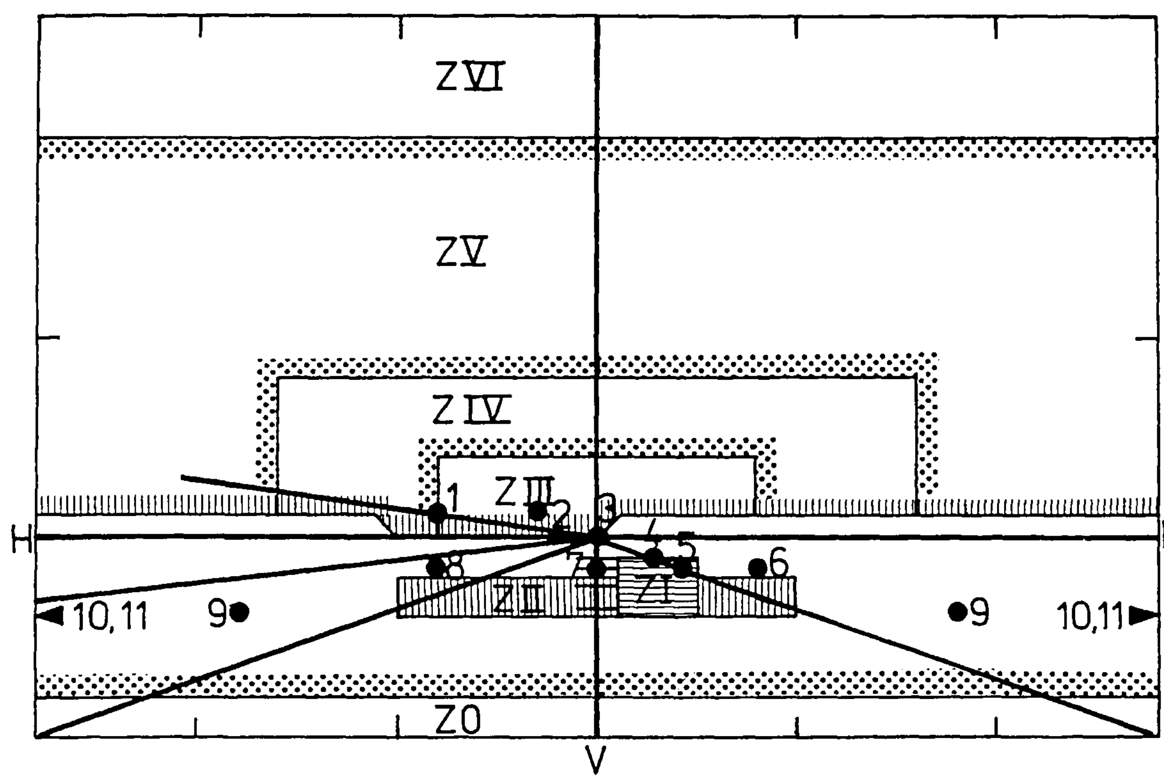

Figure 11: Proposal for a measuring screen for a low beam headlamp

Table 2 New measuring screen			
zone	**angles**		**illumination**
	$\alpha/°$	$\beta/°$	E/lx
0		$\leq$-4.0°	$\leq 30\% \ (E_{max})$
I	+0.5°...+2.5°	-0.5°...-2.0°	$E \geq 24lx$ and E_{max}
II	-5.0°...+5.0°	-1,0°...-2°	$E \geq 6.0lx$
III	-4.0°...+4.0°	above hatched line $\leq$ +2,0°	$0.1lx \leq E \leq 0.7lx$
IV	-8,0°...+8,0°	$\leq$ +4.0°	$0.1lx \leq E \leq 0.7lx$
V		$\leq$ +10.0°	$E \leq 0.7lx$
VI		> +10°	$E \leq 0.01lx$
measuring point 1(B50L)			$\leq 0.4lx$
2(B50L')			$\leq 0.4lx$
3(HV)			$\leq 0.7lx$
4(75R)			$\geq 18lx$
5(50R)			$\geq 18lx$
6(50r)			$\geq 8lx$
7(50V)			$\geq 8lx$
8(50l)			$6lx \leq E \leq 20lx$
9	-9.0°,+9.0°	-2.0°	$\geq 2.5lx$
10	-15.0°,+15.0°	-2.0°	$\geq 1.5lx$
11	-20.0°,+20.0°	-4.0°	$\geq 1.5lx$

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CURRENT NHTSA DROWSY DRIVER R&D

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Paper Number 96-S2-W-14

ABSTRACT

The NHTSA drowsy/fatigued driver research program focuses on the development of a vehicle-based driver drowsiness detection and warning system. NHTSA has developed a detailed concept of system operation and is supporting R&D on detection algorithm refinement, sensor development, and driver interface; i.e., advisory messages and alerting stimuli. A large system development, test, and evaluation project is fabricating a field-testable prototype for use in combination-unit trucks, and obtaining over-the-road data on system performance and useability.

In addition, ongoing Intelligent Transportation System (ITS) R&D relating to larger classes of crashes will also have the potential to reduce drowsy driver crashes. This includes ITS crash countermeasure performance specification programs on single vehicle roadway departure crashes and rear-end crashes, and the ITS concept of Automatic Collision Notification which will speed the emergency medical response to crashes in general.

NHTSA's research program also seeks to better assess the driver drowsiness/fatigue problem. Recent analyses of NHTSA crash databases have enhanced our understanding of crash characteristics and, in particular, have led to better estimates of crash problem size. Ultimately, however, direct observation of drivers using in-vehicle monitoring devices will provide the most valid data on driver drowsiness. NHTSA has developed, and is deploying, a sophisticated, unobtrusive instrumentation suite in a fleet of vehicles to obtain "real world" data on safety-related driver performance, behavior, and alertness.

Finally, the agency recognizes that technology is not the total solution to driver drowsiness/fatigue. NHTSA's program encompasses R&D on non-technological approaches – i.e., public information and education – to prevent these crashes. Ultimately, these two approaches will have synergistic and, perhaps, fundamental effects on the public's driving behavior and sleep hygiene.

INTRODUCTION

Most people think of "drowsiness" as an internal state of mind rather than something externally observable. Nevertheless, there is overwhelming evidence that drowsiness *is* observable, both in terms of the psychophysiological changes and the performance decrements that accompany it. Not only is the state of drowsiness *observable*, but there is now convincing evidence that various degrees of drowsiness are *measurable*. Observable, measurable changes in driver psychophysiology and performance may be the keys to preventing motor vehicle crashes associated with drowsiness and also to obtaining valid and reliable scientific data on the incidence and consequences of driving while drowsy. The fact that drowsiness is observable and measurable may also potentiate future efforts to inform and educate the driving public about the dangers of driving while fatigued. This paper addresses these crash prevention and crash analysis opportunities arising from the *in situ* observation and measurement of driver drowsiness. Note that the terms "drowsiness," "fatigue," and "drowsiness/fatigue" are used synonymously in this paper to describe the same state, or states, of reduced driver alertness and performance.

NHTSA DROWSY DRIVER R&D

In-Vehicle Drowsy Driver Detection/Warning

Loss of driver alertness is almost always preceded by a period of measurable performance decrements and associated psychophysiological signs[1,2]. Unfortunately, drivers themselves are often unaware of their deteriorating condition or, even when they are aware, are often motivated to keep driving[3]. As part of its Intelligent Transportation System (ITS) and driver ergonomics research programs, the NHTSA Office of Crash

Avoidance Research (OCAR) is supporting research to develop in-vehicle systems to continuously monitor drivers and their driving in order to provide a warning to drivers of their deteriorating alertness and performance[4,5]. Scientific support for the feasibility of this concept is provided by research showing that:

- Drowsiness can be detected with impressive accuracy using driving performance measures such as fluctuations in vehicle lateral lane position and "drift-and-jerk" steering.
- The use of direct, unobtrusive driver psychophysiological monitoring (e.g., of eye closure) could potentially enhance drowsiness detection significantly.
- Incipient drowsiness/fatigue is generally measurable well before the occurrence of episodes of involuntary sleep. The opportunity exists to intervene to advise/alert the driver several minutes or more before he or she "drops off".

In 1991, NHTSA initiated a cooperative agreement with Virginia Polytechnic Institute and State University (VPISU; also known as Virginia Tech) to develop a vehicle-based capability for unobtrusively monitoring driver performance. The scientific basis for this program was established by driving simulation studies performed in the late 1980's by Dr. Walter W. Wierwille and his associates at Virginia Tech. The envisioned system entails continuous measurements of driver performance, data processing to "decide" whether the driver is drowsy, and an appropriate system interface with the driver which might include both advisory messages and alerting stimuli[6]. Unlike most ITS crash avoidance systems, this would be a driver status warning system as opposed to a warning of a specific, imminent collision threat.

A successful system will require very high detection accuracy and, even more importantly, a low false alarm rate[7]. False alarm rate is critical because, for example, a hypothetical system with 100% accuracy of detection (drowsiness is always detected) and a 1% false alarm rate (i.e., warning sounds 1% of time when driver is not drowsy) would still yield more false alarms than "hits". For example, if drivers are actually drowsy 0.1% of all time driving, this hypothesized system would still have 10 false alarms for every "hit".

In current experiments at VPISU, sleep-deprived subjects are tested on a driving simulator; the test scenario is a desolate rural highway at night. Data from these experiments are used to refine multiple linear regression and other mathematical prediction models in which driving performance measures and their derivatives are the operational predictor measures. Examples of such driving performance measures/derivatives include standard deviation of lateral lane position and various measures of the proportion of time the steering wheel is held still, an indicator of "drift". During the monitoring, measures are computed for each six-minute time interval or epoch. They are then combined through multiple regression to predict "actual" drowsiness during that six-minute period. One sensitive and reliable definitional measure of "actual" drowsiness is the proportion of time that the driver's eyelids are closed 80% or more. This definitional measure has been named "PERCLOS". Predictor measures such as standard deviation of lane position are potentially obtainable in vehicles during actual driving, whereas definitional measures like PERCLOS need be obtained only in the research setting.

Excessive eye closure is regarded as a *prima facie* indication of driver impairment not only because of its relation to other indicators of drowsiness. It is estimated that drivers obtain 85 to 90 percent of the information necessary to drive via the visual channel[8]. One simply cannot drive safely unless visual information is processed continuously.

Multiple regression coefficients as high as +0.9 have been obtained between aggregated performance measures and actual drowsiness as measured by eyelid droop (PERCLOS). A way to translate this correction into statistics on classification accuracy is to consider three levels of alertness: alert, marginal, and drowsy. A "large error" would occur, for example, if the driver were actually alert (as determined by eye closure measures) but his or her aggregated performance measures indicated drowsiness. In these studies, a "large error" rate of only 2% was obtained. A higher percentage of "small errors" occurred – for example, the aggregated performance measurements indicated "marginal" when in fact the driver subject was drowsy. In these trials the "small error" rate was 17%. In 79% of the trials, the system was exactly on-target in terms of classifying aggregated performance into one of three levels of alertness (alert, marginal, or drowsy)[4].

Further "proof of concept" for in-vehicle driver drowsiness detection is provided by findings such as those shown in Figure 1. This figure shows results for one sleep-deprived subject on the Virginia Tech driving simulator. The solid line shows physiological drowsiness as measured by PERCLOS. The dots show aggregated driving performance as derived through multiple regression analysis using group data from 12 subjects (and cross-validated on 12 other subjects). Note the high correlation and the relative slowness of the trend toward drowsiness. High correlations between psychophysiological and behavioral deterioration imply that either, or both, processes could be used for detection. That is, deterioration in either domain indicates deterioration in both. The relative slowness of the trend implies that there is the opportunity to intervene well before the driver actually reaches a dangerously impaired state. Drowsy driver countermeasures have the potential to intervene many minutes before an imminent crash. This contrasts with most other ITS crash avoidance countermeasures, which will intervene just seconds, or even fractions of a second, before a crash.

As noted briefly above, the detection algorithm described above was developed using a group of 12 driver subjects on the simulator. Validation trials were performed to determine whether algorithms derived from the primary trials would transfer to a new set of driver-subjects. Results from these validation trials generally indicated virtually no loss in detection accuracy[4]. This implies that the algorithms are *generalizable*; i.e., algorithms developed using experimental subjects will be applicable to other drivers (i.e., the drivers actually using the countermeasure).

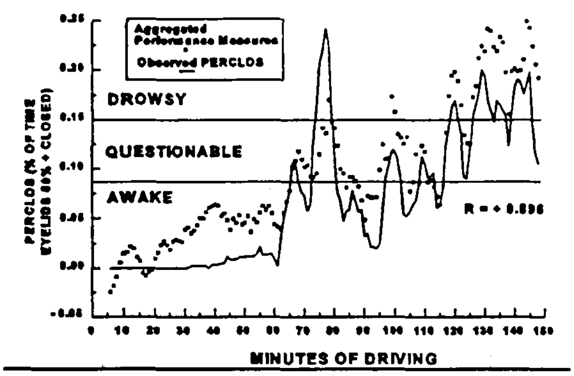

Figure 1. Proof of Concept.

Regarding psychophysiological driver measures, the R&D challenge is to develop unobtrusive or, "minimally-obtrusive" devices that drivers are willing and able to use regularly and which do not interfere with normal driving performance. Device cost is also critical due to cost benefit and marketability concerns. Through the DOT Small Business Innovation Research (SBIR), NHTSA is supporting an R&D effort to develop a device to directly measure eye closure. A device under development by MTI Research, Inc. (formerly MacLeod Technologies, Inc.) detects eyelid closure using opto-electronic techniques and miniaturized emitters and sensors. MTI's device can be mounted on the stem of eye glasses or a head set and is able to both detect and measure the duration of eye blinks[9]. Like PERCLOS, eye blink duration is thought to be a reliable measure of fatigue[10,11]. This eye monitoring approach is minimally obtrusive, employs established technologies, and has the potential to be very low-cost (e.g., less than $100). It has the potential to function as a self-contained unit usable while driving but also in many other operational environments. Currently-funded work is validating the capability of the device to measure drowsiness and is assessing its reliability, acceptability to drivers, and practicality for in-vehicle use.

Another approach to eye closure detection involves the use of a dashboard-mounted video camera and sophisticated image processing. This approach is completely unobtrusive, and could be adapted for applications other than drowsiness detection. For example, it could discern the driver's point of regard (i.e., where the driver is looking) and thus be used to monitor the driver's attention to the roadway ahead as well as his or her general level of alertness. PC-based prototype systems exist, although at present they may be too expensive for widespread commercial use. Extensive image processing is required to deal with problems such as driver head movements and the partial obstruction caused by eyeglasses. Nevertheless, a number of U.S. vendors are actively exploring and promoting this technology[12]. Commercial applications may come later as device cost decreases.

Thus, the envisioned vehicle-based driver drowsiness detection system would continuously and unobtrusively monitor driver performance and/or driver psychophysiological status (in particular eye closure). The most effective system would likely be one that measures *both* performance and psychophysiological status and combines the two types of information through sophisticated logic routines (yet to be developed) in making its assessment of driver status. Figure 2 illustrates a system schematic of this concept. The four major components are driving performance measures, driver psychophysiological measures (probably optional), a processing unit/decision algorithm (including a recording system), and the driver interface (advising/alerting system).

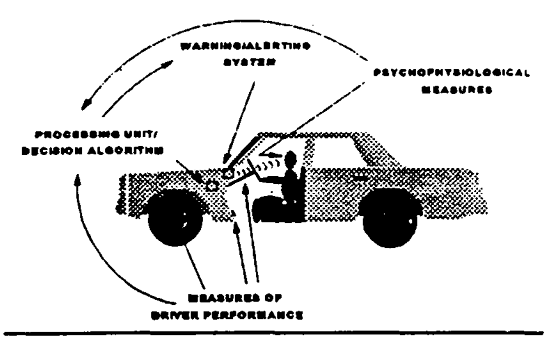

Figure 2. System Schematic.

Work at Virginia Tech is addressing the optimal driver interface (advising/alerting system) given a drowsiness episode. The envisioned system will continuously monitor driving and intervene with an advisory message (most likely recorded voice) to drivers when a clear trend toward drowsiness is detected. If the driver does not respond promptly to the advisory by some prescribed manual response, an immediate alarm may be sounded. In order to sustain the driver's alertness long enough for him or her to reach a rest stop, various alertness-maintaining stimuli may be used, including seat vibration, release of stimulating aromas, secondary verbal tasks (using recorded voice and voice recognition technology), and a "lane minder" – that is, an in-vehicle rumble strip function which emulates a real roadway rumble strip when the road edge marker is crossed. The lane minder system could actually function better than a roadway rumble strip since it could *anticipate* an imminent roadway departure based on vehicle trajectory and thus could provide an earlier warning[13]. Obviously, there are many critical requirements which must be met before such stimuli could be used. They must be proven to be effective, acceptable to drivers, and non-disruptive of the driving task. The current concept calls for drivers to select one or more of several possible alerting stimuli choices.

NHTSA-supported work is addressing the many important R&D challenges relevant to this countermeasure concept while fostering industry efforts to develop essential sensor components and field test various detection algorithms and advisory/alerting signals. A system development, test, and evaluation project now underway at Carnegie Mellon Research Institute is integrating current knowledge, fabricating a field-testable prototype for use in heavy trucks, and obtaining over-the-road data on system performance and useability. Widespread use of these systems will likely occur later this decade in long-haul trucks, followed by the general vehicle fleet, including non-commercial passenger vehicles, in the early 2000s.

To date, NHTSA R&D on driver drowsiness/fatigue has not considered alcohol as an interacting factor. Nevertheless, the agency is aware that the performance-degrading effects of drowsiness/fatigue and alcohol are mutually-potentiating. Future research may focus on this interaction, addressing, for example, the generalizibility of drowsiness/fatigue detection algorithms to alcohol-induced states of reduced driver performance. Through such research, the scope of target crashes addressable by this countermeasure concept may be greatly expanded.

Related ITS Initiatives to Reduce Driver Drowsiness/Fatigue Crashes

In addition to the drowsy driver R&D program *per se*, two NHTSA ITS crash avoidance initiatives will help prevent driver drowsiness/fatigue crashes along with other broader categories of crashes: single vehicle roadway departure crashes and rear-end crashes. A third NHTSA ITS program will reduce the post-crash injury consequences of all crash types, but with particular relevance to drowsy driver crashes.

Eighty (80) percent of drowsy driver crashes are single-vehicle roadway departures or collision with parked vehicles[14]. One countermeasure concept is to prevent the road departure event itself, regardless of driver status. A major ITS program at NHTSA is determining optimal in-vehicle device designs for preventing road departures[13,15]. The envisioned countermeasure would actually be a combination of two systems: a "lateral" system designed to detect and intervene when dangerous lateral trajectories of the vehicle are detected, and a "longitudinal" system designed to detect and intervene when the vehicle is traveling too fast for an upcoming roadway segment (e.g., a curve). The functioning of both systems could be modulated by information on roadway condition (e.g., coefficient of friction).

About 4,000 police-reported crashes annually are rear-end crashes with the police accident report (PAR)-cited principal causal factor of drowsiness/fatigue of the driver of the striking vehicle[14]. This represents about seven percent of PAR-cited drowsy driver crashes but only about 0.3 percent of police-reported rear-end crashes. The ITS headway detection/forward obstacle detection concept will address these crashes[16]. These devices will warn drivers, and/or initiate low-level automatic braking, when their vehicle is closing too fast on the vehicle ahead. Although drowsiness/fatigue is not known to play a major role in rear-end crashes, driver inattention *is* known to play a major role. Indeed, driver inattention in its various forms is the principal cause of rear-end crashes[17]. It is possible, though not proven, that many of the transient attentional lapses which cause rear-end crashes are related to drowsiness/fatigue. To the extent that drowsiness is involved in rear-end crashes, headway detection/forward obstacle detection systems may be regarded as driver drowsiness countermeasures. Analytical studies show these devices to be among most promising of the ITS crash avoidance concepts[17,18].

Once a crash has occurred, crash *avoidance* technologies are no longer applicable. However, ITS technology can be employed to greatly expedite and ensure that emergency help will be on the way immediately. An Automatic Collision Notification (ACN) system employs a crash sensor, a communication system, and a geo location system to provide an immediate signal to emergency medical services containing information on crash location and severity[19,20]. ACN is currently undergoing a NHTSA-sponsored operational test. ACN systems are applicable to all crashes but will probably have the greatest injury-reduction effect for rural single-vehicle, single-occupant crashes – a classic drowsy driver scenario.

Crash Problem Size Assessment Research

NHTSA tries to keep a broad perspective in understanding the causes of crashes. Broad-based studies of crash causation – i.e., those that consider a broad sample of crashes and all possible causes – have included the classic Indiana Tri-Level Study[21], a recent NHTSA study[18], and a recent General Motors study[22]. These studies have indicated that "asleep-at-the-wheel" is an important but not a *leading* cause of crashes. Leading causes of crashes and fatal crashes include driver inattention (distraction or attentional lapses), improper lookout ("looked but didn't see"), excessive speed, alcohol intoxication, misjudgement of traffic gaps/velocities, violations of signs and signals, obstructed vision (e.g., another vehicle occluding view of traffic), evasive maneuvers resulting in loss-of-control, and loss-of-control on slippery roads. Several major caveats are applicable to these findings, however. First, the identification of drowsiness/fatigue as a crash factor is especially difficult[14]. Second, drowsiness/fatigue may increase the likelihood of many kinds of mental errors including both recognition and decision errors (further discussion below). Third, it appears that the relative role and importance of drowsiness/fatigue as a safety factor differs greatly between commercial and non-commercial (private) drivers (as addressed in the next section).

NHTSA estimates that about 100,000 police-reported U.S. crashes annually, about 1.5 percent of crashes, have drowsiness/fatigue as a principal causal factor. This is based on a recent review of 1994 National Accident Sampling System (NASS) General Estimates System (GES) cases[23] and data from the 1982-84 NASS. Both the recent NASS GES and the 1982-84 NASS samples were obtained from numerous sampling locations nationwide which were selected to be nationally-representative. The 1982-84 NASS investigations were "medium depth", in that they included Police Accident Report (PAR) reviews, vehicle inspections, scene inspections, and driver interviews. The findings of other broad-based studies, including the three cited in the previous paragraph and preliminary data from the 1995 NASS Crashworthiness Data System, have been generally consistent with these findings. Non-broad-based studies, such as studies involving high risk roadway types (e.g., Interstate highways and turnpikes) have generally provided larger percentage estimates.

We know less about the role of fatigue in *fatal* crashes than in police-reported crashes in general, due largely to the fact that often the driver himself or herself is the fatally injured party and thus cannot be interviewed. Data from the 1989-93 Fatal Accident Reporting System (FARS) indicate that drowsiness/fatigue was cited as a factor in an annual average of 1,357 fatal crashes resulting in 1,544 fatalities. This represents approximately 3.6 percent of all fatal crashes and also 3.6 percent of fatalities during those five years[14]. These estimates should be regarded as conservative.

We need more data – especially in-depth data – on the prevalence of drowsiness/fatigue as a principal causal factor in crashes – i.e., "asleep-at-the-wheel" crashes. However, NHTSA believes that the most important question relating to the role of drowsiness/fatigue in crash causation is, *How many of the huge number of crashes caused by attentional lapses are related to fatigue?* Roughly one million crashes annually – one-sixth of all crashes – are caused by driver attentional lapses. For example, as previously noted, studies of rear-end crashes show that most rear-end crashes are causes by such lapses. Most driver attentional lapses are attributed to either distraction (inside or outside of the vehicle) or "daydreaming" (i.e., competing thoughts). But drowsiness/fatigue may play a role that is not discernible to crash investigators or even to drivers themselves.

To answer this question, NHTSA plans to use sophisticated, unobtrusive vehicle instrumentation suites to obtain *in situ* data on safety-related driver performance and behavior. The agency has designed and fabricated a prototype portable Data Acquisition System for Crash Avoidance Research (DASCAR) which employs miniaturized videos (of the driver and the roadway) and multiple measures of driving performance[24]. Psychophysiological monitoring devices, if unobtrusive, may also be employed. DASCAR-based studies may not only provide direct empirical data on "asleep-at-the-wheel", but may also provide data on the role of drowsiness/fatigue in the huge population of crashes that involve recognition failure and other mental errors. At this writing, the agency is beginning the fabrication of two more full-system DASCAR suites and an additional number (to be determined) of partial suites. Initial studies will focus on gathering baseline data on normal driving, including data on driver alertness and attention. Later studies will determine the driver attentional correlates of performance-failure events, such as the longitudinal encroachment of the test vehicle to vehicles ahead in the same travel lane (i.e., a rear-end crash "near miss"). Such performance failure events would be identified from a headway detection sensor. Video recordings and other data would be used to classify the accompanying driver state (e.g., drowsy, distracted, apparently daydreaming). DASCAR studies are not likely to capture significant numbers of crashes, but they will be capable of capturing sufficient numbers of the kinds of driving performance errors known to cause crashes.

Monetary Indices of Crash Problem Size

Table 1 provides a monetary perspective on the U.S. drowsy driver crash problem, with particular attention to vehicle type differences. These statistics were developed using crash problem assessment algorithms described by Wang, Knipling, and Blincoe (1996)[25] and monetary value metrics derived by Blincoe (1996)[26]. Two levels of monetary value of crashes can be derived:

- "Economic" values are based on narrow economic loss criteria -- i.e., what is the cost of the crash in terms of actual monetary loss including medical care, legal services, vehicle repair/replacement and, significantly, lost productivity?
- "Comprehensive" values represent a higher level of monetary valuation, since they incorporate both economic losses (as described above) *and* a valuation of less tangible human consequences such as "pain and suffering" and loss of life. In other words, comprehensive value includes not only the monetary value *of* crash consequences but also the additional monetary value society *places upon* crash consequences such as loss of life or disability.

Table 1 shows the above two levels of monetary value for five different monetary statistical metrics for four vehicle type categories: all vehicle types (combined), passenger cars, combination-unit trucks, and single-unit trucks. The five statistical metrics are total annual U.S. monetary cost, per-police-reported crash cost, per 100 million vehicle miles of travel (VMT), per registered vehicle annual cost, and per vehicle produced cost over a full operational life[25]. The crash statistics forming the basis of these economic estimates were obtained from the NHTSA General Estimates System (1989-93 average), but with a 50% correction for missed drowsy/fatigue driver cases which was based on the findings of Knipling and Wang (1995)[23]. In other words, it was assumed that the "real" number of drowsy/fatigue driver crashes was 50% greater than GES statistics indicate. This is considered a reasonable working assumption based on available research. Of course, a valuation for the role of drowsiness/fatigue in driver attentional lapses is not included in this monetary analysis since the extent of this role is unknown.

Table 1.
Monetary Estimates of the U.S. Drowsy/Fatigued Driver Crash Problem for Four Vehicle Type Categories (derived from 1989-93 GES)

Vehicle Type Category: / Monetary Statistical Metric:		All Vehicles	Pass-Cars	Combo-Unit Trucks	Single-Unit Trucks
Total annual U.S. monetary cost*	E C	$3.8B $12.5B	$2.3B $7.5B	$280M $765M	$32M $21M
Per-police-reported crash cost	E C	$34K $120K	$28K $95K	$87K $234K	$48K $127K
Crash cost per 100 M vehicle miles of travel (VMT)*	E C	$169K $570K	$156K $510K	$286K $779K	$59K $159K
Crash cost per registered vehicle annually*	E C	$20 $68	$20 $60	$170 $470	$10 $20
Crash costs per vehicle produced over a full operational life*	E C	$220 $730	$180 $580	$2,060 $5,600	$90 $240

* Inflated by 50% for presumed undercounting in GES per the narrative discussion; E: "Economic" value of crash consequences; C: "Comprehensive" value; M: Million.

Table 1 shows dramatically the high monetary cost of the U.S. drowsy driver crash problem. Since these crashes are often severe, the monetary consequences are high -- $34,000 in economic losses per police-reported crash ($120,000 per crash when comprehensive costs, including pain and suffering, are tabulated). For combination-unit trucks, these per-crash costs are even greater: $87,000 (E) and $234,000 (C). Note in addition that, even though combination-unit trucks represent a small portion of the overall national picture ($280M of $3.8B, or 7%), their per-vehicle costs, both annually and over a full operational life, are many times that of other vehicles. The high per-vehicle-produced monetary costs for combination-unit trucks ($2,060 [E] and $5,600 [C]) mean that these vehicles are by far the most promising platforms for cost-effective applications of vehicle-based drowsiness countermeasures. A positive cost-benefit picture will be much easier to achieve for combination-unit trucks than for any other vehicle type, even though prevention of their drowsiness/fatigue-related crashes will not significantly alter the overall U.S. crash picture.

Table 1 shows single-unit trucks to be a relatively unpromising platform for the application of drowsiness countermeasures. They represent a paltry percentage of the national picture (less than 1%) and their per-vehicle-produced crash costs are less than half that of the vehicle fleet in general and less than 1/20 of combination-unit trucks. These statistics likely reflect the relatively small mileage exposure and local/short haul operational usage of these trucks.

Public Information and Education

In addition to its technology and analytical R&D programs, NHTSA is responding to a U.S. Congressional directive to develop and evaluate a drowsy driving public information & education program. The planned elements of this new $1 Million/year program include the following:

- Analyze the role of fatigue, sleep disorders, and inattention in highway crashes. This work will be accomplished in cooperation with the National Center on Sleep Disorders Research of the National Institutes of Health.
- Investigate instances of fatigue-related events in motor vehicle operation. This will entail DASCAR direct observational studies as described above.
- Develop and test educational countermeasures for fatigue-related highway crashes. This activity will include specification of target populations, determination of message themes and motivational approaches, and development of dissemination strategies.
- Develop strategy and lay foundation for public information campaign. This will include the development and testing of draft materials as well as assessment/refinement of the overall strategy.
- Formally evaluate the information/education program, including the collection of pre- and post-campaign data.
- Broader-scale implementation of validated information/education program.

CONCLUSION

An intriguing aspect of this research is the many potential *synergies* which arise. One synergy is that between performance and psychophysiological measures of alertness. Researchers have compiled data showing both approaches can be reliable, sensitive, and robust. When used together, it appears that extremely accurate, and predictive, measures of driver alertness will be possible.

Another synergy relates to the role of DASCAR in the NHTSA drowsy driver research program. DASCAR research may be the only scientifically-credible way to answer the most far-reaching question about the drowsy driver crash problem size – the role of drowsiness/fatigue in the huge number of crashes caused by driver attentional lapses. In addition, DASCAR technology is being employed to create an in-vehicle testbed for drowsiness countermeasures. This will include short-term test and evaluation of devices as well as long-term studies to answer such questions as whether drivers will habituate to alertness-maintaining stimuli or, more fundamentally, whether drivers will use drowsy driver detection devices in a responsible manner – that is, to decrease their driving during drowsiness as opposed to increasing or sustaining such driving.

Perhaps the ultimate synergy will be that between technology and public awareness. Public education efforts by many organizations over the past few years have sensitized the public to the fundamental need for sleep and to the effects of sleep deprivation on human performance and wellness. NHTSA's new public information and education initiative on driver drowsiness will add to this public awareness. The agency believes also that the introduction of reliable and accurate driver performance monitors will result in a quantum increase in users' awareness of their own levels of alertness and performance, and of sleep hygiene practices which affect alertness/performance -- most obviously, obtaining sufficient sleep prior to driving. Driver performance monitors will provide continuous, quantitative, and anticipatory feedback to drivers. It is hoped that, over the long term, they will use this feedback to refine their driving and even their lifestyles.

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Technical Session 3

Improved Frontal Protection (Offset) and Advanced Occupant Protection Systems
Chairperson: Maryvonne Dejeammes, France

AUSTRALIAN RESEARCH IN DEVELOPING THE OFFSET FRONTAL DEFORMABLE BARRIER TEST PROCEDURE

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Paper Number 96-S3-O-02

ABSTRACT

Frontal crashes are the cause of the majority of deaths and injuries on the roads. In 1995, the Federal Office of Road Safety (FORS) introduced Australian Design Rule (ADR) 69 for full frontal crash protection which has seen the majority of passenger cars fitted with at least driver's side airbags. When coupled with Australia's high seat belt wearing rate of over 95% in the front seats, significant reductions in road trauma are expected.

FORS has participated in the work of the European Experimental Vehicle Committee (EEVC) to develop a globally harmonised test procedure for offset frontal crash testing. Other participants include the USA, Canada and Japan. This paper summarises the outcomes of Australia's offset frontal crash test program which have been provided to EEVC Working Group 11.

Australia's aim is to have a set of frontal crash standards which will result in vehicle designs that protect occupants both in high deceleration head-on crashes as well as "softer" offset crashes which usually result in intrusion based injuries. While serious lower limb injuries are rarely life threatening, they usually result in extremely high societal costs associated with life-long debilitation.

INTRODUCTION

The Australian Design Rules (ADRs)[1] set down a comprehensive range of performance and design requirements for motor vehicle safety and are among the most stringent in the world. The ADRs are administered under a type approval system by the Federal Office of Road Safety.

Since the first set of ADRs were implemented in 1969, there have been significant reductions in fatalities through the ADRs and other Australian Government initiatives such as compulsory seat belt wearing and drink driving campaigns. This saw the 1992 fatality figure fall to half that of 1970.

Accident statistics show that frontal crashes are the cause of the majority of deaths and injuries on the roads.

A $1 million standards development program[2][3] begun in 1989 by the Federal Office of Road Safety (FORS) led to the introduction of ADR 69 for full frontal crash protection which sets head, chest and leg injury criteria. All new passenger cars will be required to comply with it by 1996 and vehicle manufacturers have indicated that the majority of passenger cars will be fitted with at least driver's side airbags to demonstrate compliance.

OFFSET CRASH TEST PROGRAM

After head-on crashes, the next most prevalent type of frontal accident are offset crashes where only part of the vehicle's front structure absorbs the impact. There are currently no regulations anywhere in the world for this type of crash situation.

This was the reason the Federal Office of Road Safety decided to participate in the work of the European Experimental Vehicle Committee (EEVC) to develop a globally harmonised test procedure for offset frontal crash testing. Australia, Japan, Canada and the USA are participating together with most European countries.

The FORS test program aimed at addressing the following issues:

- Base research to assist WG 11 in determining the best offset, test speed and barrier design to incorporate into the test procedure.
- Examine the effects of drivetrain asymmetry on test outcome.
- Further research on barrier face design as a result of initial test series.

The dummy data for the head, neck, chest and legs are summarised in the Appendix at the end of this paper.

The outcome of this work has been presented for consideration by EEVC Working Group 11.

WHY AN OFFSET DEFORMABLE BARRIER FACE

When vehicles have a head-on crash, engagement of the front structures of the impacting cars causes high initial decelerations which start the car's crumple zones collapsing. This is replicated in a regulatory test of the car into a rigid barrier.

In an offset test, these high initial decelerations do not always occur (until a stiff structure such as the engine/drivetrain is engaged). Without these high initial decelerations the car's stiff crumple zones may not start collapsing but rather transfer the crash energy into the passenger compartment. A deformable barrier face was chosen as the means of replicating this by limiting these high initial decelerations.

EEVC BASE RESEARCH

Many popular passenger cars are now designed with a transverse front engined, front wheel drive configuration. The initial WG 11 work was predominantly on European left hand drive (LHD) vehicles with the small and medium test cars having this configuration with the gearbox on the left hand side. The large car used had a longitudinal front engine, rear wheel drive configuration.

The initial research conducted by the EEVC used a 50 psi compression aluminium honeycomb barrier. Following analysis of the data, it was found that the load paths generated by the barrier were different to those seen in the car to car crash of the same vehicle into itself. This resulted in the introduction of a small 250 psi compression bumper element in the front of the barrier face to produce the correct load path.

A test speed of between 56 km/h to 60 km/h appeared to reproduce the deformations seen in the car to car test of the same vehicle each travelling at 50 km/h.

AUSTRALIAN BASE RESEARCH (PHASE 1 TESTS)

In parallel with the initial EEVC work, FORS began a test series using a small (Toyota Corolla) right hand drive (RHD) transverse front engined, front wheel drive vehicle in the same drivetrain configuration as the EEVC small and medium test vehicles. Because the Australian vehicle was RHD, the engine was offset to the driver's side.

This initial program consisted of the following four tests:

- 40% overlap test into a deformable barrier (no bumper) at 60 km/h (B3014)
- 50% overlap test into a deformable barrier (no bumper) at 60 km/h (B3015).
- 50% overlap car to car test with each vehicle travelling at 50 km/h (B4009).

• 40% overlap test into a deformable barrier (with bumper as per Figure 1) at 60 km/h (B4054).

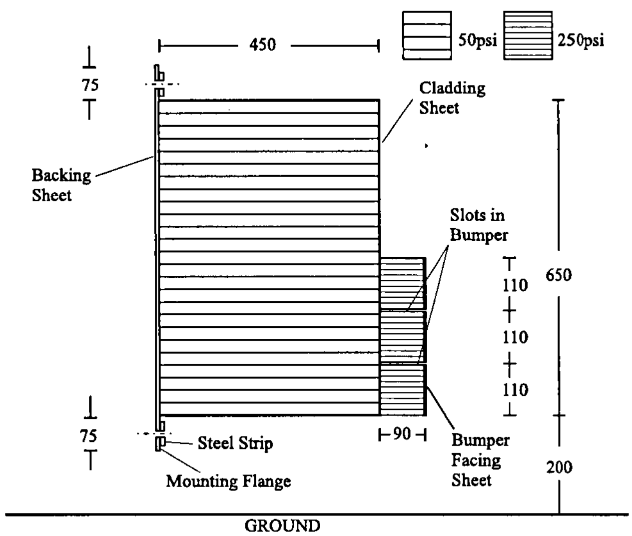

Figure 1. Deformable barrier used for Phase 1 test B4054.

The FORS Phase 1 tests confirmed that the bumper element was required to achieve the correct load path into the vehicle and that 40% appeared to be the correct amount of overlap for the test. However, a test speed of 60 km/h was required to reproduce deformations approaching those seen in the car to car test of the same vehicle each travelling at 50 km/h.

The following summarises the outcome of the Phase 1 tests:

- The time histories of both the engine and B-pillar transducers were quite different between the car to car test and the deformable barrier tests. The car to car test showed an earlier onset and higher peaks (Figure 2).

- As with the vehicle decelerations, the time histories of the dummy responses showed the same differences between the car to car test and the deformable barrier tests.
- The firewall decelerations in both the car to car test vehicles were much higher than in the deformable barrier tests.
- The lower leg injury levels in the car to car tests were higher than those in the 40% overlap deformable barrier test.

These results indicated that the EEVC deformable barrier did not reproduce the vehicle and dummy kinematics for the drivetrain configuration of the RHD test vehicle.

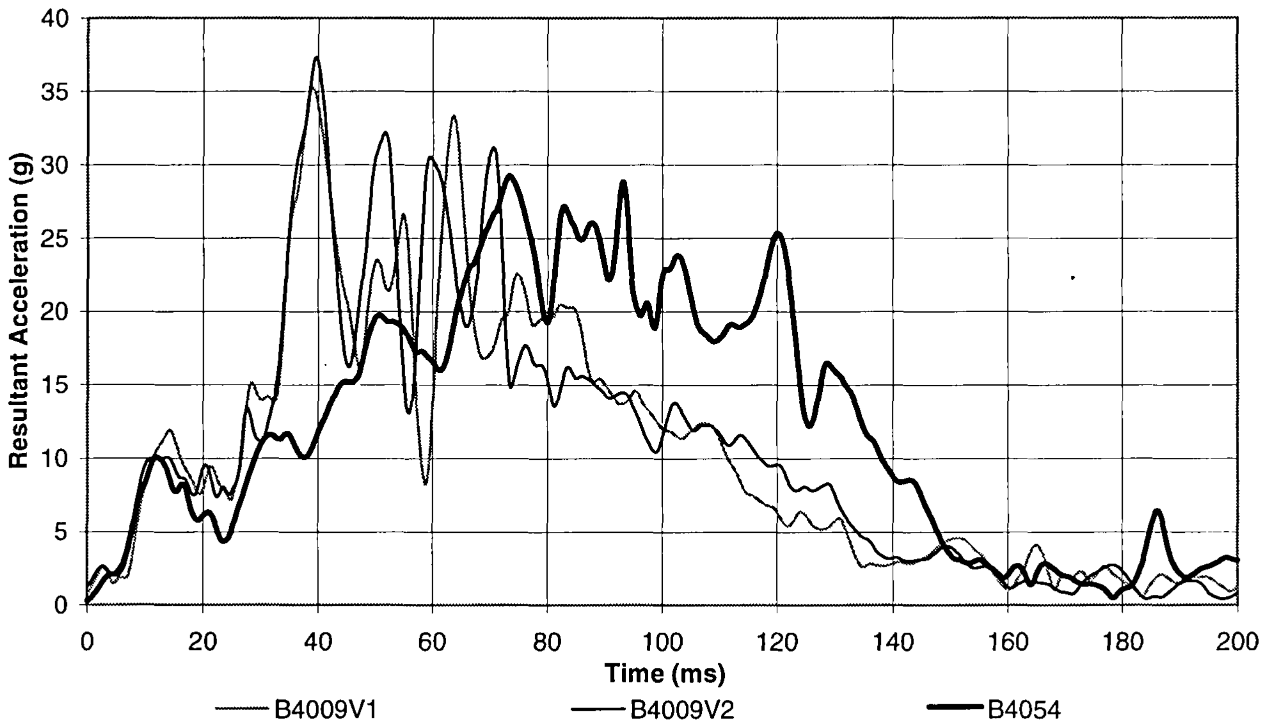

Figure 2. Right B-pillar resultant acceleration time histories for Phase 1 RHD Corolla car to car test (B4009) and 40% overlap barrier test with bumper element (B4054)

EFFECT OF DRIVETRAIN SYMMETRY (PHASE 2 TESTS)

This part of the FORS research aimed at complementing the EEVC work by examining the effects of drivetrain asymmetry on test outcome. Three LHD Corollas were tested as follows:

- 50% overlap car to car test with each vehicle travelling at 50 km/h (B5020).
- 40% overlap test into a deformable barrier (with bumper) at 60 km/h (B5027).

On examination of the vehicles after the LHD car to car test, the following points were noted when compared to the RHD car to car test:

- Both dashboards moved up so that the steering wheels rotated to a more horizontal position.
- Sill at the bottom of the A pillar has failed in compression. On the RHD cars, the sill failed in buckling and further back (about 1/3 the distance towards the B pillar).

- Significantly more passenger compartment intrusion than on the RHD cars. Both driver's lower legs were jammed between the intruded floorpan and dashboard.

- The deformation of the two LHD cars were very similar whereas the degree of deformation of the two RHD cars was quite different.

- The driver's HICs for the LHD cars were both around 1100. This compares with 675 and 1174 for the RHD cars with the higher figure corresponding to the vehicle with greater deformation.

The following observations can be made when the RHD and LHD pulses are compared:

- Without the engine bridging effect on the RHD vehicle, the LHD pulse has a more gradual onset.

- The LHD cars have a lower peak deceleration compared to the RHD cars.

- The LHD cars peak occurs later than the RHD cars.

- The LHD cars exhibit a higher level of deceleration for longer than the RHD cars after this peak. This behaviour resembles that observed in the deformable barrier tests using a homogeneous main honeycomb block.

Unfortunately, all the vehicle structure decelerations from the deformable barrier test were lost due to a data acquisition problem. A retest will be conducted early in 1996.

However, the dummy responses indicated that the LHD car to deformable barrier test correlated well with the LHD car to car test (Figure 3). This suggested that when the engine of a transverse engined front wheel drive car is not on the driver's side, the EEVC offset deformable barrier produces similar vehicle and dummy responses to the car to car test of the same vehicle into itself.

The deformation characteristics of the LHD cars indicates that without the bridging effect of the engine onto the firewall seen in the RHD cars, the front longitudinal, A-pillar and sill are required to dissipate more energy.

The results confirm that for offset frontal impacts, drivetrain and structural symmetry are important parameters to be considered when selection of representative test vehicles are made.

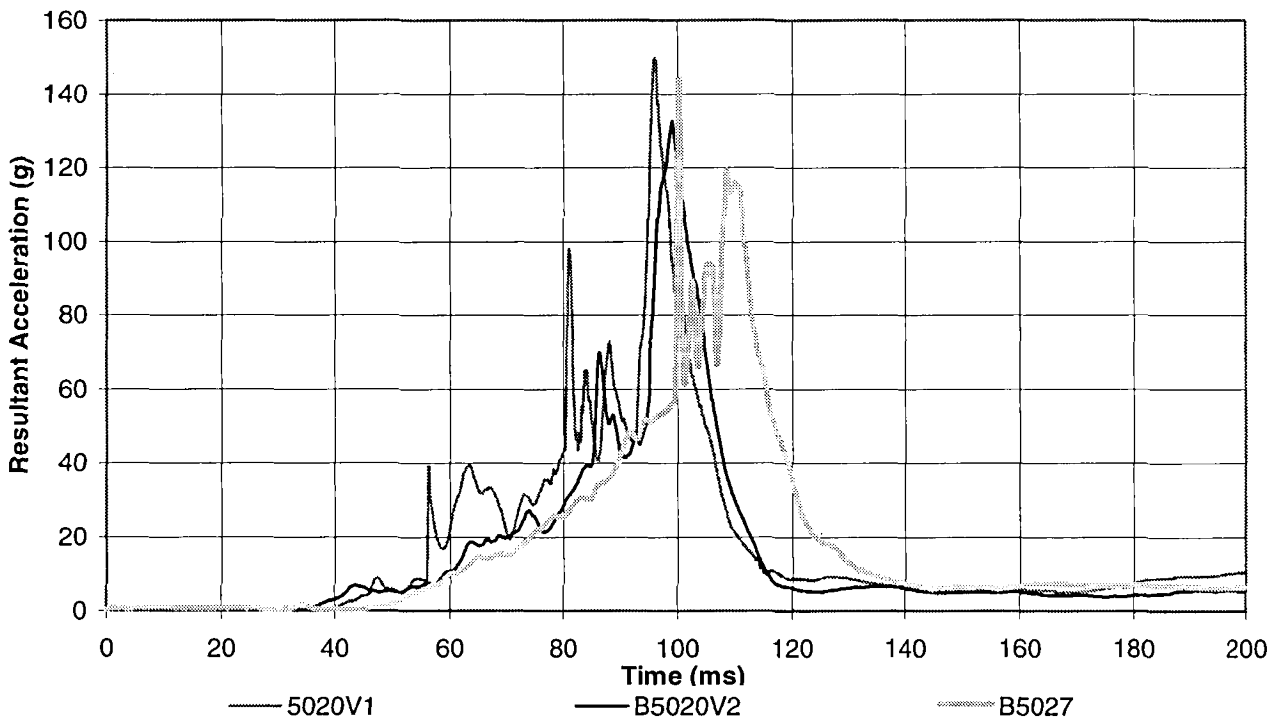

Figure 3. Driver head resultant acceleration time histories for Phase 2 LHD Corolla car to car test (B5020) and 40% overlap deformable barrier test with bumper element (B5027)

HOW DOES USING A DIFFERENT MODEL CAR AFFECT THE OUTCOME (PHASE 3 TESTS)

There was concern with some members of the EEVC WG11 that the test series based on the Toyota Corolla might lead researchers to an outcome that was vehicle specific.

For this reason, it was decided to conduct a car to car crash using a RHD Toyota Corolla impacting a RHD 1995 model Ford Laser Liata. The engine on both models is offset to the driver's side. This would allow comparison of the Corolla's vehicle and dummy response in car to car tests where it runs into itself and also into another vehicle model in the same weight class.

The Ford Laser Liata was also subject to an offset deformable barrier test using the barrier described in Figure 1 by the Australian New Car Assessment Program. This test was performed at 40% overlap and a speed of 60 km/h. As FORS has no direct affiliation with the NCAP testing, their technical committee was approached to purchase the crash test data for the Laser offset deformable barrier test. This provided another vehicle model to compare the vehicle and dummy responses of a car to car test against an offset deformable barrier test.

In the RHD car to car tests, the vehicle and dummy kinematics of the Corolla were similar irrespective of whether the other test car was also a Corolla or a different vehicle, in this case a Ford Laser.

For both the Corolla and the Laser, the EEVC offset deformable barrier did not produce the same vehicle or dummy kinematics as the car to car crash of the Corolla into the Laser or the Corolla into itself (Figure 4).

The outcome of Phase 3 supported the proposition that the vehicle and dummy responses seen in the Phase 1 tests were not vehicle specific to the Toyota Corolla (Figure 5).

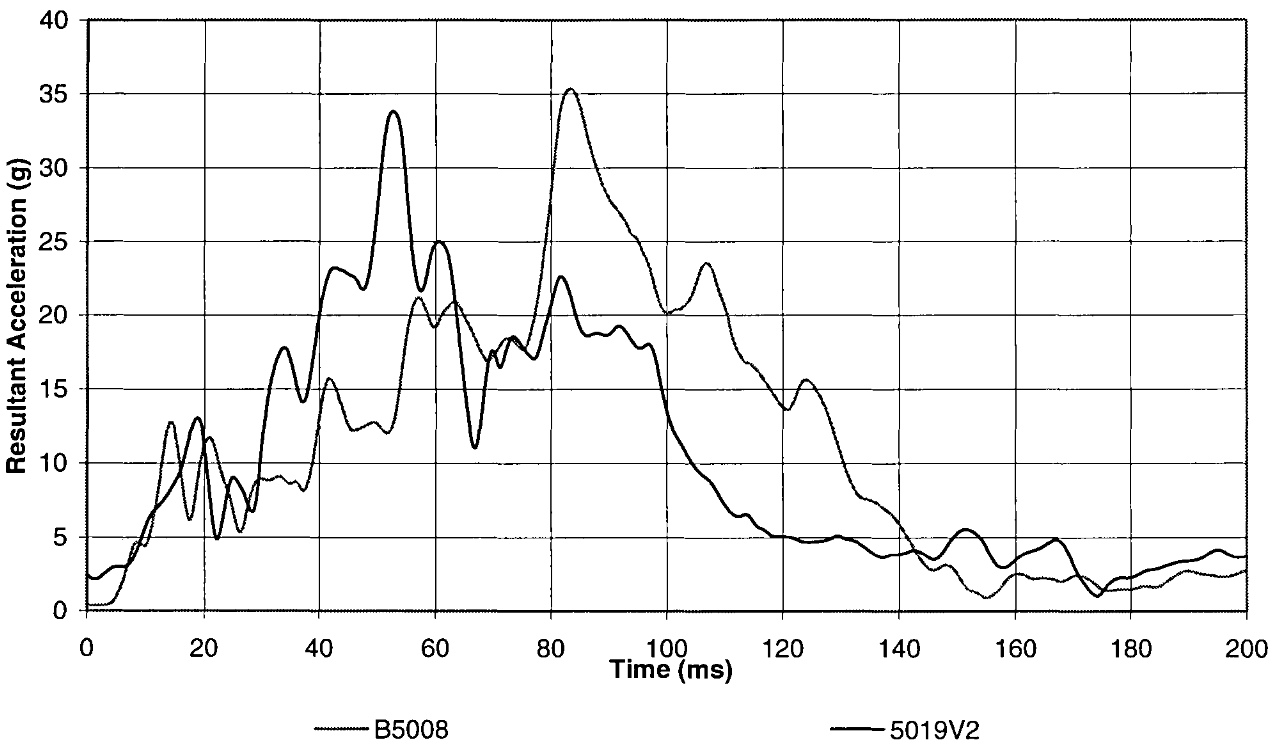

Figure 4. Right B-pillar resultant acceleration time histories for Ford Laser in Phase 3 RHD Laser/Corolla car to ca test (B5019) and Laser 40% overlap deformable barrier test with bumper element (B5008)

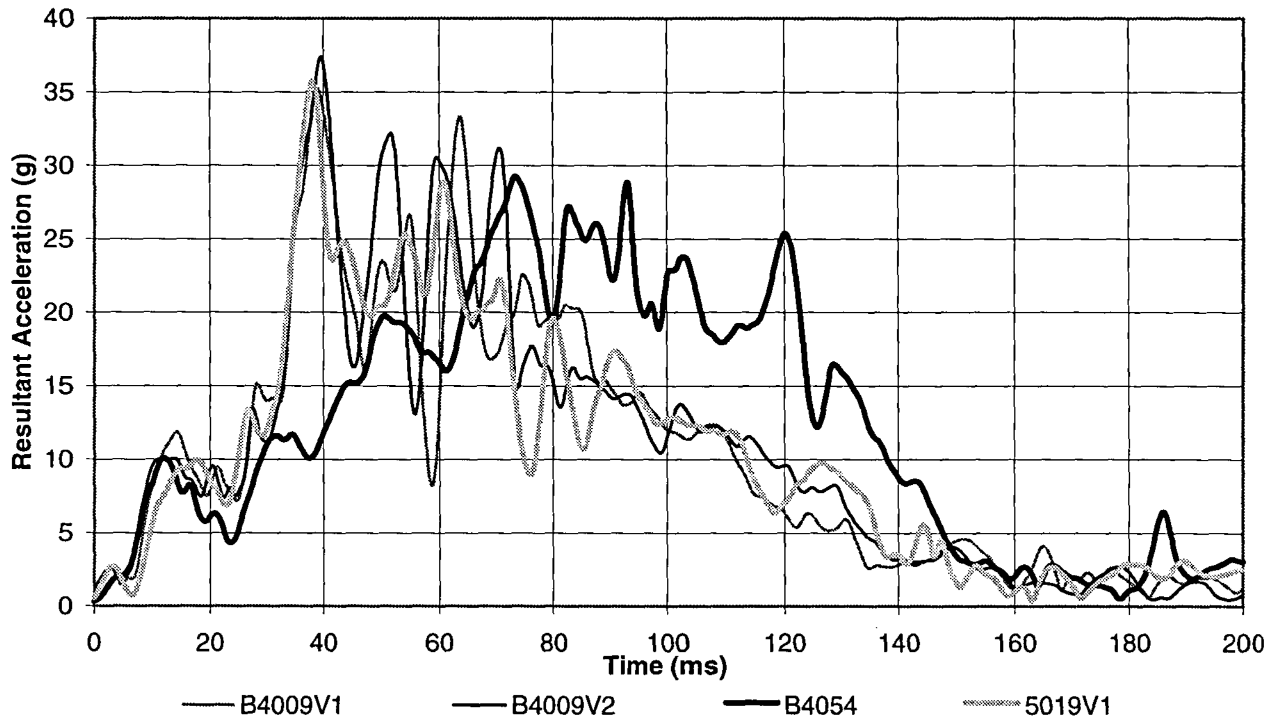

Figure 5. Right B-pillar resultant acceleration time histories for Phase 1 RHD car to car test (B4009), Phase 3 Laser/Corolla car to car test (B5019) and Phase 1 40% overlap deformable barrier test with bumper element (B4054)

TESTS WITH A MODIFIED DEFORMABLE BARRIER (PHASE 4 TESTS)

The outcomes of the first 3 phases suggested that drivetrain configuration on the transverse front engined vehicles had a significant bearing on the dummy and vehicle responses in both a car to car crash and a test into a deformable barrier.

The Phase 1 tests indicated that the deformation characteristics, deceleration and dummy readings were different between the car to car test and the test into EEVC offset deformable barrier. The LHD test series showed that the EEVC offset deformable barrier appeared to correlate reasonably well with the LHD Corolla car to car test.

This indicated that when the engine is offset to the driver's side in a car to car crash the engagement of the engines result in a different response than that obtained when the engine is not engaged.

This posed the question of whether the deformable barrier could be modified to replicate this engine engagement and the subsequent bridging effect of the engine to the firewall.

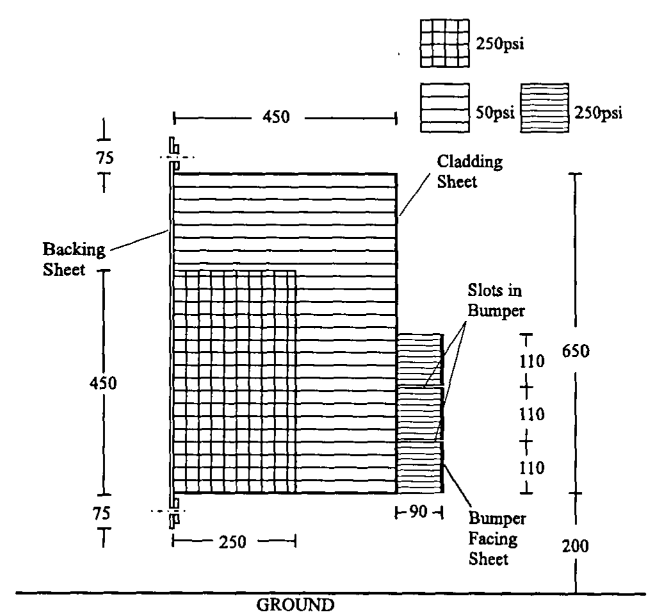

Figure 6. Modified deformable barrier with inserted core used for test B5021

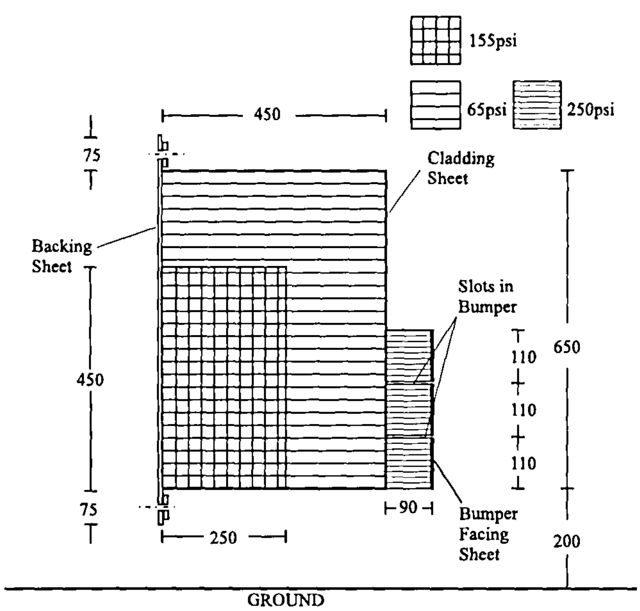

Figure 7. Modified deformable barrier with inserted core used for test B5028

To examine this, two tests were performed using a modified offset deformable barrier design. Both barriers had 3 bumper elements of 250 psi honeycomb at the front and were mounted 200 mm above the ground.

The first test (B5021) used a deformable barrier face consisting of a 50 psi compression aluminium honeycomb main core with an inserted core of 250 psi honeycomb at the rear (Figure 6). This first test was conducted at 60 km/h at 40% overlap.

The first test resulted in severe collapse of the passenger compartment which required the driver dummy's legs to be unbolted before the dummy could be removed. The injury data were much higher than for the car to car test. However, the vehicle and dummy responses were now better aligned with that seen in the car to car crash. The inserted core showed very little crush. However, the initial crush phase through the 50 psi honeycomb was still more gradual than the car to car pulse.

The second test (B5028) used a deformable barrier face consisting of a 65 psi compression aluminium honeycomb main core with an inserted core of 155 psi honeycomb at the rear (Figure 7). The second test was conducted at 55 km/h at 40% overlap. The 65 psi honeycomb was chosen to try and address the gradual onset in the initial crush phase.

The second test produced vehicle deformations

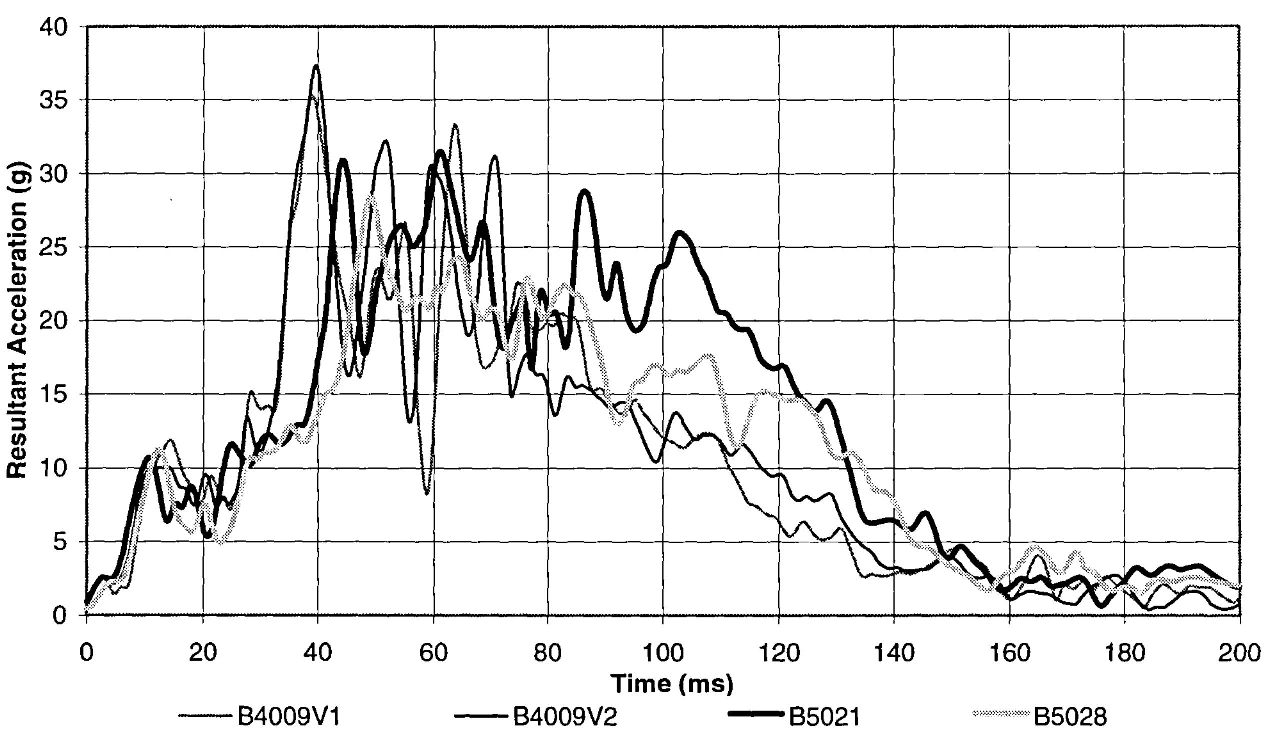

Figure 8. Right B-pillar resultant acceleration time histories for Phase 1 RHD Corolla car to car test (B4009) and Phase 4 40% overlap modified deformable barrier test with bumper element (B5021 and B5028)

similar to the car to car test and dummy data about half way between the outcomes of the two cars in the car to car test. Although there was more crush of the inserted core, the 65 psi honeycomb did not appear to address the "soft" initial crush. However, the front of the vehicle appeared more like the car to car test without the centre "notch" seen with the 50 psi homogeneous barrier face.

Figure 8 compares the B-pillar resultant acceleration time histories of the two Phase 4 tests with the original RHD car to car test (B4009).

FUTURE DEVELOPMENTS

The recent endorsement by the European Parliament to move towards a draft EC directive which specifies an offset frontal deformable barrier test based on the EEVC WG 11 proposal appears to have accelerated the process of getting this test procedure "on the books".

The September 1995 WG 11 meeting in Sweden finalised the draft offset frontal test procedure[4] for consideration by the European Parliament in early 1996. At the meeting, there was considerable debate on the efficacy of the current instrumented lower legs. It was finally agreed to proceed with the regulation using newly developed legs which will improve test to test repeatability.

Australia believes that the move to a deformable barrier element is a significant first step forward in improving offset frontal crash protection.

Therefore FORS will be preparing a draft Australian Design Rule which will incorporate the requirements of the finalised WG 11 draft test procedure. This draft ADR, together with a cost benefit analysis will be issued for public comment during the first half of 1996.

SUMMARY

- ADR 69 for full frontal impact protection is already in place in Australia and will test the vehicle's restraint system in a high deceleration crash situation. This ADR will continue in parallel with an ADR on offset frontal crash protection.

- The offset test will test the vehicle's structural integrity and, with lower leg injury criteria applied, the vehicle's ability to prevent debilitating leg injuries. It will also mitigate upper torso injuries resulting from reductions in occupant survival space.

- The offset test should result in vehicle designs with improved crash energy management.

- In the meantime, a draft ADR will be prepared and issued for public comment incorporating the requirements of the finalised WG 11 draft test procedure for offset frontal crash protection It will be proposed that the draft ADR be implemented towards the end of the decade.

ACKNOWLEDGMENTS

The author wishes to acknowledge the assistance of Russell Higgins and Mark Terrell of the Federal Office of Road Safety in purchasing the test vehicles, deformable barriers and preparing the charts and figures for the final report of this project.

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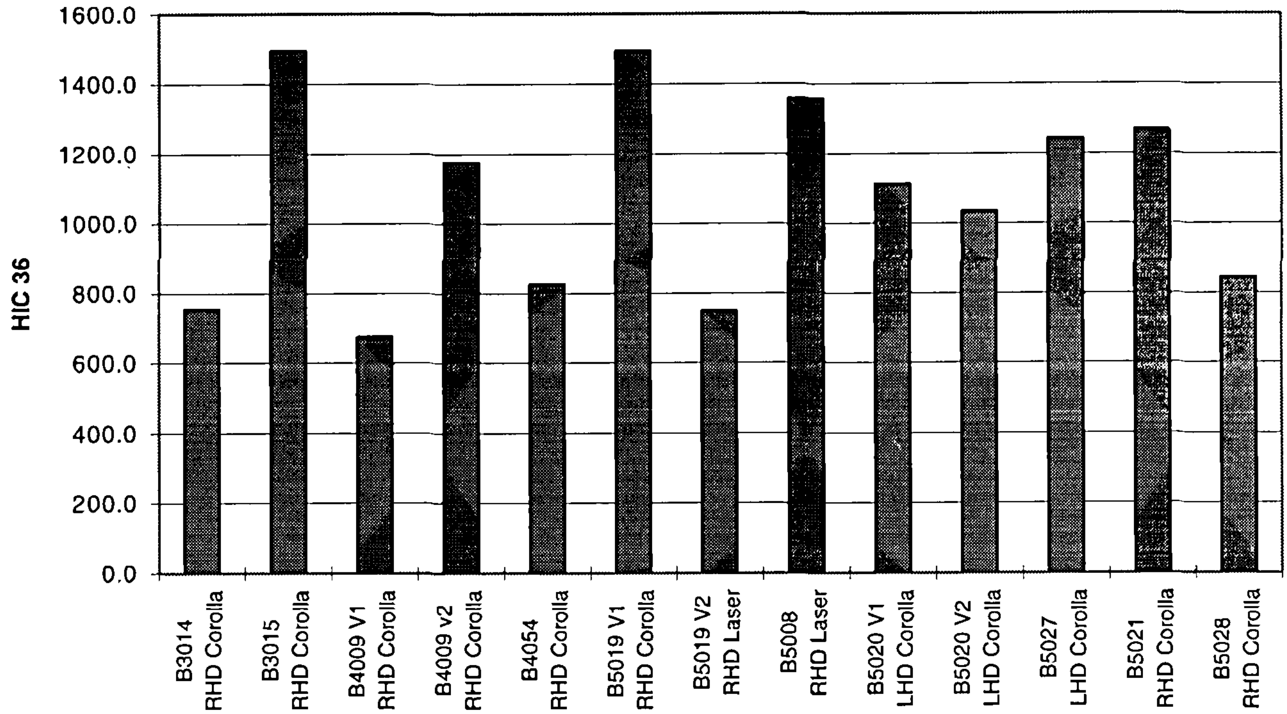
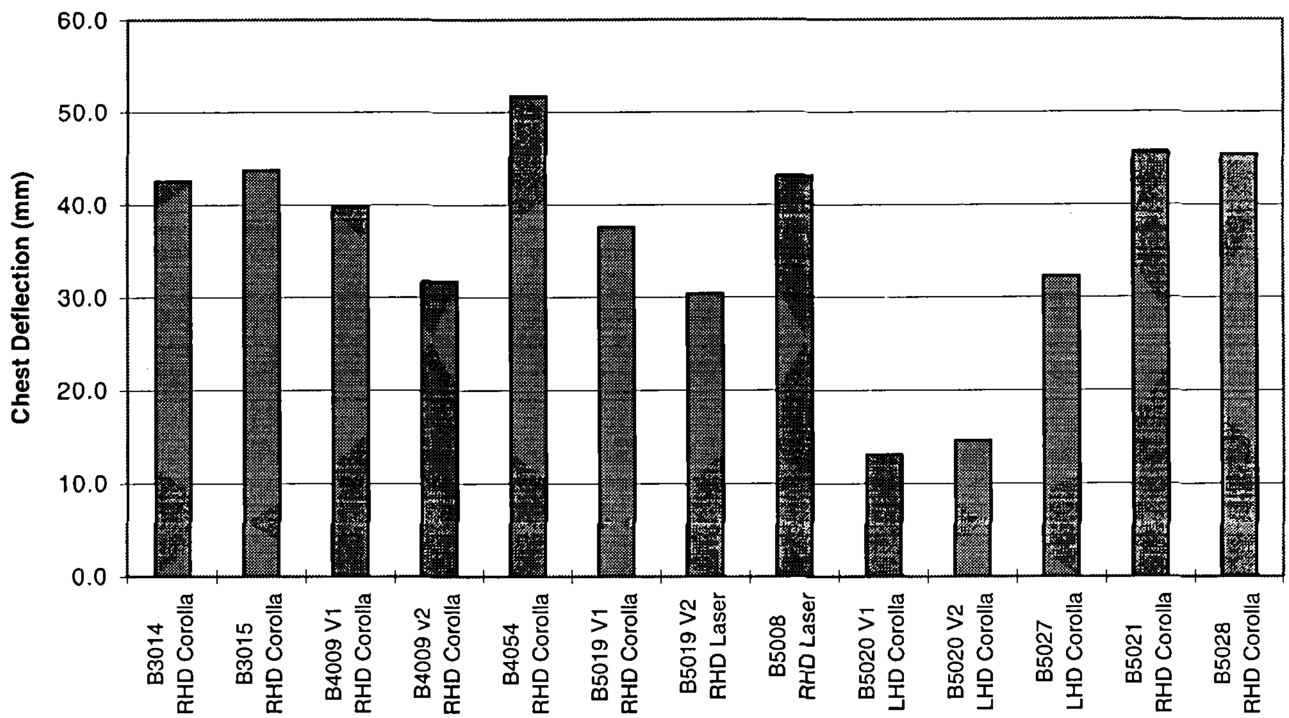
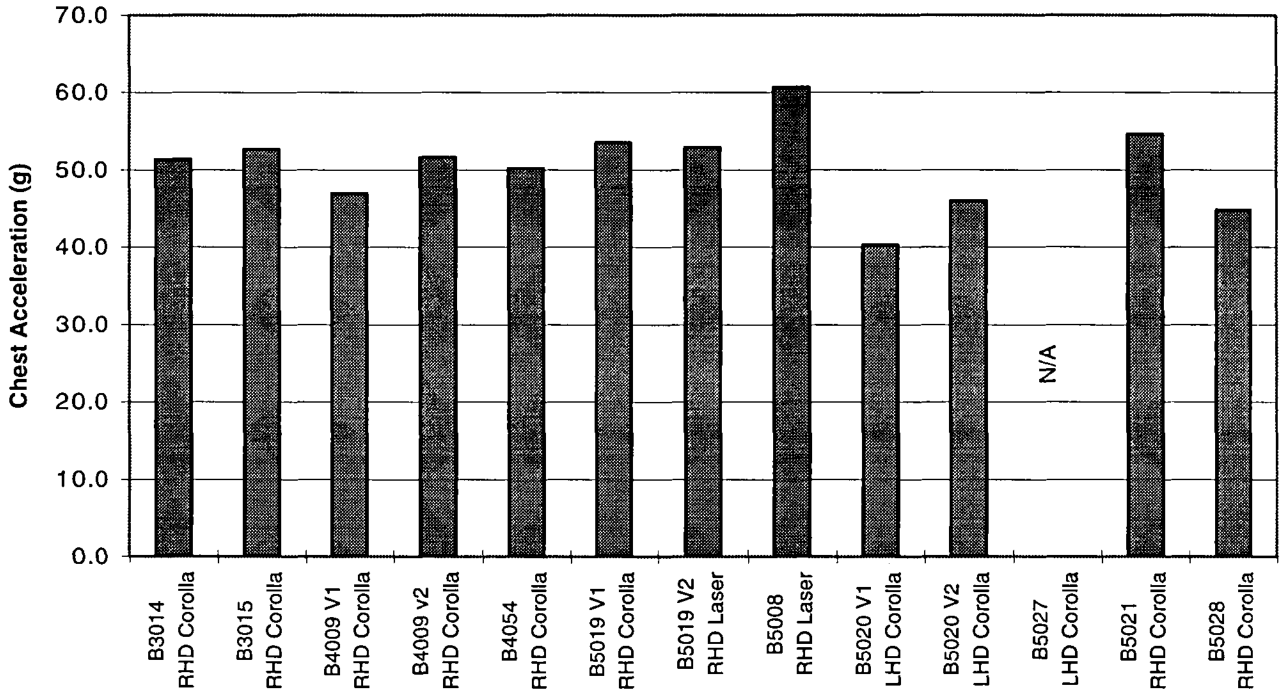


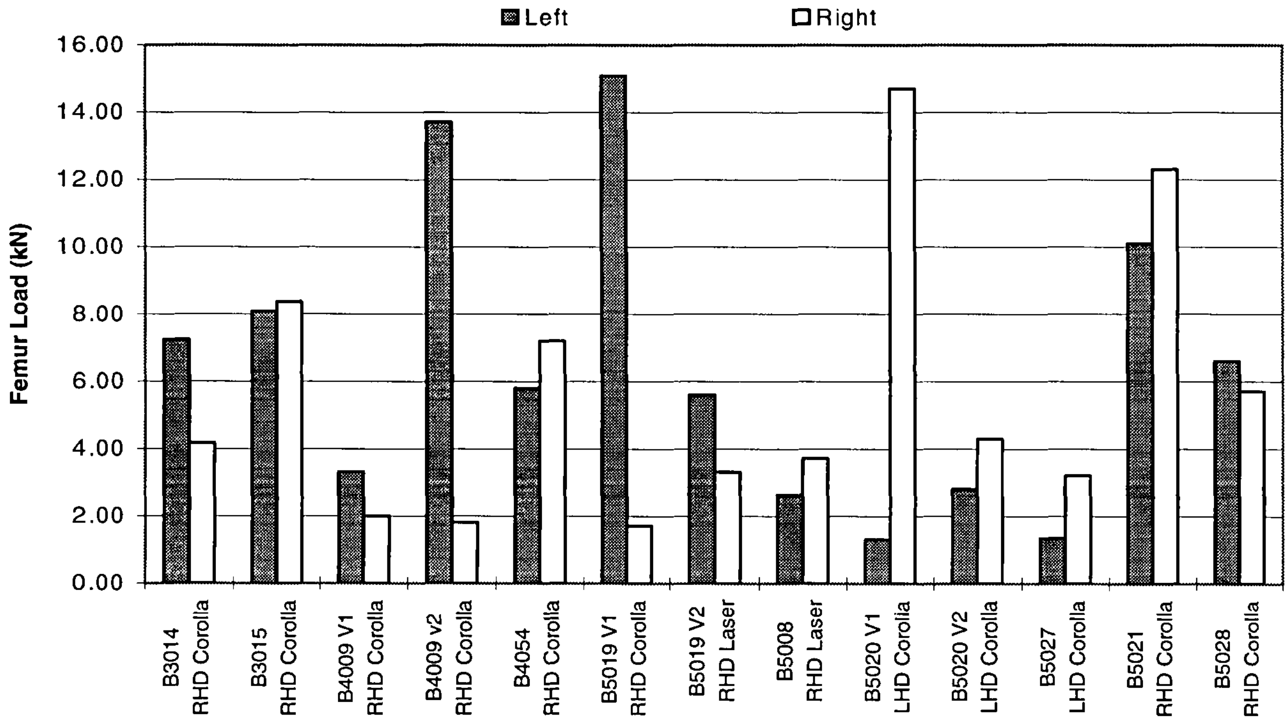
Head Injury Criterion (HIC) calculated over 36msec interval. Proposed Limit 1000



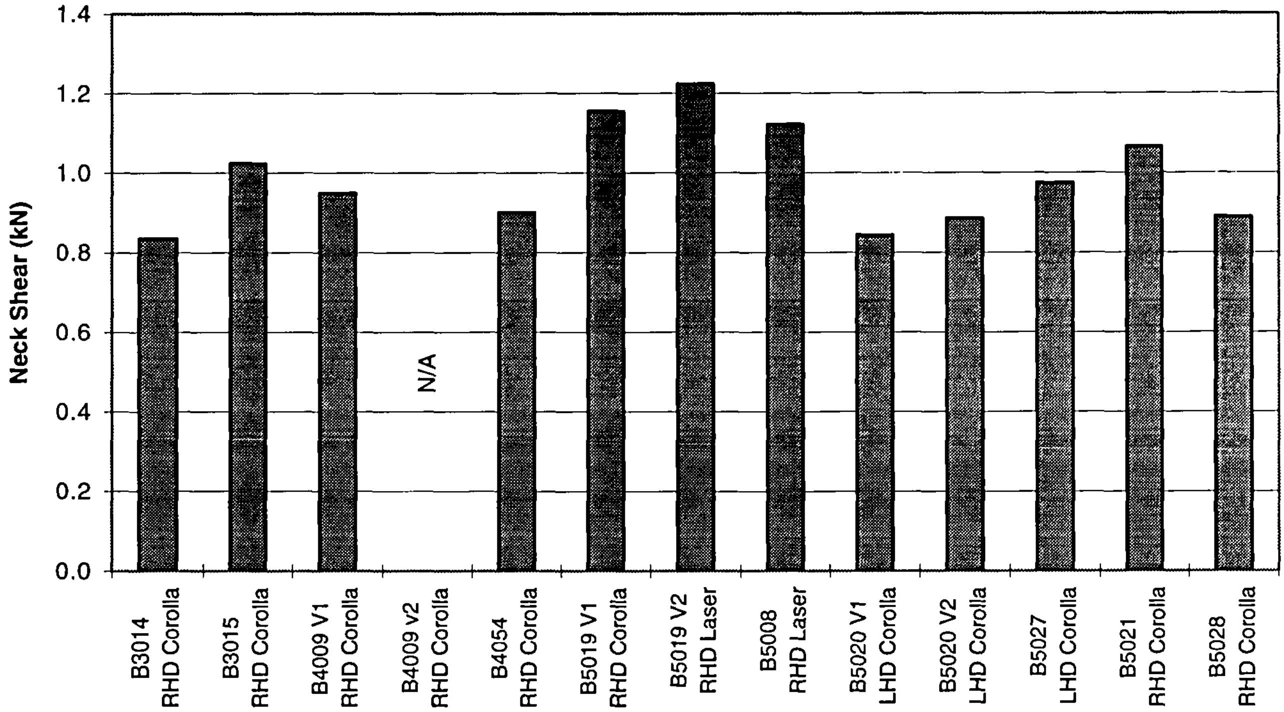
Chest deflection. Proposed limit 50mm



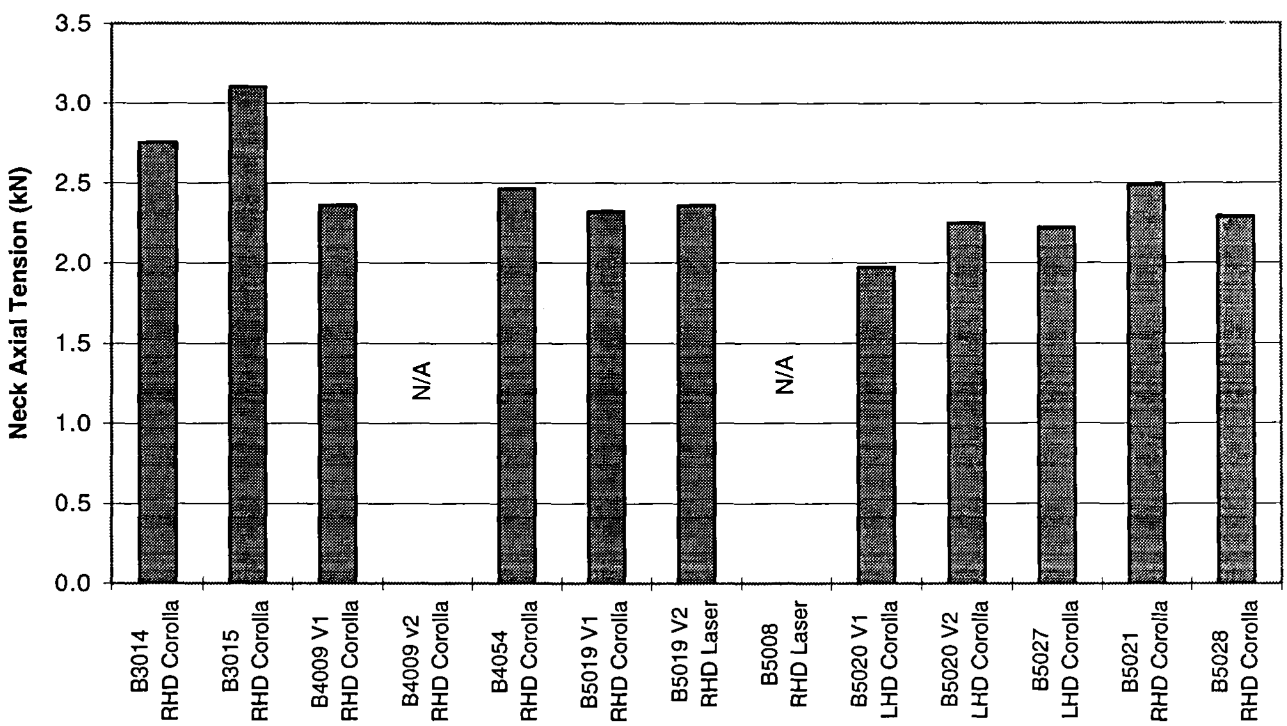
Chest Acceleration. Proposed Limit 60g



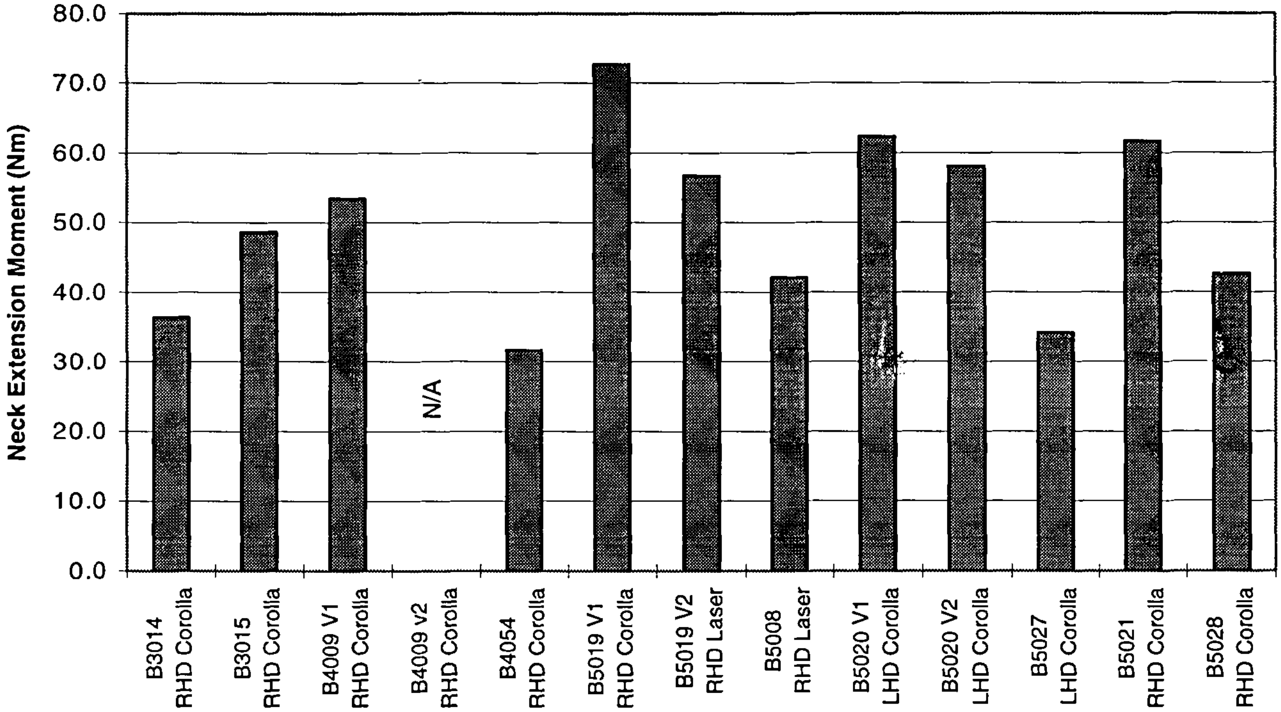
Femur Load. Proposed limit 9.1kN (time dependent)



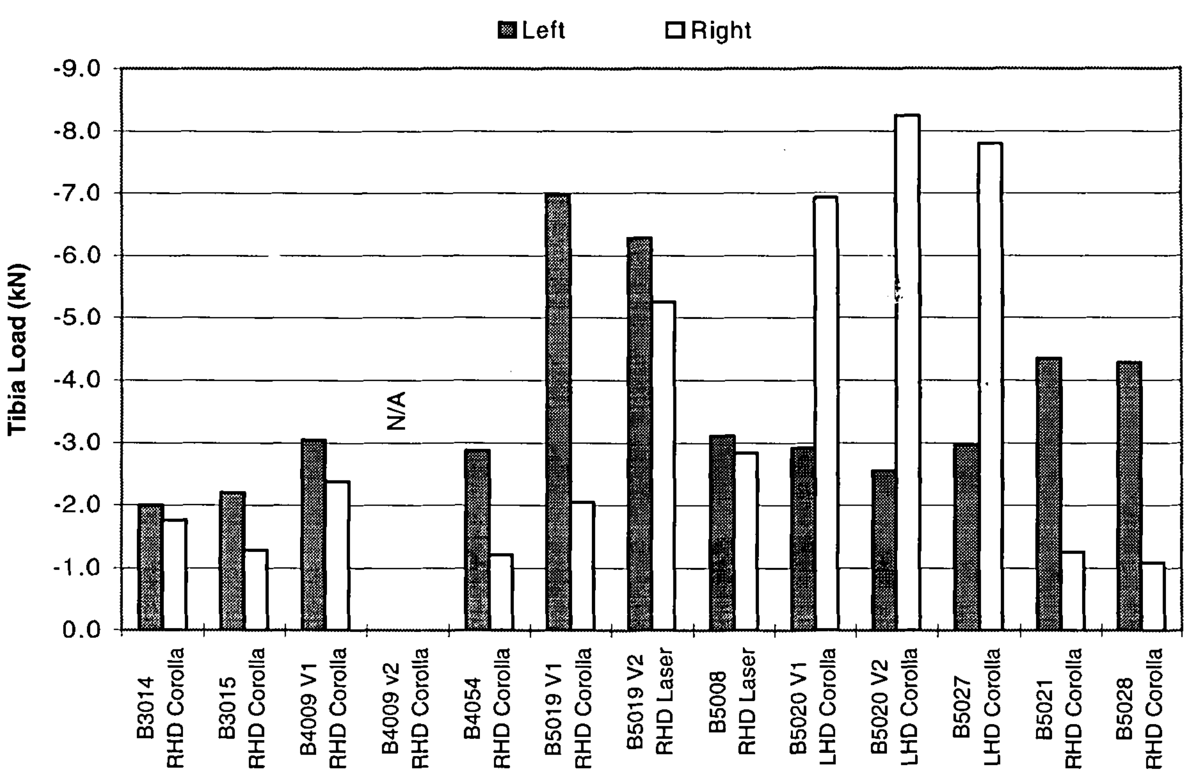
Neck Shear. Proposed limits 3.1kN (time dependent)



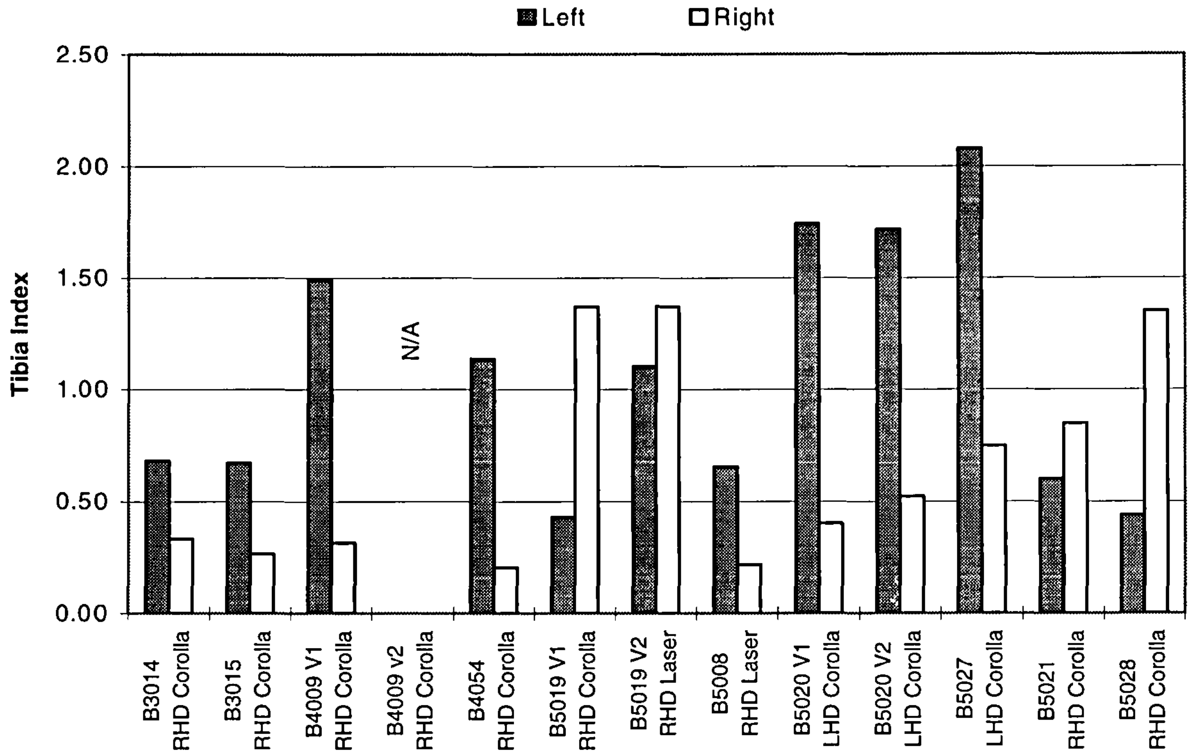
Neck Tension. Proposed limit 3.3kN (time dependent)



Neck Extension Moment. Proposed limit 57Nm



Tibia compressive load. Proposed limit 8kN



Tibia Index. Proposed limit 1.3

CONSIDERATION FOR BELTED FMVSS 208 TESTING

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Paper Number 96-S3-O-03

Abstract

Years of testing and field experience demonstrate that today's airbags in combination with safety belts are performing well and that the vast majority of airbag-related injuries caused by contact with an inflating airbag are minor. However, the reported incidence of severe-to-fatal injuries establish a need to further enhance airbag designs. The present paper introduces an overview of both testing and math modeling of driver airbags. This overview helps to identify both the design opportunities and the design constraints for such enhancements. The design parameter that shows the greatest significance is the inflator. The design constraint that poses the most difficulty is the unbelted occupant test requirement. A case is accordingly made for lower-energy inflators as a means to further reduce the risk of injury from an inflating airbag. Toward that end, a new inflator specification is proposed to facilitate understanding of an inflator's performance in occupant-near-module scenarios. Additionally, the implications of implementing lower-energy inflators are addressed from the standpoint of both FMVSS 208 and real-world frontal collisions. Finally, a critical review of the relevance of unbelted testing is presented.

Introduction

In the early 1970's, when airbag technology was first being developed, the National Highway Traffic Safety Administration (NHTSA) initiated rulemaking that would have required vehicle manufacturers to install front and rear seat airbags and meet specified injury criteria (including 40 Chest Gs) utilizing unbelted instrumented dummies in a variety of tests including frontal, angled, and rollover crashes. The rule which ultimately emerged in 1985, Federal Motor Vehicle Safety Standard (FMVSS) 208, required that the vehicle be equipped with passive restraints, and meet certain specified injury criteria utilizing unbelted instrumented dummies under specified barrier conditions. Pursuant to FMVSS 208, tests against a rigid barrier must be conducted with full engagement of the front end at 0 degrees and +/-30 degrees to the front of the vehicle.

The phase-in of FMVSS 208 started in 1987, and in 1994, 85% of new cars sold were equipped with airbags (nearly half of which had both driver and passenger frontal impact airbags). The number of dual airbag-equipped cars and trucks is increasing rapidly. With an increasing number of airbags in the vehicle fleet, accident data related to the effectiveness of airbags in real-world collisions is beginning to accumulate. Analysis has accordingly begun.

The objective of this paper is to review the benefits of airbags, to identify the potential "side effects" of airbags, and to suggest measures that may further improve the overall effectiveness of airbags by reducing the risk of "side effects."

Accident Data: Airbag Effectiveness

Several studies have attempted to measure the real-world effectiveness of airbags. Malliaris, *et al.* [1,2,3] have analyzed data from both the National Accident Sampling System (NASS) and the Fatal Accident Reporting System (FARS) to determine the field performance of airbags. In February 1996, NHTSA provided a report to Congress [4] on the effectiveness of occupant protection systems and their use. Each of these studies show that driver airbags with a properly used 3-point belt system are much more effective in reducing the risk of moderate-to-severe injuries and fatalities when compared to the no-restraint condition. However, the effectiveness of the airbag without the belt system is substantially lower than those of the airbag+belt or belt-alone scenarios. Specifically, Reference 4 reports that the airbag-alone condition may have a negative effect in reducing the risk of Abbreviated Injury Scores (AIS) 2 to AIS 4 injuries. To explain these observations, Malliaris, *et al.* [1] have identified some behavioral differences between drivers who wear safety belts and those who do not. They report the following: drivers who do not wear their belts are four times more likely to be drunk than those that do; non-belt-wearing drivers are involved in substantially higher-speed accidents; non-belt-wearing drivers are two to three times more likely to be involved in rollover accidents.

NHTSA's report to Congress [4] estimates the fatality reduction effectiveness of airbags as 28-35% in pure frontal crashes (12 o'clock) and between 15-18% in all frontal crashes (10-2 o'clock). If the pure frontal crashes (12 o'clock) are removed from the analysis, the 10 and 2 o'clock crashes show no significant benefit of the airbag. This observation is contradictory to laboratory data from FMVSS 208 crash tests in which the dummy occupant Chest Gs are invariably less in the +/-30 degree impacts when compared to those of pure frontal impacts. NHTSA's report further indicates that the effectiveness of the airbag alone (i.e., without the 3-point belt) in reducing moderate or greater injuries (AIS 2+) is negligible. Similar results were derived by Malliaris, *et al.* [3]. In their study of the NASS data, they have derived the effectiveness of various restraint systems in reducing total harm to car drivers in towaway crashes (where total harm accounts for all injuries, including fatality). The results of their analysis indicate that the belt system alone and the airbag+belt condition each reduce the risk of total harm by approximately 65-70%. However, the airbag, when used without the belt system, reduces the risk of harm by approximately 10% (with a large standard error of estimation). Given that all restraint systems have been designed utilizing the same crash test procedure and injury criteria, such real-world variations in effectiveness indicate that the airbag-alone crash test as required by FMVSS 208 has questionable real-world benefit.

Accident Data: Airbag "Side Effects"

As with any injury- or fatality-reduction countermeasures, "side effects" can be expected. Some of these "side effects" are adverse in nature to some in the population. The Insurance Institute for Highway Safety (IIHS) Status Report published in February 1996 has identified some injuries associated with airbag contact. They report that 43% of all airbag deployments result in airbag contact injuries, and the vast majority of these contact injuries (95.6%) are minor (AIS 1), but 0.5% of these injuries are serious-to-fatal (AIS 3-6). NHTSA's report to Congress [4] also indicates that airbag inflations are associated with increased risk of arm injuries when compared to injuries without airbag deployment. These arm injuries are not solely attributable to airbag contact, but also to arm flailing (which may result in vehicle interior or occupant facial contacts).

The above discussion of the accident data leads the authors to conclude that today's airbags, which must meet or exceed the FMVSS 208 unbelted requirements, are effective in reducing the risk of injuries and fatalities for belted occupants. For unbelted occupants, the real-world effectiveness of the airbag in reducing fatalities is much lower, and may be insignificant in reducing the risk of moderate-to-severe (AIS 2+) injuries.

Experimental and Theoretical Research

Airbags that are designed to reduce the risk of injury to unbelted occupants in 48 km/h, rigid barrier-type crashes are designed to be fully inflated approximately 40 ms or less from the time of initial impact. As a result, a substantial amount of energy must be generated in a very short time in order to inflate the airbag in time to help reduce the risk of injury to an occupant from contact(s) with stiffer portions of the vehicle interior. A number of experimental studies have been conducted over the last twenty-five years to help assess the potential of injury from deploying airbags.

Minor-to-Moderate Injuries: Among the most frequent airbag contact injuries is that associated with skin and flesh. This mode of injury seems to be independent of crash severity. An experimental study by Reed, *et al.* [6] shows that skin and flesh (integumentary) injury is related to the velocity of the airbag fabric. Whereas bag fold and tethers have some effect on fabric velocity, Reed, *et al.* [6] showed that lower capacity inflators have the most significant effect in lowering fabric velocities.

Eye injury, although rare, may also be related to airbag fabric velocity. Kikuchi, *et al.* [7] have shown that the threshold of corneal injury is reached when the fabric velocity at the point of impact is approximately 41 m/s. A recent study by Powell and Lund [8] reports that the leading edge fabric velocities of driver airbags, designed to meet and exceed FMVSS 208, range between 47.5 m/s and 91 m/s. Theoretical modeling studies at Ford indicate that airbag leading edge velocities and the distance from the inflator when the maximum velocity occurs are substantially reduced with less energetic inflators. This is in agreement with experimental results reported by Sugimoto, *et al.* [9] Moreover, when the same models are then extended to include an occupant, the facial contact (slap) forces associated with lower-energy inflators are predicted to be less than those associated with the higher-energy inflators (see Figure 1).

Arm fractures are the most frequent bone fractures caused by airbag contact during its inflation phase [3,4]. These fractures can occur for either belted or unbelted driver occupants. The authors have developed a complex mathematical model of the airbag, cover, and forearm. Results from the model (shown in Figure 2) indicate that substantial reduction in contact forces on the arm from possible airbag/cover/forearm interaction can be attained with lower-energy inflators. These lower-energy inflators also substantially reduce the peak velocity of the arm, suggesting that arm flailing effects could also be reduced.

The aforementioned experimental and theoretical studies indicate that, by using lower-energy inflators, further reductions in the risk of the most frequent minor-to-moderate injuries attributed to airbag inflation could be achieved.

Severe-to-Fatal Injuries:

Child Injuries: The earliest investigation of the risk of airbag inflation-related injury to occupants in close proximity to a deploying airbag was by Patrick and Nyquist [10] in 1972. In their experimental study, they identified the potential for serious head, neck, chest, and abdominal

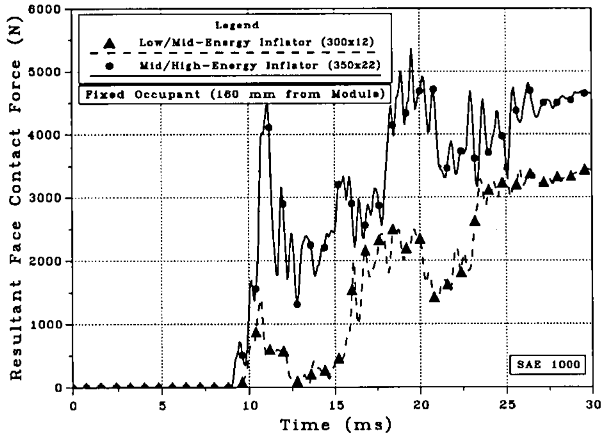

Figure 1: Predicted Face Contact Forces for Two Inflators

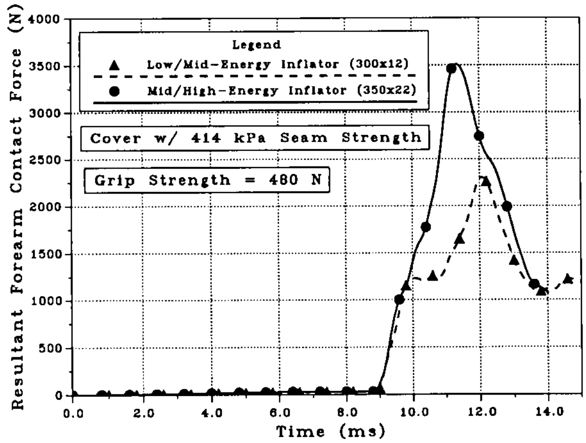

Figure 2: Predicted Forearm Contact Forces for Two Inflators

injuries to children in close proximity to a deploying passenger side, frontal impact airbag. Aldman, *et al.* [11] in 1974 and Takeda and Kabayeshi [12] in 1980 came to similar conclusions. More exhaustive studies were conducted by Mertz and his group, and were presented at the 1982 Experimental Safety Vehicle (ESV) conference [13,14,15]. Mertz's reports were followed by a study by Prasad and Daniel [16] in 1984. Collectively, these studies showed that severe-to-fatal head, neck, chest and abdominal injuries could be caused by contact with deploying airbags if child surrogates were in close proximity to the airbag inflator when inflation was initiated. Mertz [13] also noted a strong association between the higher level inflators (peak combustion chamber pressures of 15 MPa or greater) and the more severe injuries (AIS 5-6).

The probability of occupant near-module exposure has also been previously investigated. Montalvo, *et al.* [14] estimated that in 1 million car-years of exposure, 51 unrestrained, front-seat children ranging in ages from infant to four-years old would be close to the instrument panel at the time of collision. Some of these children would be near the instrument panel due to their initial position before crash and others due to pre-impact braking. In a 1980 report, NHTSA estimated that in one million car-years of operation, "a maximum of only 15 infants and small children are likely to be in the region of the airbag at the time of deployment, and few if any of them are likely to be injured by bag deployment." NHTSA further believed that the risk of injury to the small child could be substantially reduced by careful design, tuning, and development testing. Although significant improvements in the passenger airbag designs have occurred since 1980, the more current accident data discussed earlier in this paper indicate that passenger airbags designed to meet or surpass the requirements of the FMVSS 208 can still cause serious-to-fatal injuries to unbelted child occupants who are in close proximity to the airbag at the time of deployment.

Adult Injuries: The earliest investigation of airbag interaction forces with human surrogates was reported by Horsch and Culver [17] in 1979. In their study with driver airbag inflations against both a body block and a Hybrid III dummy, it was shown that the peak interaction force between the airbag and human surrogate varied inversely as separation between the chest and the airbag module increased. In their tests with the Hybrid III dummy, the dummy chest was essentially bottomed out (with greater than 75 mm of sternum deflection) when the dummy chest was within 25 mm of the airbag at the time of deployment initiation. They concluded that there was a high probability of severe chest injuries to human occupants in close proximity to the airbag systems tested. Further studies involving both the mid-sized male and small female Hybrid III dummies in various configurations relative to the driver airbag module [18,19] support the conclusion that occupants in close proximity to the airbag, during the inflation process, can be severely injured in the thorax, neck, and head areas. These studies postulate two phases of airbag inflation: 1) a deployment initiation phase involving airbag-in-module inflation, module cover swelling, and subsequent tear seam separation, and 2) a membrane phase consisting of airbag pressurization as it expands to its "taut" state. The initiation phase is more related to the risk of thoracic injury while the membrane effect phase is more related to the risk of neck injury.

The following sections of this paper will primarily address the deployment initiation phase of driver airbag inflation. Topics will include a survey of various present-day driver airbag systems, an assessment of the most significant airbag system design parameters with respect to injury risk reduction, and possible countermeasures and their related ramifications with respect to FMVSS 208 compliance.

Test Survey of Current Driver Airbag Systems

A variety of driver airbag systems presently sold in the United States were acquired and tested with both the Hybrid III mid-sized male and small-sized female dummies in a static, laboratory test environment. The test dummy was positioned in accordance with the International Standards Organization-recommended dummy positions [20]. Two different initial positions were considered: 1) the chest-on-module condition which maximizes thoracic response, and 2) the neck test which emphasizes airbag/neck interaction. When possible, repeat experiments were conducted to help assess variability (with the averaged results reported herein). Dummy responses of particular interest were chest deflection, viscous criterion (V*C), and neck loads/moments. In these tests, chest deflections ranged between 38 and 77+ mm (which bottoms out the dummy chest) and the V*Cs between 1.0 and 3.0 m/s. Sternum velocities ranged from 6 m/s to 16 m/s. Neck extension moments from 12 to 92 N.m were also observed. (The specifications for the tests yielding these results will be delineated below.)

It should be noted that the aforementioned laboratory tests using instrumented anthropomorphic dummies were static in nature. Out-of-position occupant response relationships between static and dynamic tests have been previously published [18, 21]. These two reports show a static-to-dynamic scaling ranging from 1-to-1 to 2-to-1, respectively. The exact relationship appears to be dependent on the particular nature of the static test. The authors expect the dynamic tests to result in higher chest and neck loadings than their static counterparts for the specific test configurations chosen for the present experimental survey due to their similarity to those of Reference 21. Further investigations of this topic are warranted.

For both of the aforementioned test conditions, a number of driver airbag system design parameters were considered in screening experiments. These parameters included the following: inflator output, cover stiffness, tear seam pattern and strength, airbag folding pattern, airbag volume, thoracic stiffness, etc.

As shown in Reference 22, the airbag folding pattern can influence membrane effects. Results from our chest-on-module experiments shown in Figure 3, however, indicate that the folding patterns tested (Leporello and Petri) do not enhance a system's performance subject to the chest-on-module condition. The working hypothesis for this phenomenon is that, during the deployment initiation phase of the inflation process, membrane loading is nearly non-existent, with peak V*Cs occurring before the airbag is even visible (via examination of high-speed films).

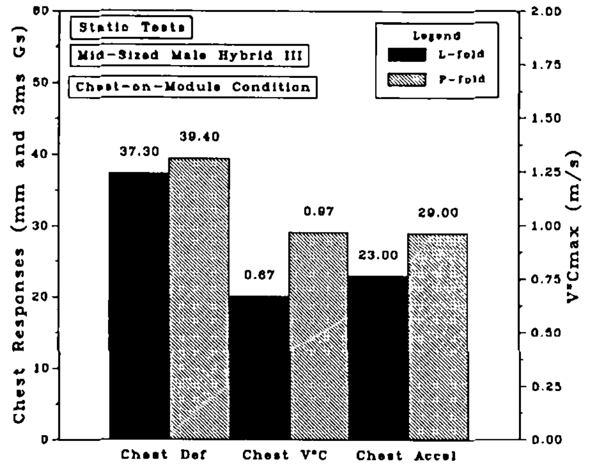

Figure 3: Comparison of Two Airbag Folding Patterns

Airbag module cover effects were then considered in the experimental screening process (c.f. Figure 4). Note that, for identical inflators and airbags, stiffer covers generally lead to slightly greater V*Cs for the chest-on-module condition. Yet, when the cover was completely cut from the module, thoracic responses were still likely to result in AIS 4+ injuries with the mid-high energy inflator. Results supporting these claims are found in Figure 4.

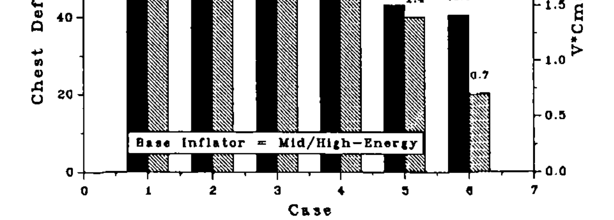

Figure 4: Screening Test Results Involving Covers, Seam Strengths, and Energy Levels of Inflators

Also shown in Figure 4 is the introduction of inflator effects for screening purposes. It was in this area that the most significant variations in dummy occupant responses were observed. To better explain the results, some background is warranted.

Quantification of Inflator Specifications

Traditional Specifications: A tank test is conducted with intent to derive an inflator's characteristics via thermodynamic analysis [24]. The inflator of interest is discharged into a rigid tank. The pressure in the tank, P(t), is measured. Additionally, the mass loss from the inflator is measured and used as an approximation of the total gas mass supplied by the inflator, m_{max}. Peak tank pressure, P_{max}, is one important parameter traditionally used to specify the inflator. The second is the maximum time derivative of the tank pressure, $\dot{P}_{max}$, known as the "rise rate." (Often the derivative is calculated from a moving 5 or 10 ms linear curve fit of the tank pressure data.) For example, an inflator that, when discharged into a 28.3 liter, rigid tank produces a peak gage pressure of 350 kPa and a maximum 10-ms based rise rate of 22 kPa/ms, is traditionally, deemed a "350 x 22" inflator.

The "peak pressure x rise rate" inflator specification that has proven to be adequate with respect to the occupant restraint system design for FMVSS 208 may be inadequate for occupants seated close to the module. As evidenced by Reference 18, the relationship between peak tank pressure and chest-on-module V*C are not very well correlated (with the same being said for the correlation between rise rate and chest-on-module V*C). It is proposed here that maximum inflator thrust is a better candidate for correlating an inflator characteristic with peak V*C for the chest-on-module condition.

Proposed Specifications: It is proposed that, in addition to peak pressure and rise rate specifications for inflator performance, a term based on peak inflator thrust be introduced to better quantify the pre-20 ms phase of the inflation process. The authors believe a "peak pressure x rise rate x thrust term" specification would better facilitate understanding of an inflator's performance for occupants both near and far from the inflator at the time of deployment initiation.

An estimate of peak inflator thrust can be derived from tank tests. Simple control volume analysis of the tank conservation laws for mass and energy (in combination with the perfect gas law) yields an inflator's mass flow rate and stagnation temperature [23]. Maximum inflator exit thrust may then be quantified as follows:

$$\dot{m}V|_{max} = \dot{m}_{max}\sqrt{\gamma R T_{stag}}$$

Note that for inflator-to-inflator comparisons involving same-gas output, the product under the radical of the above equation could just as well be represented by only the stagnation temperature. This convention will be applied hereafter since all of the inflators studied generate nitrogen.

The resulting inflator thrust variable, $\dot{m}_{max}\sqrt{T_{stag}}$, may serve as a means to correlate an inflator's characteristics with peak V*C of a mid-sized male Hybrid III in the chest-on-module condition. An example of the calculation of this variable from traditional tank curves is found in Appendix A.

The correlation between the peak inflator thrust variable and peak dummy V*C is illustrated in Figure 5. The data were generated from a combination of tests (consisting of both screening tests and an inflator design-of-experiments experimental survey). Wherever applicable, the entire system (airbag, inflator, module) was kept in its original equipment state. Figure 5 shows good correlation between the peak inflator thrust variable and peak dummy V*C. Even with the variety of folding patterns, cover materials, seam strengths, and variability attributed to both inflator and dummy placement, a correlation coefficient (R^2) of 0.77 was observed. Consequently, the characteristics of the inflator are deemed to be highly significant when considering the chest-on-module condition.

The effects of an inflator's contribution to airbag injury potential are conflicting. In tests with dummies and in real-world crashes, occupants contact a fully-inflated airbag. A higher-energy inflator's contribution in combination with safety belts has been shown to be highly positive with respect to injury risk reduction potential. Yet, from

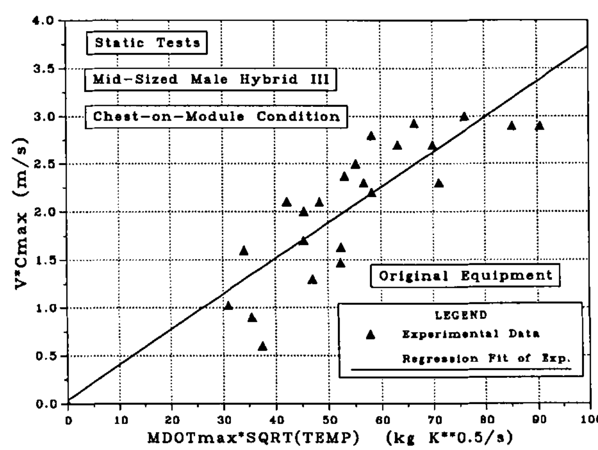

Figure 5: Inflator Thrust Variable versus Peak V*C

the vantage point of those occupants not belted, a higher-energy inflator may be deemed negative (with potential airbag-inflation contact injuries including minor injuries such as skin abrasion, moderate injuries such as arm fractures, and the most severe viscous criterion-related fatalities). Therefore, inflator characteristics have been identified as playing a major role -- in both overall benefits and collective "side effects." Since the inflator is deemed the most significant inflatable restraint system design parameter, inflator output tailoring would accordingly be an obvious approach toward a significant reduction of adverse, inflation-related "side effects."

Inflator Tailoring

The first attempts at inflator output tailoring have shown mixed results. First, math models of the deployment initiation phase of the chest-on-module condition were developed to aid in the process of inflator tailoring. The models were then shown to be faithful to the hypothesized maximum thrust-maximum V*C trend as shown in Figure 6. With knowledge of the early phase thrust limits (i.e., from initiation to initiation+15 ms) derived from Figure 6 and estimates of the sufficient airbag pressure needed after 25 ms to maintain acceptable in-position, mid-sized occupant responses, a theoretical inflator was quantified (Figure 7). The resulting tank curve was deemed an "S-shape" type curve with a slow onset and a subsequent rapid ascent to its peak pressure. First attempts to emulate the performance of such a curve in hardware have identified a flaw in its conception: it produces a low viscous response, 0.43 m/s, but a high neck moment (108 N.m) for the chest-on-module test condition. These results are contained in Table 1. Much more work is therefore required to tailor an inflator whose output is favorable to all occupant sizes in both test modes. (A revised study which attempts to theoretically quantify such an inflator can be found in a later section.)

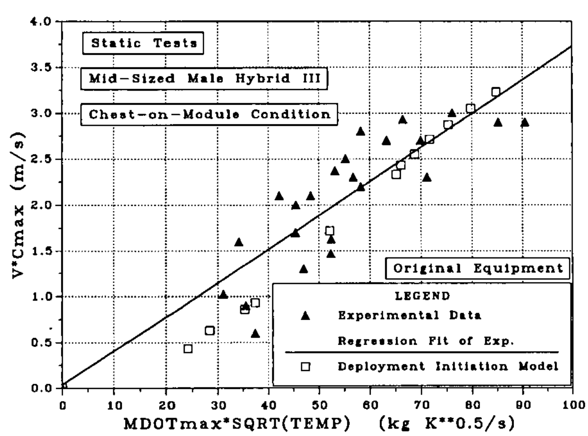

Figure 6: Model Validation for Inflator Tailoring

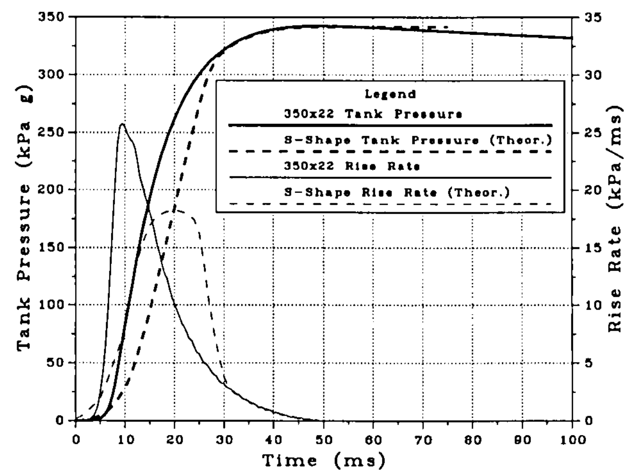

Figure 7: Tank Test Information Derived for Mid-Sized Hybrid III, Chest-on-Module Inflator Tailoring

TABLE 1. Tailored "S-Curve" Inflator Test Results for Hybrid III Mid-Sized Male in Various Test Positions

Condition	Avg Chest Def (mm)	Avg V*C (m/s)	Avg Up My Extension (N.m)
Chest-on-Module	35	0.43	108
Nose-on-Rim	30	0.20	47
Published Provisional Ref. Value	50 (belt) 65 (airbag)	1.0	57

The preceding survey of existing driver airbag modules has been rather extensive, including a wide variety of possible design parameters (sometimes including extreme, non-practical designs), a variety of test configurations, and a variety of occupant sizes. From analysis of the related occupant response data, it can be concluded that no known production inflator/bag/cover/fold pattern combination exists that can significantly reduce risk to all occupants in all positions.

Other Opportunities

Another possible countermeasure to help reduce the risk of injury of airbag/occupant interaction involves introduction of technologies that either suppress or adaptively tailor the belt/bag/inflator characteristics. These technologies have come to be known as "smart restraint systems." Such countermeasures, while desirable, pose significant technical challenges in their implementation. Moreover, smart restraint systems may be able to reduce the inflator energy in low-speed collisions, which undoubtedly will reduce the potential for inflation-related contact injuries, but these inflators may still be required to deliver higher-energy levels in higher-speed collisions to meet or exceed the unbelted test requirements.

A number of years would be required to design, verify, and implement a "smart restraint system" on a mass scale. Over this period of time, given the rapid growth of airbags into the total vehicle population, the vehicle market will be saturated with higher-energy airbags that meet or exceed the current unbelted, FMVSS 208 regulation. The effectiveness of smart restraint systems would accordingly take <u>even longer</u> to manifest itself in real-world accident data. What is needed is a means to significantly reduce inflator energy that can be implemented <u>as soon as practicable</u>. Smart restraint systems, while acknowledged to have significant potential in the area of injury risk reduction, have a state-of-readiness issue that must be considered when assessing various design enhancement opportunities.

Implications of Lower-Energy Inflators in Simulated Rigid Barrier Impacts

Another possible step toward a significant reduction of the risk of airbag-related, contact injuries is unbelted testing at a lower speed; the logic being that, given the increased time to fill the airbag before contact with the unbelted occupant, lower-energy inflators could be introduced.

In order to conduct mathematical studies to determine the effect of inflator energy, candidate inflators needed to be selected. Examples of present-day inflators are the 350 x 22 x 71 (previously designated "350 x 22") and the 300 x 12 x 37, representing "mid/high" and "low/mid"-energy inflators, respectively. It is important to note that, although the 300 x 12 x 37 inflator has significantly less energy than the 350 x 22 x 71 inflator, it is not free of risk of injury to the small female Hybrid III dummy in the chest-on-module condition as shown in Table 2. Consequently, further reduction in inflator output is required to further reduce the risk of injury to small female-size driver occupants.

A theoretical attempt to quantify an inflator for the small driver (an analysis technique similar to that done for the mid-side male Hybrid III) was conducted by means of a regression analysis of available experimental data for the chest-on-module condition. The results are shown in Figure 8. Note that,.according to the regression curve for the small female dummy, an inflator with a $\dot{m}_{max}\sqrt{T_{stag}}$ value of 28 kg K**0.5/s would accordingly yield a peak V*C of approximately 1.0 m/s. By direct scaling of the mass flow

TABLE 2. Small Female Hybrid III Dummy Responses for the Chest-on-Module Condition Tests involving both Low-to-Mid-Energy and Mid-to-High Energy Inflators

Inflator	Chest Def (mm)	V*C (m/s)	Upper Neck Moment (N.m)
300 x 12 x 37	48.3	1.3	17.0 Flex/ 27.2 Ext
350 x 22 x 71	61.0	2.3	14.1 Flex/ 92.2 Ext
Published Provi-sional Ref. Value	41 (belt) 65 (airbag)	1.0	104.0 Flex/ 31.0 Ext

rate of the 300 x12 x 37 inflator (while keeping the same average stagnation temperature), a theoretical inflator with a 230 x 9 x 28 specification results. (Note that no scaling between the static test-based V*C and the dynamic test-based V*C was introduced.) This mathematical construct of an even lower-energy inflator can then be considered from the standpoint of FMVSS 208 compliance.

Mathematical studies were subsequently performed to determine the effect of lower-energy inflators on a mid-sized male Hybrid III driver responses in full barrier-type collisions. The speed range selected was between 23 and 56 km/h. Two restraint conditions were considered -- belted+airbag and airbag alone. Three inflators were studied viz., 350 x 22 x 71, 300 x 12 x 37, and 230 x 9 x 28. A mid-sized vehicle was considered for purposes of analysis. The results of the mathematical modeling are shown in Figure 9.

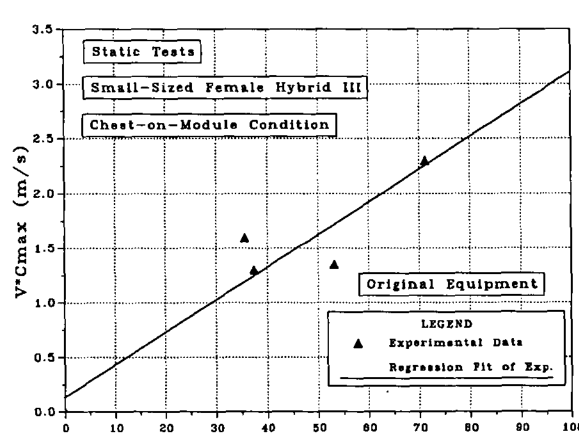

Figure 8: Inflator Thrust Variable vs. Peak V*C (Small Female Hybrid III Dummy)

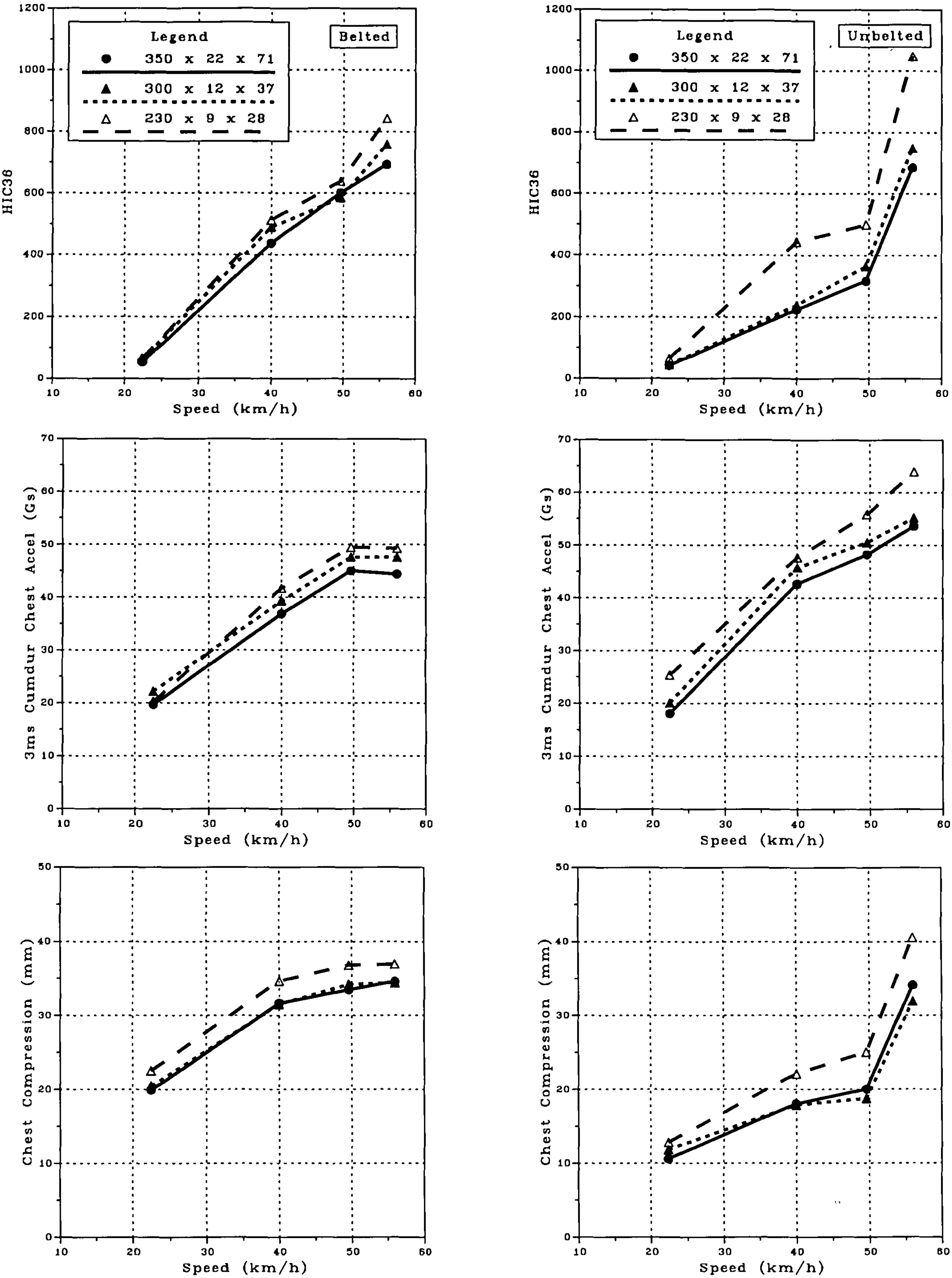

Figure 9: Predictions of Mid-Sized Male Hybrid III Responses in a Mid-Sized Car
(involving Various Speeds, Various Levels of Restraint, and Various Levels of Inflator Energy)

Initial comparisons of the results of Figure 5 and Figure 8 from the standpoint of inflator-energy changes (involving both belted and unbelted occupants) show that occupant thoracic responses are significantly affected in the chest-on-module condition. But it should also be noted from Figure 9 that demonstration of FMVSS 208 compliance at 48 km/h would not be possible with statistical confidence utilizing the hypothetical 230 x 9 x 28 inflator even for a mid-sized car's comparatively mild crash pulse. Consequently, given that small cars and trucks tend to have stiffer crash pulses than mid-sized cars, it could be inferred that FMVSS 208 compliance with small-sized cars and trucks would be highly improbable with such inflators as well.

Also evident from inspection of Figure 9 is the prediction that apparent trade-offs do exist with use of the lower-energy inflator. For example, the vehicle's New Car Assessment Program (NCAP) rating [24] would decrease by one star with the theoretical 230 x 9 x 28 inflator compared to the 350 x 22 x 71 inflator for a belted occupant subjected to a 56 km/h rigid-barrier impact. The significance of this one-star reduction in injury risk reduction, though, needs to be considered. The validity of the NCAP Star System as it pertains to airbags is questionable due the use of 36-ms-based HIC (instead of its 15-ms counterpart) for head injury risk prediction [25] and chest acceleration as a surrogate measure for chest injuries (instead of chest compression which is a more direct measure of chest injury potential).

Figure 9 also indicates that the occupant responses predicted for belted occupants are less sensitive to inflator variations than their unbelted counterparts. Additionally, it is noted that unbelted occupants subjected to rigid barrier impact speeds of approximately 40 km/h in the modeled mid-sized car would meet the response criteria necessary for FMVSS 208 compliance with the 230 x 9 x 28 inflator. However, compliance to the FMVSS 208 injury criteria may not be possible with such an inflator for smaller cars at 40 km/h.

The relative importance of the trade-offs associated with idealized, rigid barrier frontal impacts (with respect to real-world events) will be addressed in the following section.

Real-World Frontal Accidents

The severity of any impact without significant intrusion is determined by the change in velocity (ΔV) and average deceleration. Tarrierre, *et al.* [26] studied a sample of 330 real-world frontal crashes and found that, for 50 km/h ΔV accidents, the mean deceleration ranged from 7 to 17 Gs. However, experimental impacts against full overlap, 90 degree rigid barrier ranged between 14 and 25 Gs. This fact led them to believe that 90 degree impact into a rigid barrier was not representative of real-world accidents. They also found that the mean deceleration of crashes against 30 degree rigid barrier and car-to-car 50% offset crashes were more representative of real-world accidents.

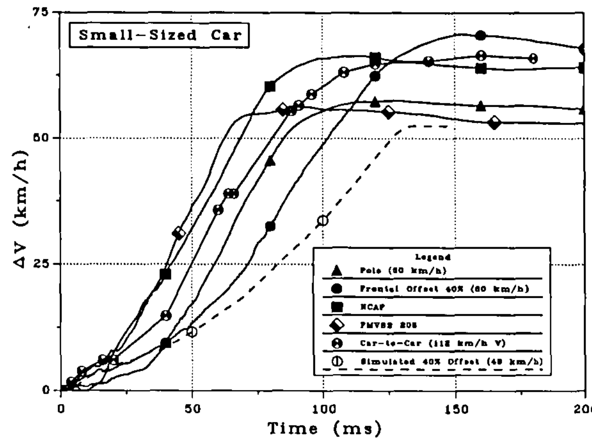

Figure 10: Crash ΔVs for Various Modes of Frontal Impact (Small Car)

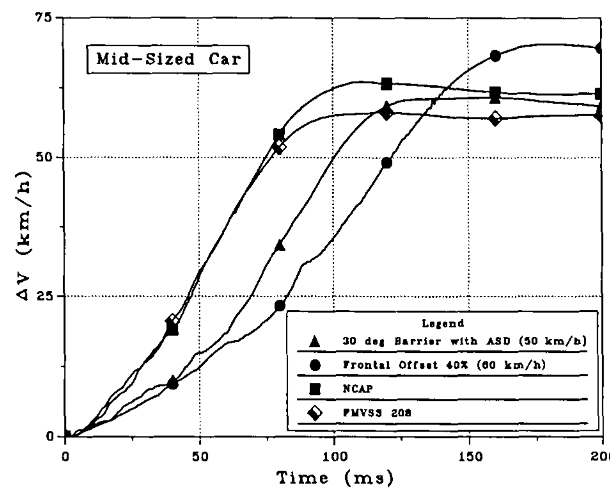

Figure 11: Crash ΔVs for Various Modes of Frontal Impact (Mid-Sized Car)

Accident reconstruction with load limiting shoulder belts were performed by Mertz, *et al.* [27] with sled testing involving ΔVs ranging between 44 and 53 km/h. To increase the severity of the crash, peak acceleration levels were varied between 14.5 to 19.0 Gs. Based on chest deflection data and its correlation with real-world thoracic injuries, it was concluded that only 5% of the real-world accidents approached the severity of a 48 km/h rigid barrier impact. Since all of their tests were conducted in the range of the FMVSS 208 ΔV, it could be concluded that, in only 5% of real-world accidents, at 48 km/h ΔV, would the acceleration levels be comparable to those of a rigid barrier impact. It can be further concluded that the crash durations are substantially longer than those encountered in the FMVSS 208 full barrier test condition -- a case that would allow for lower-energy airbags.

Support of the aforementioned hypothesis is further provided by the authors' experience with frontal crash tests in various modes. Figures 10 and 11 show the velocity-time histories of various frontal crashes for a compact and mid-sized car. It can be seen that distinct differences in the time durations exist between full-barrier and other types of crash tests. Similar results have been reported by Lund, *et al.* [28]. The average accelerations in the non-FMVSS 208 crashes are in general agreement with those reported by Tarriere, *et al.* [26]. In the 40% offset deformable barrier impacts, the chest accelerations and HICs are substantially lower than those in NCAP tests as reported by Prasad and Smorgonsky [29]. Once again, this indicates that real-world accidents involve lower average accelerations and longer time durations than those in full-barrier types of crashes.

The math model that was used to predict mid-sized, unbelted, Hybrid III responses in a mid-sized car subjected to 48 km/h, rigid barrier frontal impact was then used to make similar restraint assessments and predictions associated with crash pulses of the 30 degree, anti-slide rigid barrier type. The results are shown in Tables 3 and 4. It can therefore be concluded that lower-energy inflators, while not adequate for FMVSS 208 rigid barrier compliance, may provide substantial injury risk reduction to unbelted occupants in the vast majority of real-world frontal impact accidents.

In a similar fashion, belted occupant responses may be considered. Tables 5 and 6 show predictions of the performance of various inflators for both 90 degree rigid barrier and 40 percent offset deformable barrier frontal impacts at 56 km/h and 60 km/h, respectively. The hypothetical 230 x 9 x 28 lower-energy inflator therefore shows excellent real-world potential for reducing the risk of injury to belted occupants in frontal impacts.

Relevance of Unbelted Testing

The United States is the only country in the world requiring unbelted frontal crash tests. When FMVSS 208 was initially proposed, the belt wearing rate was approximately 10% and was unlikely to substantially increase in the near term. However, a substantial increase in belt wearing rate has now been achieved via belt wearing regulations and public education. The current belt wearing rate in the United States is approximately 67% and as high as 85% in primary enforcement states. As a result, whereas the unbelted test in FMVSS 208 had basis when first proposed, present day belt-wearing rates necessitate its reassessment.

TABLE 3. Model Results for 90 degree Frontal Impact with Rigid Barrier (Mid-Sized Car, 48 km/h, Unbelted Mid-Sized Male Hybrid III)

Inflator	HIC36	3ms Chest Gs
230 x 9 x 28	499.6	55.9
300 x 12 x 37	364.5	50.6
350 x 22 x 71	316.8	48.3

TABLE 4. Model Results for 30 degree Frontal Impact with Anti-Slide Device (Mid-Sized Car, 48 km/h, Unbelted Mid-Sized Male Hybrid III)

Inflator	HIC36	3ms Chest Gs
230 x 9 x 28	510.8	48.2
300 x12 x 37	241.9	40.7
350 x 22 x 71	232.0	41.2

TABLE 5. Model Results for 100% (Full) Frontal Impact with a Rigid Barrier (Mid-Sized Car, 56 km/h, Belted, Mid-Sized Male Hybrid III)

Inflator	HIC36	3ms Chest Gs
230 x 9 x 28	842.6	49.3
300 x 12 x 37	758.6	47.6
350 x 22 x 71	692.3	44.4

TABLE 6. Model Results for 40% Frontal Impact with a Deformable Barrier (Mid-Sized Car, 60 km/h, Belted, Mid-Sized Male Hybrid III)

Inflator	HIC36	3ms Chest Gs
230 x 9 x 28	423.9	33.9
300 x 12 x 37	282.8	29.4
350 x22 x 71	309.3	30.5

Whereas airbag design is dictated by the unbelted testing of FMVSS 208, accident data indicate that unbelted occupants have a greater risk of airbag contact injury than their belted counterparts. If airbags were designed for the belted occupants, the risk of "side effects" associated with both belted and unbelted occupant- airbag interactions would be substantially reduced.

Summary and Conclusions

It has been shown by means of a review of accident data and previous research and our own experimental and modeling studies that the energy of inflators should be reduced substantially from present levels to lessen the risk of minor-to-fatal airbag contact injuries. However, these lower-energy inflators will not allow car manufacturers to meet the unbelted FMVSS 208 requirements. A case has also been made that, by reducing the inflator energies, the effectiveness of airbags in real-world frontal impacts would be increased substantially by both reducing the risk of airbag contact injury while providing sufficient injury risk reduction from occupant impacts with stiffer components of the vehicle's interior. Consequently, the following conclusions relating to the factors affecting airbag risks can be made:

1. Current airbags designed to meet or exceed the FMVSS 208 requirements are effective in reducing the risk of serious injuries to belted occupants, but not as effective in reducing the risk of injuries and fatalities to unbelted occupants.

2. With respect to the proposed lower-energy designs, the current FMVSS 208-driven airbag designs increase the risk of minor-to-fatal airbag contact injuries to both belted and unbelted occupants (with unbelted occupants at greater risk than belted occupants).

3. Inflator energy currently needed for FMVSS 208 requirements should be reduced in order to lessen the risk of airbag contact injuries.

4. With these lower-energy inflators, the unbelted requirement of FMVSS 208 cannot be met with the statistical margin needed to ensure compliance. Hence, an amendment to the FMVSS 208 Regulation is required. This may be in the form of total elimination of unbelted testing, testing at lower speeds, or testing at severities more representative of real-world frontal collisions.

5. Modeling studies show that lower-energy inflators will reduce the risk of injury to belted and unbelted occupants for frontal impact severities that are more representative of real-world events.

6. By reducing the risk of airbag contact injuries (minor-to-fatal) in real world accidents, the effectiveness of airbags will accordingly be further enhanced.

Acknowledgments

The authors would like to thank the following people for their valued contributions: Roger Daniel, Jeffrey Nadeau, and Carolyn Ridley for the assistance in the generation of this document; Joe Marsh for information related to the airbag field performance; Matt Nieslu-chowski for information relative to smart systems; Chris O'Connor for his consultations regarding inflator tailoring; N. Ramanujam for his exacting efforts in conducting the related experiments; and Don Weerappuli for his considerable efforts in the related occupant modeling.

Appendix A

Maximum inflator thrust can be derived from traditional tank test results (by the average temperature method of [23] and application of continuous derivatives) as follows:

$$\dot{m}_{max}\sqrt{\gamma R T_{stag}} = m_{max}\frac{\dot{P}_{max}}{P_{max}}\sqrt{\gamma R\frac{P_{max}V_{tank}}{\gamma R m_{max}}} = \sqrt{\frac{m_{max}V_{tank}}{P_{max}}}\dot{P}_{max}$$

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THE VALIDATION OF THE EEVC FRONTAL IMPACT TEST PROCEDURE

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Paper Number 96-S3-O-28

ABSTRACT

The EEVC Working Group 11 proposed a new frontal impact test procedure, based on a partial overlap impact to a deformable faced barrier, at the 14th ESV Conference in 1994. This test procedure has been subject to a validation programme to evaluate the repeatability, reproducibility and the applicability of the test procedure to a range of passenger car types and sizes. It also considered the effect of an increase in the impact speed from 56km/h to 60km/h. As well as the European validation programme, parallel testing in support of the EEVC work has been performed in Australia, Canada, Japan and the USA.

This paper presents the results of the validation programme and the conclusions regarding the current recommendations for the EEVC front offset deformable impact test procedure.

INTRODUCTION

In 1990, the EEVC created a Working Group (WG11) with the objective of determining the most beneficial ways in which evaluation of the performance of vehicle in front impacts could be improved. It was concluded that modifications to the 'component' regulations were unlikely to produce a large effect. The greatest benefit was considered to be achievable through a new frontal impact test, more representative of the impact conditions of car-to-car front impacts. In the interests of improving the possibilities of future harmonisation of test procedures, the EEVC invited the participation of experts from the governments of the United States of America, Canada, Japan and Australia. In addition, experts from the automobile industries of Europe, the USA and Japan have provided advice to the Group.

The accident studies indicated the importance of intrusion in the production of fatal and serious injuries and demonstrated the importance of replicating, in the dynamic test, the dynamics of structural deformations occurring in accidents. It quickly became apparent that an offset impact into a deformable barrier greatly improved the replication of deformations in these accidents.

EEVC WG11 created a test programme designed to enable the development of a test procedure that would achieve the objectives. The test programme selected car-to-car impacts between three different vehicle models as the baseline and compared car to barrier impacts against these baseline results. The initial deformable barrier face was based on the mobile deformable barrier face used in FMVSS 214, which itself was based on impact research by NHTSA. Previous research had indicated that this was a reasonable representation of car-to-car impacts and that the principal effects were not too sensitive to the actual stiffness of the deformable element. Deformable barrier tests were performed at 40, 50 and 60 per cent overlap, 55, 60 and 65 km/h and with both the initial barrier face design and an alternative design with a second stiffer element behind the element used in the initial design. Additional tests with a further revised barrier face design, incorporating a wide bumper element ahead of the element used in the initial design, and tests at 50km/h were added following analysis of the results of the first phases of testing.

Working Group 11 proposed a new impact test procedure based on the result of 25 full scale impact tests performed in this EEVC-EC Research Programme and over 30 tests performed outside the EC Research programme by organisations collaborating in the EEVC work.

The main conclusions of the test programme, reported at the 14th ESV Conference(1), were that the test parameters which best replicated the baseline 50km/h 50 percent overlap car to car impact were as follows:

- the most appropriate design of deformable barrier consisted of a block of aluminium honeycomb of crush strength 50psi of depth 450mm with a smaller piece of 250psi honeycomb attached along the bottom edge of its front surface to act as a nominal 'bumper'. The barrier was mounted 200mm from the ground with its top surface at 850mm.
- the overlap which gave results most similar to a 50 percent, 50km/h car to car impact was 40 percent of the car's width.
- the test speed to replicate the 50km/h 50% overlap baseline test should be between 55km/h and 60km/h, but closer to the former. It was agreed that the most appropriate test speed to replicate the baseline tests was 56km/h.

However, the accident data showed that to address an adequate proportion of fatal and serious injuries, the test should replicate a car to car impact speed of 60km/h or greater. Initially, WG11 had included offset deformable barrier (ODB) tests at 65km/h. These tests had indicated that compliance with the test requirements might not initially be possible at such impact speeds. The recommendation of WG11 regarding speed was that it should initially be 56km/h but that it should rise to 60km/h or more once the manufacturers had become accustomed to the engineering involved in meeting the requirements.

Finally, EEVC recommended that, as the test procedure was based on a range of tests but using only three car models, the test procedure should be validated for a wider range of vehicle models.

VALIDATION PROGRAMME

An extensive programme was established to validate the test procedure. The five main aspects for evaluating the procedure were as follows.

Objectives.

Practicability This aspect examined whether there were physical problems in conducting the test according to the procedure or any shortcomings of the equipment needed which had not been shown up by the baseline programme. For example, where measurements were to be made, it was necessary to prove that they could be made accurately for all vehicle types. Similarly, the validation phase needed to ensure that problems did not exist with the deformable barrier face in tests with different vehicle categories.

Suitability of Performance Criteria For the first time in Europe, it was being proposed that injury criteria, measured on a dummy, should form the basis of a frontal impact test performance evaluation. Dummies have been used for many years in the United States and several of the injury parameters relating to the head and chest are well established. However, the procedure sought to encompass serious injuries to other body areas, most notably the lower legs, as accident studies are now demonstrating the importance of these injuries. Such injuries are disabling, expensive to treat and can have long-term effects. Less experience was available in Europe for the dummy injury criteria relating to these parts of the body compared with the upper body. To account for injuries in areas not encompassed by a single test using a single dummy size, EEVC proposed a limit on steering-wheel displacements and easy removal of the dummy after the test. The validation programme examined each of the proposed parameters in terms of the following.

a) that the parameter should be easily measurable using widely available equipment
b) that measurements of the parameter should be repeatable between nominally identical tests and
c) that the parameter should relate to some aspect of the car's behaviour.

Test Speed None of the three cars tested in the earlier programme had performed well at test speeds of 60km/h or greater and this had led to the proposal for a phased increase in the test speed. This would enable cars to meet the requirements at the introduction of the test whilst also covering a significant proportion of serious and fatal injuries in the longer term. The validation phase examined the performance of more recently designed vehicles. To evaluate whether the earlier concerns were valid also for more modern designs, tests were carried out both at 56km/h and 60km/h.

Repeatability The validation programme sought to establish the repeatability of tests using the offset deformable barrier. Three identical cars were tested at the same establishment and the results compared.

Reproducibility The validation programme examined the reproducibility of the test procedures by comparing the results of nominally identical tests conducted on the same model of car by different test-facilities.

Validation Tests

Table 1.
Validation Test Programme

	Car Model code	56 km/h	60 km/h
		Number of tests	
Repeatability	C	3 (C1,2,3)	
Reproducibility	C	2 (C4, 5) [C2]	
Small car	A	1 (A1)	1 (A2)
Small car	B	1(B1)	
Small car	L	1(L1)	
Medium car	C	1 (C7)	1 (C6)
Medium car	D	1(D1)	
Family car	E	1 (E1)	1 (E2)
Family car	F	1 (F1)	
Large car	G	1 (G1)	1 (G2)
People carrier	I	1 (I1)	
Off road	J	1 (J1)	
Minibus	K	1 (K1)	

Table 2.
Vehicle Models Tested

Model code	Description	Model Year	Airbag
A	Small Hatchback	1989	✓
B	Small Hatchback	1993	✗
C	Family Hatchback	1992	✓
D	Family Hatchback	1988	✗
E	Large Family Hatchback	1993	✓
F	Large Family Hatchback	1993	✗
G	Large Executive Saloon	1994	✓
I	Multi Purpose Vehicle	1990	✓
J	Off-Road Vehicle	1995	✗
K	Minibus	1992	✗
L	Small Hatchback	1986	✗

Table 1 shows the test performed within the Validation Programme and Table 2 described the characteristics of the vehicles used.

Barrier Face Design. At the end of the first phase of the WG11 programme, the agreed barrier design consisted of a main block honeycomb with a smaller piece of stiffer material attached to the front lower edge of this block to act as a nominal 'bumper' section. Before testing for the validation phase had begun, a problem experienced using this barrier design was reported to the Working Group which was seen to be due to the stiff longitudinal member of the car aligning with the upper part of the bumper section and the impact rotating the bumper horizontally rearward into the main honeycomb block with little or no bumper deformation. This caused the barrier face to be pushed downwards and the vehicle to be pushed upwards, overriding the bumper section. With the main honeycomb block dragged downwards, its stiffness in the impact direction was greatly reduced.

This phenomenon was replicated and overcome using a trolley with a simulated rigid bumper impacting the standard barrier design and modifications to it. It was found that, by introducing two horizontal slots across the whole width of the bumper splitting it into three equal parts, rotation of the whole bumper section was avoided. This revision to the deformable face was incorporated into the barrier design and was used throughout the Validation phase.

For the development phase, where different overlap extents were used, the width of the barrier was 1500mm. It was found that, in all tests, a large proportion of the barrier was undeformed after the impact and had effectively played no part in the test. The minimum width necessary to allow testing of 40 percent of the widest vehicles currently in production was calculated and it was decided to reduce the width of the barrier to 1000mm. in the Validation Programme

RESULTS

The results of the test programme are presented in Figures 1 - 19.

Practicability.

Barrier Face During the validation phase tests, it was found that some tall vehicles (J, K) impacted the rigid concrete block above the deformable face. This contact was not representative of an impact with a conventional car. To eliminate such effects, the barrier specification has been changed to include the requirement that no part of the vehicle should impact any structure at a height more than 75mm above the upper surface of the deformable face in the test. The 75mm was simply to allow for the mounting flange on the barrier. This would require the deformable face to be mounted away from the rigid block for some tests.

For the other aspects, the barrier face as then specified was found to perform satisfactorily. In some cases, some stiff members penetrated the deformable element. This was generally not considered to be of major importance because one of the main advantages of a deformable face is the removal of the initial very high inertial force generated when the stiffer members of the car structure impact a rigid wall. The deformable face achieves this very successfully. With the modifications proposed, the barrier face design is considered to be satisfactory and the only validated design available for offset frontal impact testing. Tests by several institutes reported to WG11 show that the effect of further changes to the barrier design will be small in comparison to this change. Nevertheless, it is expected that research on the barrier face design will continue and the design may be revised to take into account future research findings on the requirements for improved compatibility.

Dummy Removal WG11 proposed that, after the test, the dummy should be capable of being removed without the use of tools and without adjusting the seat position. Furthermore, the dummy should not be broken and should still be within calibration and suitable for use in other tests.

It was found that the ease of dummy removal was subjective in the same way as the door-opening requirements i.e. it depended largely on the person involved. Moreover, in some countries, Health and Safety regulations limit the weight that an individual should be required to lift in the workplace. Nevertheless, it is recommended that this requirement for the dummy removal be retained, predominantly as a method of limiting intrusion at the facia and footwell levels. The use of equipment to support the weight of the dummy should be permitted.

The experience among the test laboratories participating in the discussions is that a dummy can sustain damage during a test which is undetectable in a visual examination but the dummy can still meet the performance corridors of its calibration requirements. Thus any obvious damage to the dummy in a test could have been precipitated by damage in previous tests and therefore cannot be an independent assessment of the severity of impact to the dummy in that test. For this reason it is recommended that the requirement for the dummy to be undamaged and fit for further use should be withdrawn.

Performance Criteria.

Head Injury Criteria. The Head Injury Criterion (HIC) is well established an no problems were experienced in recording and calculating this parameter. The 36ms value was selected in conformity with that specified for use in FMVSS208, although unlimited and 15ms values were recorded also. There was no case of a vehicle exceeding 1000 for HIC or HIC_{36} but meeting 700 or 1000 for HIC_{15} .

A presentation from Transport Canada to the working Group for a requirement for the peak resultant head acceleration not to exceed 80g was considered. The basis for this is that this limit has been found to give a reliable indication of rigid or hard impact by the head. All road accident studies indicate that head injury in the absence of hard head contact is almost unknown. It was felt that this did provide a useful additional protection criterion, but concern for spurious spikes in dummy instrumentation led to the introduction of a 3ms exceedence for this criterion. One vehicle exceeded this limit without exceeding 1000 for HIC_{36}. This vehicle was not equipped with an airbag.

Neck. Five neck injury criteria were included in the original proposals: Flexion and extension moments (peak values) and axial tension, compression and shear (time -duration limits). EEVC were asked to consider whether five parameters were necessary. The neck is a very complex structure which can fail through a number of mechanisms in vehicle accidents. However, it was recognised that, in a regulatory test it would be necessary to limit the requirements to those that are most likely to be relevant to the frontal impact situation. While EEVC WG11 feel that all five parameters are appropriate for scientific research, the recommended neck injury performance criteria for use in a regulatory test procedure are reduced to neck tension, neck shear and neck extension , considered to be most relevant to frontal impact testing with restrained occupants after consultations with SAE biomechanics experts.

All parameters were straightforward to record and compliance with the duration-exceedence limits proved to be simple to determine. The Working Group would like to see further research in the area of neck injury biomechanics to improve the confidence in the injury criteria. Nevertheless, these three parameter are considered to be the best available and to be suitable parameters for use in the test procedure.

Chest. Chest compression limits of 50mm and a Viscous Criterion value of 1m/s had been proposed. No problems were experienced in measuring and calculating these parameters. No vehicle exceeded either criterion in any test within the Validation Programme. It is recommended that they continue to be specified for use in the test procedure.

Abdomen. No requirement for the abdomen was specified in the original proposal as no suitable criterion could be found for use with the Hybrid III dummy. WG11 agrees that the compression and probably the rates of compression of the abdomen should be limited, but currently cannot recommend a satisfactory procedure for use in a regulatory test. It is recommended that this aspect of dummy design and instrumentation be addressed as soon as possible.

Femur. A force-duration exceedence curve was proposed. As with the neck force-time criteria, this was found to be easy and practical to measure and to determine compliance. no vehicle exceeded the criterion in any test in the Validation Programme. It is proposed to retain this criterion.

Knee Joint Translation. During the Validation programme, problems of binding were experienced with the original knee slider joints. Characteristic slip-stick responses were observed from the transducers. This will lead to an underestimate of the injury risk to the knee ligaments. This would be of greater concern if the results were likely to give an over estimate of the response. A replacement knee joint, using a roller bearing design, is now available and appears to have resolved this problem. Although this parameter was introduced to provide an assessment of the performance of knee bolsters used in US cars in association with airbags, it is considered to be a useful additional protection requirement for use with European cars to avoid potentially dangerous loading to this body area. It is proposed to retain this criterion used in association with the roller ball design of knee joint.

Tibia. Two criteria were proposed for use with the Hybrid-III dummy for tibia injury protection with the expectation that additionally there would be some measure of protection for the ankle joint: the peak axial compression and the tibia index..

The peak compressive force, set at 8kN based on the tolerance of the tibial condyles, was not exceeded in any test.

The performance limit for the tibia index was proposed to be 1.0. The validation programme tests demonstrated that this parameter was variable. It is not clear at present whether this is an innate feature of the parameter or whether vehicles do not perform consistently in this area, which is not currently the subject of a performance requirement. The variation of this parameter in the repeatability tests was examined and allowance for this was made in the revised proposed limit of 1.3

A problem with the foot design of the Hybrid III dummy became apparent during the evaluation of the Tibia Index results. Although the revised ankle design intended to give 45° dorsiflexion was specified and used in the Validation programme, this retained the metal to metal end stop to the articulation. This was demonstrated to give spikes in the transducer signals on occasions, leading to variation in the test results and artificially high readings. A revised foot and ankle has been designed by First Technology, in association with SAE, NHTSA and EEVC to resolve this problem. It is important that this foot and ankle be used when the tibia responses are required. WG11 have developed a certification procedure for this foot and tibia.

With the use of the new 45° foot and ankle with damped end stops, it is recommended that the tibia compressive force and the tibia index be used, with a critical value of 1.3 for the tibia index.

EEVC Working Group 12 will be evaluating new designs of leg for use with the Hybrid III dummy. If one of these proves to be significantly better than the existing leg in terms of biofidelity and injury detection, WG11 would like to see this new leg incorporated into the test procedure.

Steering Wheel Displacement. The original proposal included a limit on the rearward and upward displacement of the steering wheel hub centre and the upwards rotation of the steering column. The test programme demonstrated that, particularly with cars fitted with driver side airbags, the measurement of the displacement of the steering wheel hub was not practical. This was changed during the programme to the displacement of the top of the steering column. By removal of the steering wheel, this measurement proved to be easy and practicable. Examples were observed of gross steering wheel motion into the face or neck of the dummy but without exceeding the performance criteria, demonstrating the need for this additional requirement, especially as a single size dummy only is proposed for Europe.

It was originally considered that a limitation on the angular displacement of the steering column would provide an additional safety evaluation independent of the linear displacements. This measurement proved difficult to measure reliably in the Validation Programme and did not appear to provide any additional information. Therefore it is proposed to retain only the rearward and vertical displacement of the end of the steering column.

Footwell Intrusion. During the test programme, some concern was expressed at the large and high speed displacement of the brake pedal. As the dummy foot is placed on the accelerator (in harmony with FMVSS208), this would not be detected by dummy readings. A criterion based on brake pedal residual displacement was considered but it was felt that the correlation with injury mechanisms and risk was too tenuous for this to be included in the final proposals. No footwell intrusion requirements are proposed.

Dummy Condition An additional requirement proposed was that the dummies should be capable of being removed without tools and that they should not be broken in the test and should still be within calibration and suitable for use in other tests. The former requirement has been discussed above.

Since a dummy can sustain damage during one test which is undetectable in a visual examination and the dummy can still meet the performance corridors of its calibration requirements in preparation for the subsequent test, it is not an appropriate condition for use as an approval criterion. Therefore it is recommended that the requirement for the dummy to be undamaged and fit for further use should be withdrawn.

Test Speed. The first eight figures show the results for 11 different vehicles tested at 56km/h for HIC, Head Acceleration (3 ms exceedence), Neck extension moment, Chest compression, V*C, Peak femur force, Peak tibia compressive force and Tibia Index.

Table 3
Proportion of Models Tested Meeting the Proposed Performance Criteria

Performance Criterion	56km/h		60km/h	
	No. within limit	Models not in limit	No. within limit	Models not in limit
HIC (36)	10/11	J	4/4	
Head 3ms g	9/11	J B	4/4	
Neck Tension.	8/10	J B	4/4	
Neck Shear	11/11		4/4	
Neck extension.	11/11		4/4	
Chest comp.	10/10		4/4	
V*C	10/10		4/4	
Femur Force	11/11		4/4	
Knee slide	11/11		4/4	
TI (1.3)	11/11		1/4	C E G
TI (1.0)	8/11	E J L	1/4	C E G
Tibia Compress. Force	11/11		4/4	
All dummy criteria (TI 1.3)	8/10	J B	1/4	C E G
[All dummy criteria (TI 1.0)	7/11	B E L J	1/4	C E G]
Steering col; Vertical	8/11	B G I	3/4	G
Steering Col; Horizontal	8/11	D J L	3/4	A
All geometric.	5/11	B D G I J L	2/4	A G
ALL Crit (TI 1.3)	4/10	B D G I J L	0/4	A C E G

All of the vehicles met the proposed dummy performance criteria requirements except for the Off-road vehicle, which exceeded the HIC limit, the head acceleration (3ms) limit and the neck tension limit and one small car (without airbag), which exceeded the head acceleration (3ms) limit and the neck tension limit. One small car, the off-road vehicle and one of the family cars (just) would have failed the Tibia Index Limit at the original value of 1.0. In all these cases, the kinematics and loading of the dummy, judged from examination of the high speed film and post impact vehicle condition, gave support to the high values for these parameters.

The following six figures (Figures 9, 10, 11, 12, 13 and 14) give a comparison between the results for the four models of car tested at 56km/h and 60km/h. HIC, Head Acceleration, Chest compression, Femur force and Tibia Index and steering wheel displacement are shown as examples. It can be seen that the effects of testing at the higher speed on the injury parameters are relatively small, keeping them within the proposed limits, except for the Tibia Index. Testing at the higher speed has increased the displacement of at least one dimension of the steering wheel. For these four models at least, testing at 60km/h does not appear to be as severe as the test results from the first test series, although attention would need to be paid to the footwell area and to steering wheel displacement at the higher speed.

The performances of the vehicles tested in the validation programme were significantly better than those of the older car models tested in the development programme. In the impact test development programme, the performance of the cars was such that compliance at 60km/h or higher was not thought possible. Table 3 shows the number of vehicle models meeting the proposed criteria at 56km/h and 60km/h..

At 56km/h, 8 of 10 vehicle models were within all of the dummy-based criteria while at 60km/h all four vehicle models (all equipped with driver side airbags) met the proposed dummy based criteria with the exception of the tibia index. It is , perhaps, not surprising that the tibia index criterion should be exceeded since this area of the vehicle's performance is not currently subject to any legislative test requirements and this test procedure reproduces the conditions leading to tibia injury more accurately than the current perpendicular rigid wall test.

The proposed limits on steering wheel displacement were met by 5 of 11 cars at 56km/h although the individual requirements were met by 8 of the 11. For the four models tested both at 56km/h and at 60km/h, three of these were within the proposed limits at 56km/h, but only two were at 60km/h.

Bearing in mind the superior performance of these modern vehicles in comparison with those older designs tested in the development programme, a move to the higher impact speed indicated by the accident studies should be reconsidered in the future. EEVC Working Group 11 recommends that a test programme designed to compare barrier impacts to higher speed car-to-car impacts be performed to form a basis for a future test at an increased impact speed. Due regard should be taken of the implications for overall injury rate.

Repeatability and Reproducibility. Figures 15 ,16, 17, 18, 19 and 20 give an indication of repeatability and reproducibility. Six dummy responses are shown here for illustration. The first three bars show the results for the "medium" passenger car number 1, all tested at BASt. The results for these three repeat tests indicate good repeatability for these cars, especially for the upper body parameters. The variation of the leg parameters (Femur Force and Tibia Index) is more, as might be expected for this body region. It should be remembered that the Tibia Index for each dummy is the maximum value of the tibia index expression, irrespective of location and this may not be the same in each case.

The curves in figure 21 are the time histories of the head accelerations, chest accelerations and chest compressions seen in the three repeatability tests performed at BASt. The closeness of these lines indicates the good repeatability seen in these tests.

The fourth and fifth bars in figures 15 - 20 give the results for the same model of car tested at Fiat and TNO. They indicate the reproducibility of the test - or the variations found when the same vehicle model is tested at different establishments. Here again, the variations seen are considered acceptable, with the exception of the higher result for the Tibia Index seen in the test at Fiat. The reason for this odd result is not understood; the result at TNO being in good agreement with the results at BASt.

SUMMARY OF TEST PROCEDURE AND REQUIREMENTS

Impact Test Procedure.

The impact test should be an offset frontal impact into a fixed deformable barrier.

Barrier: The barrier should be a fixed barrier with a deformable face. The front face of the deformable element should be perpendicular to the direction of travel of the target vehicle. The design of the barrier face is given in Appendix A. The barrier face should be attached to the fixed block such that no part of the block or mounting surface greater than 75mm above the top surface of the deformable barrier face can contact the vehicle during the test.

Offset: The offset of the impact is defined by the percentage overlap on the vehicle front. The vehicle should impact the deformable face such that the barrier face overlaps the driver's side of the front of the car by 40 percent (± 50mm.) of the external width of the car at the widest point (excluding wheel trims and mirrors etc).

Dummies: The test should be performed with one 50th percentile Hybrid III dummy in driver's seating position and one 50th percentile Hybrid III in outer front seat passenger's seating position. The dummies should be equipped with the 45° 'damped end stop' ankle and subject to the foot and tibia certification tests. Each dummy should be clothed in standard clothing and should wear shoes. The specifications for the use of these dummies should follow FMVSS208. Fiftieth percentile adult dummies shall be placed in each of the other seating positions (except the centre front) unless these are fitted with 3-point seat belts to ECE Reg 16 or EC Directive 77/541.

Vehicle Condition: All seat and steering wheel adjustments should follow the FMVSS208 practice. Additional to the FMVSS specification, the dummy torso should be tilted forward and back twice after the seat belt has been attached to ensure a more realistic lie of the seat belt across the torso.

Impact Speed: The speed of the vehicle immediately before impact with the deformable barrier should be 56km/h. with a tolerance range of 2 km/h.

Performance Requirements.

Dummy Response Requirements.

Head. (i) The HIC_{36} shall not exceed 1000.
(ii) The resultant head acceleration shall not exceed 80g for more than 3 milliseconds calculated cumulatively. (This should not be applied to impacts which occur during the rebound phase)

Neck. (i) The neck tension and neck shear shall not exceed the criteria-duration limits given in figures B1 and B2 respectively (Appendix B).
(ii) The neck extension shall not exceed 57Nm.

WG11 recommends that neck flexion moment should be recorded in the test for future reference without applying a performance limit.

Chest. (i) The chest compression shall not exceed 50mm.
(ii) The Viscous Criterion shall not exceed 1.0m/s.

Femur. The femur force should not exceed the force-time performance criterion given in figure B3 (App. B)

Knee. The movement of the sliding knee joints shall not exceed 15mm.

Tibia. (i) The axial compression of the tibia should not exceed 8kN
(ii) Tibia Index ($TI = Mr/M_c + Fz/F_c$) should not exceed 1.3

where Mr is to be taken as a resultant of Mx and My,
M_c (critical bending moment,) = 225Nm and
F_c (critical compressive force) = 35.9kN.

The expression for TI should be calculated both at the top and the bottom of each tibia as a continuous time function. The Tibia Index is taken as the maximum value recorded during the time histories, irrespective of location.

Vehicle Response Requirements.

Steering Column. The residual displacement of the centre of the top of the steering column shall not exceed 80mm in the vertical direction nor 100mm in the rearward horizontal direction.

Dummy Extraction The dummy must be capable of being removed after impact without tools (except for equipment to support the weight of the dummy during removal) and without adjustment of the seat position.

Future Aspects.

The impact speed and many other aspects should be reviewed after a few years' experience has been gained in applying this test procedure.

Supplementary Requirements.

Although not necessarily an integral part of this test, it would appear to be appropriate to include requirements for fuel system integrity to avoid unnecessary duplication of standards and tests. It is recommended that consideration be given to the integrity of the firewall when reviewing the fuel system integrity test. In addition, there should be a requirement that the battery should not be ejected from the vehicle during the impact to avoid danger to other road users in a crash. Manufacturers should provide a mechanism for ensuring that fuel pumps are switched off at impact or when the engine stops. The battery may be dry during the test.

Additional Testing.

The full scale test evaluates a number of very important aspects of the injury risk to the vehicle occupants in a frontal impact. There are a number of aspects that cannot be assessed in this single test and which the EEVC WG11 feels need to be addressed.

Steering wheel impacts. Even if head or face to steering wheel contact does occur in the full scale test, the single test will evaluate only one single point impact of the wheel. In addition, the injury parameter measured on conventional dummies relates to brain injury rather than the facial skeleton injury that is common in face to steering wheel impacts. Accident studies clearly indicate a wide range of actual contact locations on the steering wheel. EEVC WG11 strongly advocates the use of an additional supplementary test to evaluate the facial and brain injury from steering wheel impact.

Seat and seat attachment. The strength of the seats and seat attachment cannot be fully addressed in this test. In particular, the effect on the dynamic performance of the seat, if it is possible to leave the adjuster out of engagement or partially engaged, needs to be considered by design requirements or a separate dynamic test. This can be even more important where one or both seat belt lower anchorages are attached to the front seats. The ability of the rear seat backs to withstand the impact forces of luggage was considered for incorporation in the full scale test, but it was decided that it would be simpler to evaluate this also in a separate test.

Seat belts and anchorages. Similar considerations led to the decision that the dynamic performance of an adjustable upper anchorage that could be left in an intermediate position would be better dealt with elsewhere. It was considered that it would be desirable to maintain a component test of the seat belt to enable simple and inexpensive routine testing for production conformity to take place. This would be necessary also for such aspects as durability and wear. The need for a requirement on anchorage strength would remain as the proposed test procedure would only assess anchorage strength up to the 50th percentile person at this impact severity.

APPENDIX A

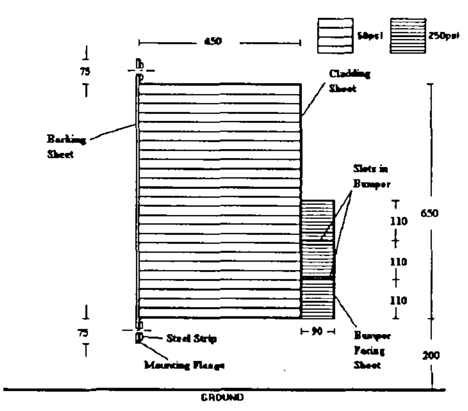

Figure A1 Deformable Barrier Design.

APPENDIX B

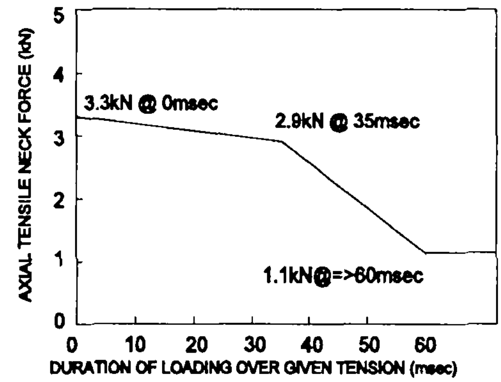

Figure B1. Neck Tensile Performance Limit

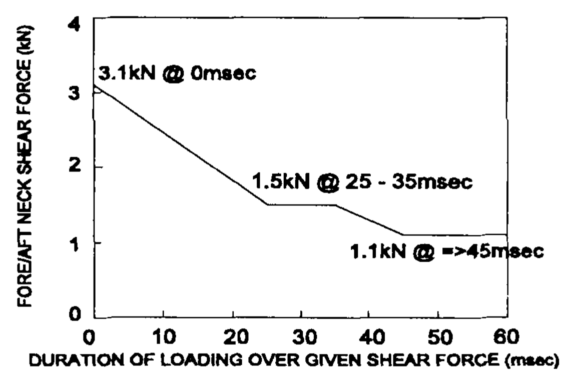

Figure B2. Neck Shear Performance Limit

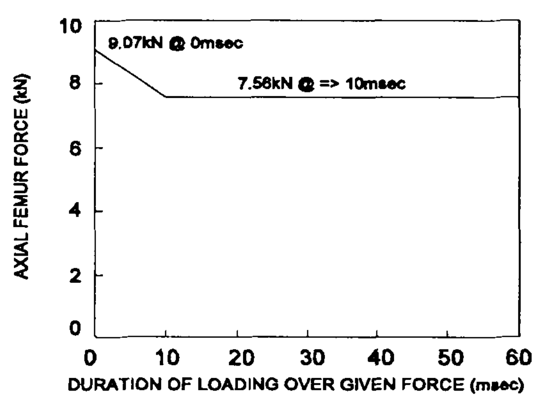

Figure B3. Femur Force Performance Limit

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D Cesari	INRETS	France	
E Faerber	BASt	Germany	
J Huibers	TNO	Netherlands	
J J Nieboer	TNO	Netherlands	
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E Fossat	Fiat	Italy
J Green	Rover	UK
R J Hitchcock	NHTSA	USA
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M Kitano	Nissan	Japan
Y Lambert	PSA	France
D Lehman	VW	Germany
A Lund	IIHS	USA
B Lundell	Volvo	Sweden
P Massaia	Fiat	Italy
K Matsumoto	MoT	Japan
M Miyakawa	JASIC	Japan
R Nagel	AAMA	USA
H Neff	Ford	Germany
R Nilsson	Volvo	Sweden
K Oki	JASIC	Japan
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K Seyer	Fed. Off. Road Safety	Australia
I Skogsmo	Volvo	Sweden
P V Skuse	Rover	UK
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T Takahashi	MoT	Japan
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C Thomas	APR	France
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Special presentations have been made by:-

T Hollowell	NHTSA	USA
H Norin	Volvo	Sweden
D Otte	Hannover Med. School	Germany
C Tarrière	Renault	France
P Thomas	ICE	UK
C Tingvall	Folksam	Sweden
U Westfal	ACEA	Europe
F Zeidler	Mercedes-Benz	Germany

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1.EEVC. EEVC Working Group 11 Report on the Development of a Frontal Impact Test Procedure. Proc. 14th Conference on the Enhanced Safety of Vehicles, Munich, May 1994. paper 94-S8-O-05.

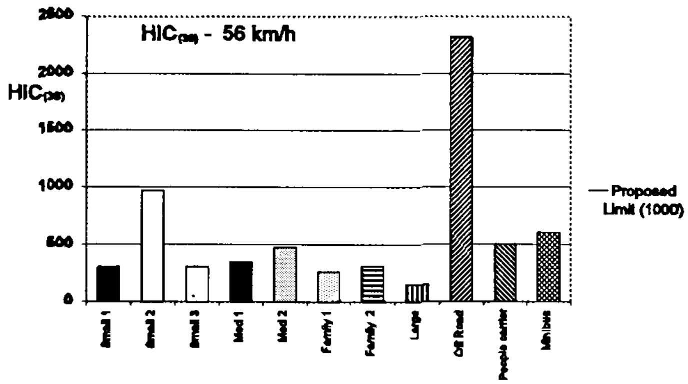

Figure 1. HIC_{36} measured in tests at 56km/h

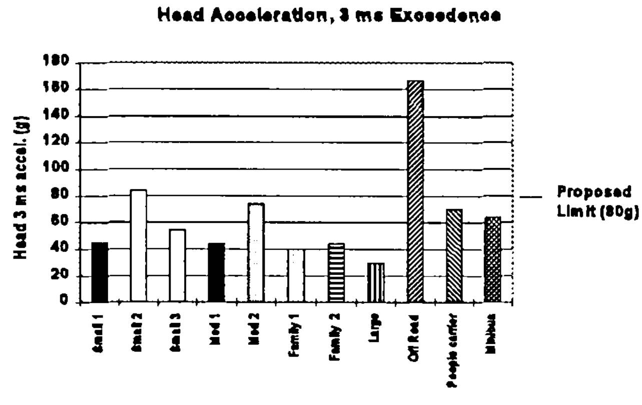

Figure 2. Head acceleration 3ms exceedence in tests at 56km/h.

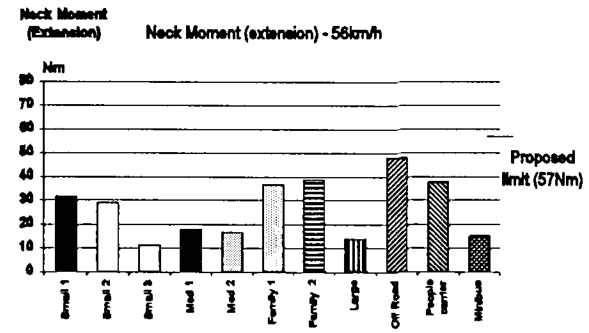

Figure 3. Neck extension moment in tests at 56km/h

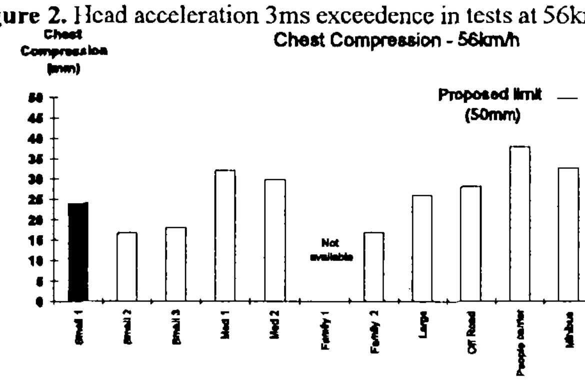

Figure 4. Peak chest compression in tests at 56km/h

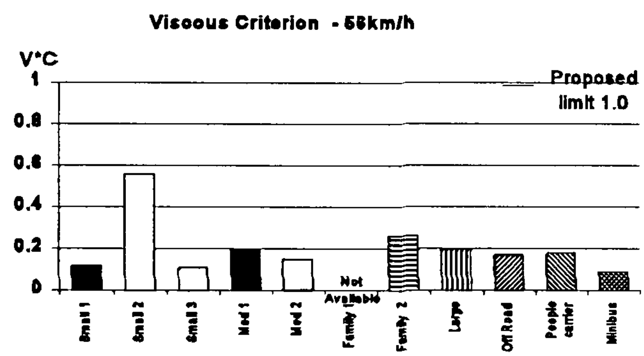

Figure 5. Viscous criterion (V*C) in tests at 56km/h

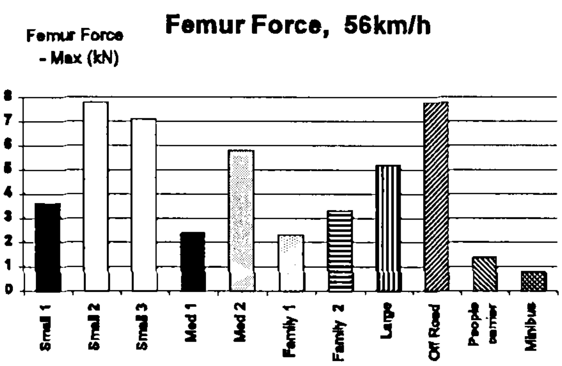

Figure 6. Peak femur force in tests at 56km/h.

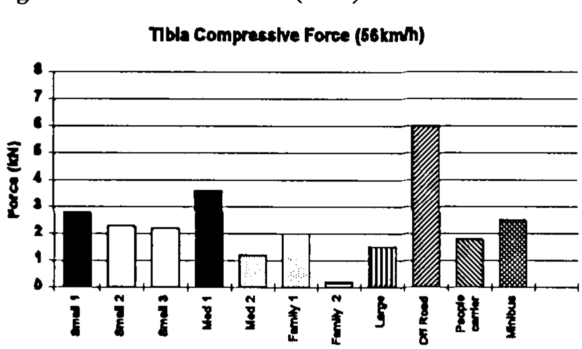

Figure 7. Tibia compressive force in tests at 56km/h.

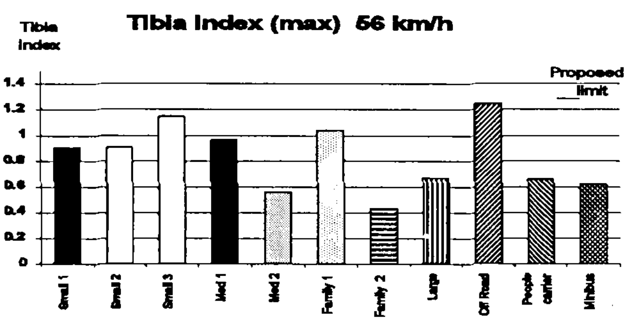

Figure 8. Tibia index in tests at 56km/h.

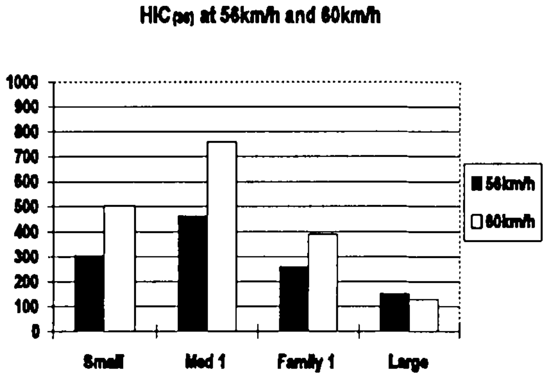

Figure 9. HIC_{36} measured at 56km/h and 60km/h

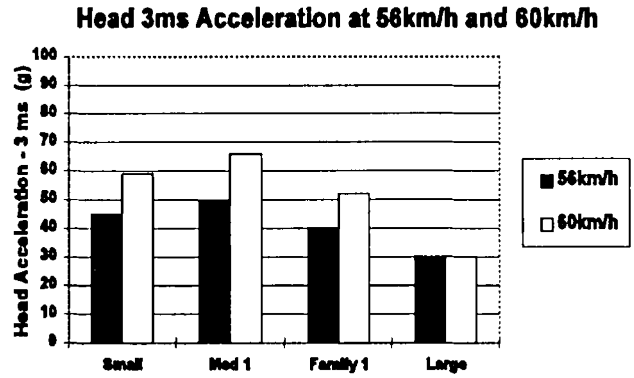

Figure 10. Head acceleration (3ms) measured at 56km/h and 60km/h

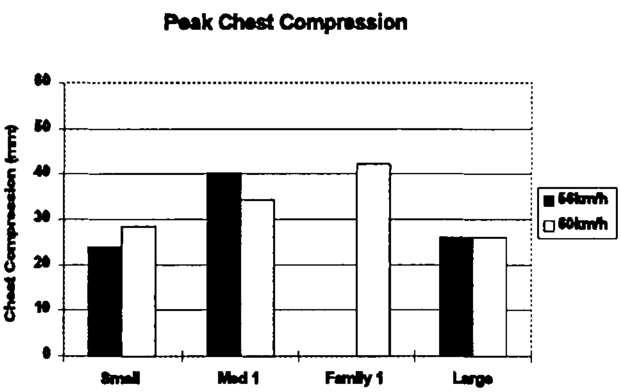

Figure 11. Peak chest compression measured at 56km/h and 60km/h

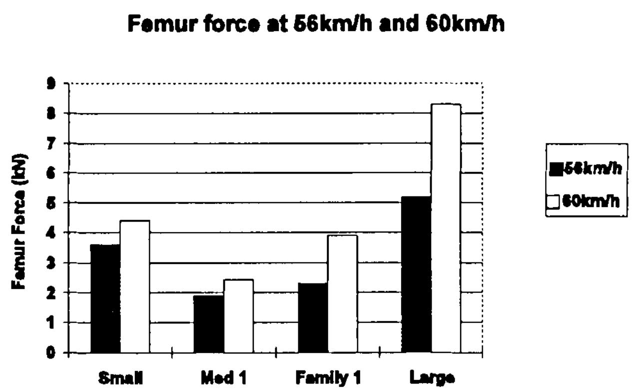

Figure 12. Peak femur force measured at 56km/h and 60km/h.

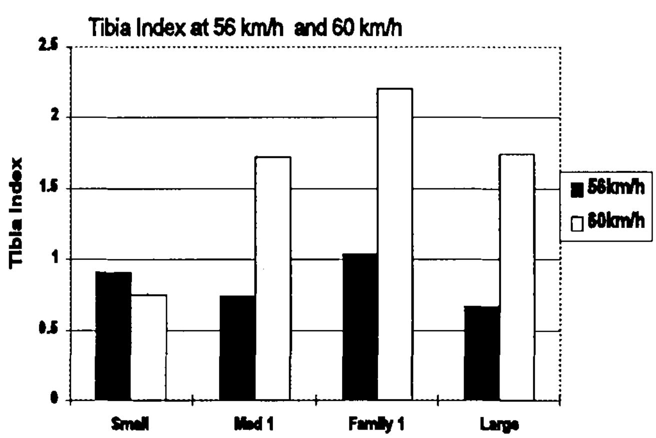

Figure 13. Tibia Index measured at 56km/h and 60km/h

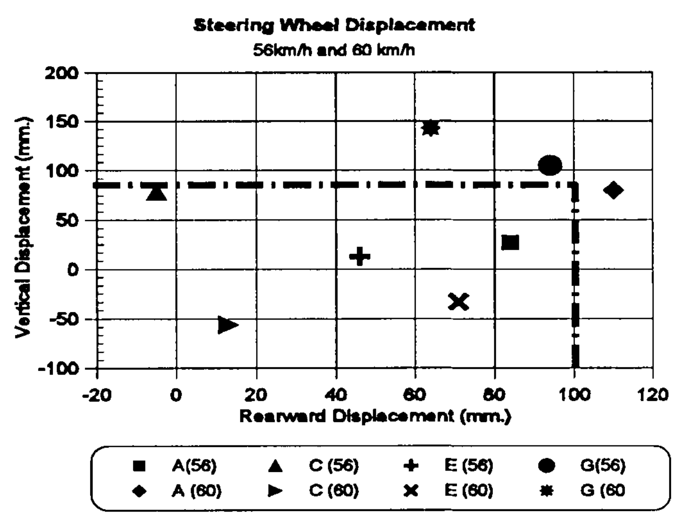

Figure 14. Vertical and horizontal steering wheel residual displacement measured at 56km/h and 60km/h

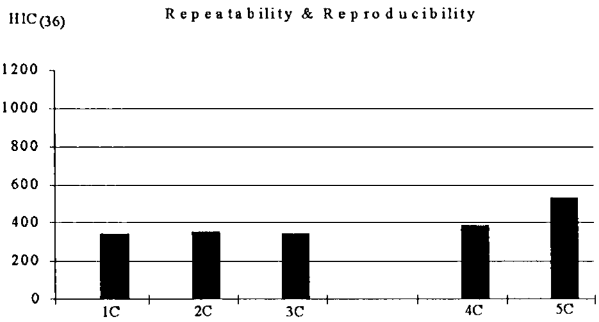

Figure 16. Repeatability and reproducibility of HIC_{36}

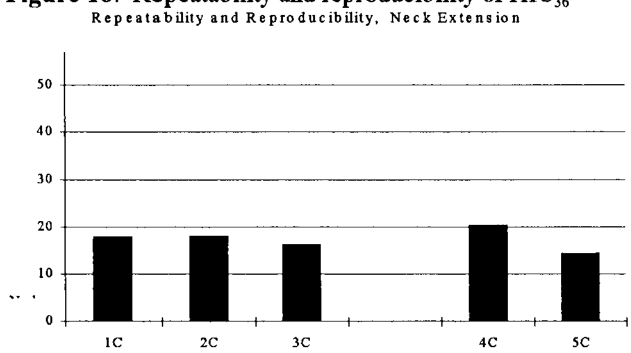

Figure 18 Repeatability and reproducibility of neck extension moment

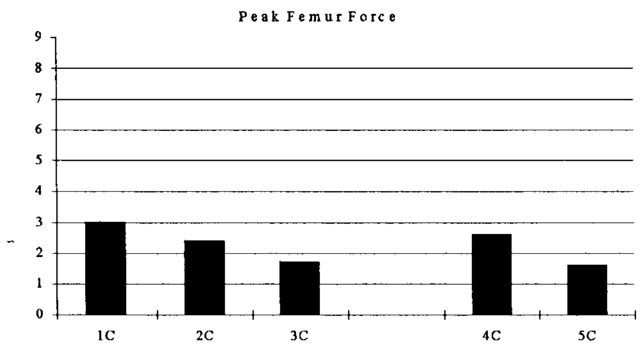

Figure 19. Repeatability and reproducibility of the femur force

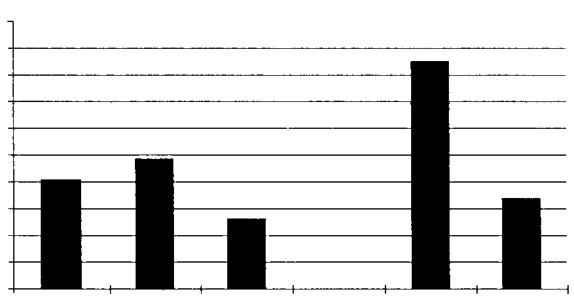

Figure 20. Repeatability and reproducibility of the tibia index.

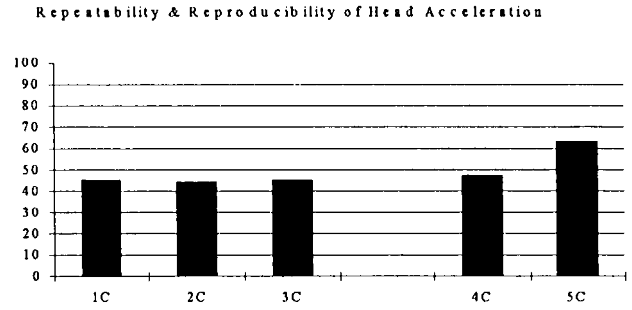

Figure 15. Repeatability and reproducibility of head acceleration.

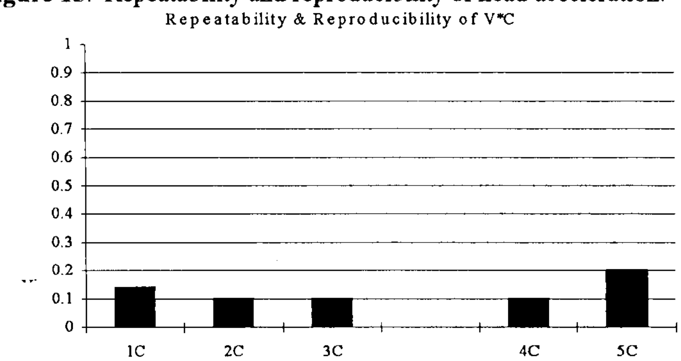

Figure 17 Repeatability and reproducibility of the viscous criterion

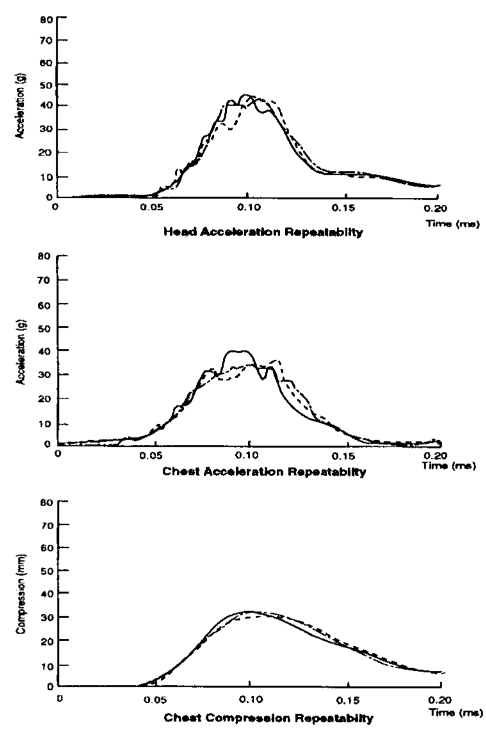

Figure 21 Repeatability of head acceleration, chest acceleration and chest compression, vehicles 1C, 2C and 3C.

LOWER EXTREMITY LOADS IN OFFSET FRONTAL CRASHES

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Paper Number 96-S3-O-04

ABSTRACT

To examine the relationship of occupant compartment intrusion and lower leg injuries, 17 new 1994-1996 midsize sedans representing 15 different models were crashed at 40 mi/h (64.4 km/h) into a stationary deformable barrier with 40 percent of the car's front width overlapping the barrier. A Hybrid III 50th percentile adult male dummy was positioned in the driver seat of each of these air bag equipped vehicles and restrained by the lap and shoulder belt. In addition to the usual head, chest, and femur instrumentation, the dummies were also equipped with instrumented lower legs and two accelerometers on each foot. Significant correlations with measures of intrusion were observed for all loads measured on the left leg. The right leg loads had fewer significant correlations, but the data from the distal tibia indicate a general trend of higher loads in crashes with more intrusion. Right proximal tibia bending moment and index were significantly correlated with maximum measures of intrusion, if not with the intrusion of specific nearby structures.

INTRODUCTION AND BACKGROUND

Lower extremity injuries are a frequent and costly consequence of motor vehicle crashes, especially frontal crashes. For occupants protected by seat belts, air bags, or both, the most frequently injured body region for front seat occupants in frontal crashes is the lower extremities. Injuries to the pelvis, femur, knee, lower leg and ankle/foot account for 24 percent of all injuries rated as moderate severity or greater on the Abbreviated Injury Scale (AIS≥ 2) for these vehicle occupants (Ore, Tanner, and States, 1993).

Rarely life threatening, only three of the lower extremity injuries listed in the AIS, 1990 Revision, are rated greater than AIS-3 (serious). Nevertheless, these injuries are often associated with long-term impairment and consequently high medical and rehabilitation costs. Miller et al. (1990) estimate that the medical and rehabilitation costs per case range from $584 (AIS-1) to $26,668 (AIS-4) when the most severe injury is to the lower extremity. Similarly, a 1989 study found that the average economic losses, which include medical expenses, wage loss, replacement services expenses and others, was $23,963 for fracture of a weight-bearing bone (All-Industry Research Advisory Council, 1989). A 1991 study of insurance benefits paid for catastrophic injuries (those with expected losses greater than $200,000) shows that in the worst cases the costs associated with lower extremity injuries can be staggering. The Insurance Research Council (1991) found that among injuries with large expected medical costs, those that involve fractures of a weight bearing bone were the third most costly at $222,600. Only brain injuries and paralysis had higher costs.

Analyses of data collected from motor vehicle crashes indicate that intrusion into the occupant compartment increases the risk and severity of lower extremity injuries. Pattimore et al. (1991) examined 263 footwell-related lower limb injuries and found a significant difference between the median level of footwell intrusion associated with injuries at the AIS-1 level (18 cm) and the median footwell intrusion for AIS > 1 injuries (31 cm). Although this suggests that intrusion is one of the factors that influences the severity of lower limb injuries, the authors noted that it is difficult to separate the effect of intrusion from the effect of crash energy, which has a well established influence on both injury severity and vehicle deformation. Using data from the National Accident Sampling System from the years 1988-1993, Crandall et al. (1995) showed that the ratio of AIS-3 injuries to all injuries was greater in crashes with more than 46 cm of intrusion than in crashes with less than 3 cm. The authors attempted to separate the effects of intrusion and crash energy by separating the data into four categories by velocity change (ΔV). Within each ΔV category, the risk of below-knee injury increased with higher levels of intrusion. Data in the United Kingdom's Co-operative Crash Injury Study (CCIS) have also shown that lower extremity injury risk is strongly influenced by intrusion into the occupant compartment (Thomas, Charles, and Fay, 1995). Lower limb injuries suffered by restrained front seat occupants in frontal crashes were examined using a multivariate logistic regression to untangle the effects of crash severity and intrusion. Their model showed that, while both footwell intrusion and ΔV were significantly related to the risk of AIS ≥ 2 injury, intrusion explained 5.2 times more variance than ΔV. Although earlier studies using field data from motor vehicle crashes could not satisfactorily separate the effects of intrusion and ΔV, the more recent studies have convincingly shown that intrusion into the occupant compartment by itself is associated with an increased risk of lower extremity injury.

To develop and evaluate appropriate injury countermeasures, crash tests that indicate relationships between intrusion and injury risk that are similar to those observed in the field data are needed. The Hybrid III dummies typically used in crash tests for research, product development, and regulatory purposes can be fitted with instrumented lower legs to measure the forces of interaction with the occupant compartment (Nyquist and Denton, 1994). Comparison of the force measurements with data from tests on biological specimens can be used to infer the risk of injury to the different parts of the lower extremity. Injury assessment reference values for different parts of the lower extremity anatomy have been published (Mertz, 1994).

Various studies of crash test data have reached different conclusions about the relationship between intrusion into the occupant compartment and loads measured on the legs of test dummies. Krueger et al. (1994) tested five different car models in 50 km/h, 40 percent overlap crashes with a rigid barrier. The dummies seated in the driver seat had accelerometers on the feet, and the authors suggested that there was a trend of higher foot accelerations in those tests with the largest footwell volume reductions. An examination of the data indicates that the relationship was not significant, but the authors remark on at least one potential confounding difference between the tests — despite similar driver seating positions, the different vehicle models' pedals were in different locations relative to the firewall.

In five different crash test modes with each of two different vehicle models (midsize and compact), Prasad and Smorgonsky (1995) found no obvious relationship between tibia loads and the magnitude of maximum intrusion in the dash/footwell area for either vehicle model. However, two of the five test configurations represent crashes of different severity than the other three. An examination of the car-to-car and rigid barrier tests, for which crash energies were comparable, also showed no clear relationship between tibia loads and intrusion except in the midsize car tests where the right distal tibia bending moments were correlated with the reported maximum intrusion. Kuppa and Sieveka (1995) examined data from seven different crash tests of the same model vehicle and found no correlation between the greater of floorpan and brake pedal intrusion and greater of right and left axial tibia forces. These tests also included acceleration measurements from the floorpan and brake pedal that were used to construct idealized acceleration pulses for these structures. The authors found that the maximum acceleration of the floorpan or brake pedal was highly correlated with maximum tibia axial loads.

Another examination of crash tests included data from 16 tests of the same model car (Zuby, Farmer, and Lund, 1995). Thirteen different configurations varying by offset and impact speed of both car-to-car and car-to-barrier tests were included and the data set produced observations from 22 different cars. Despite the large number of tests, not all of the analyses included large numbers of observations because of differences in dummy configuration among the tests. The analyses may also have been affected by the different histories of the test vehicles, whose model years ranged from 1984-1989. Nevertheless several significant correlations between dummy leg measurements and intrusion magnitude were observed. For example, upper tibia bending moments on both legs were correlated with intrusion measured at the lower instrument panel and distal tibia bending moments from the left leg were correlated with measured intrusion of the toepan. The results of the analysis were also consistent with the observations of other authors in that no correlation between longitudinal displacement of the toepan and axial forces on either tibia were observed.

DATA

The current analysis uses a test series that is more self-consistent than any other previous crash test analysis. All of the tests were approximately 40 mi/h (64.4 km/h) impacts against a deformable barrier with 40 percent overlap on the driver side. The deformable element is the same design as specified by the European Experimental Vehicle Commission Working Group 11 for future crashworthiness regulations in Europe, except that the barrier is 48 in. (122 cm) rather than 1 meter in width. Prior to the offset test, all but one of the test vehicles were subjected to a 5 mi/h (8 km/h) impact into a rigid 30 degree angled barrier on the passenger side. All structural damage sustained in the low-speed crash was repaired prior to the high speed test. Table 1 shows the vehicle model, test weight, impact speed and actual overlap for the data set.

A Hybrid III 50th percentile male crash dummy was seated in the driver seat of each test vehicle. Each dummy was fitted with instrumented lower legs that could measure the transverse bending moments in the anterior-posterior (A-P) and lateral-medial (L-M) planes at both the proximal and distal ends of the tibia tube. The axial force in the tibia, which was measured at the distal end of the tube, was also recorded. The ankle/foot in all tests was a design that has 45 degrees dorsiflexion range of motion, rather than the 30 degrees available in the standard Hybrid III ankle/foot (60 *Federal Register* 126). The feet were further modified so that two accelerometers could be attached to the internal sole plate to measure acceleration along A-P and inferior-superior (I-S) axes.

Sixteen of these tests were part of the Insurance Institute for Highway Safety Crashworthiness Evaluations (1995). One of the observations made during the series was that the data recorded from sensors in the leg frequently included high-frequency, large-amplitude oscillations characteristic of ringing that is initiated by forceful contact between adjacent metal components. It was hypothesized that the ringing arose when the ankle/foot was forcefully pushed to the extreme of its rotational range of motion. Because the recommended filter (SAE J211 CFC 600) for dummy leg measurements did not appear to ameliorate the effect of the vibrations, the data were adjusted by disregarding measurements recorded during the periods of vibration. The peak loads reported in the Crashworthiness Evaluation and in this analysis were the maximum of all measurements recorded outside the exclusion periods, which were identified using the foot accelerations where the oscillations were most readily observed. Figure 1 shows the foot acceleration and upper tibia bending moment measurements from a typical test and illustrates how the adjustments were made.

The dummy leg data included in the current analysis were the femur axial loads, vector sums of A-P and L-M bending moments for both the proximal and distal tibia, the Tibia Indexes calculated with the upper and lower bending moments, the axial force on the tibia, and the vector sum of the A-P and I-S foot accelerations. Table 2 shows the summary statistics for each of the dummy leg measures used in this analysis.

Table 1
Test Vehicles and Test Conditions

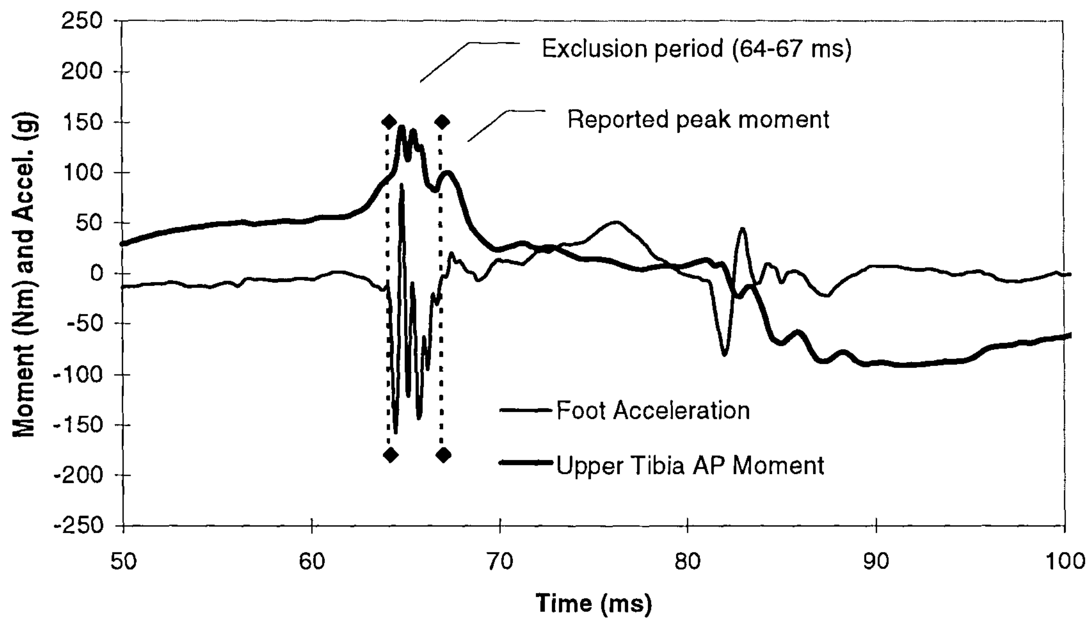
Vehicle Model	Test Weight (kg)	Impact Speed (mi/h)	Impact Speed (km/h)	Actual Overlap (percent)
Chevrolet Cavalier 1995	1362	39.7	63.9	42
Subaru Legacy 1995	1380	39.8	64.0	41
Ford Contour 1995	1431	40.0	64.4	40
Honda Accord 1995	1452	39.7	63.9	41
Mitsubishi Galant 1995	1459	40.0	64.4	41
Saab 900 1994	1476	40.1	64.5	40
Saab 900 1995	1489	39.9	64.2	40
Nissan Maxima 1995	1507	40.0	64.4	40
Toyota Camry 1995	1511	39.7	63.9	41
Chrysler Cirrus 1995	1553	40.1	64.5	41
Volvo 850 1995	1555	40.2	64.7	41
Volkswagen Passat 1995	1557	40.0	64.4	41
Toyota Avalon 1996	1584	39.9	64.2	40
Ford Taurus 1995	1565	40.0	64.4	41
Mazda Millenia 1995	1593	40.0	64.4	41
Chevrolet Lumina 1995	1645	40.0	64.4	40
Ford Taurus 1996	1651	39.3	63.2	40

Table 2
Summary of Dummy Leg Measures

Measurement	Minimum	Mean	Maximum
Left Femur Axial Force (kN)	2.2	4.5	8.3
Left Upper Tibia Bending Moment (Nm)	42	96	175
Left Upper Tibia Index	0.2	0.4	0.8
Left Lower Tibia Bending Moment (Nm)	33	258	463
Left Lower Tibia Index (Nm)	0.2	1.2	2.2
Left Tibia Axial Force (kN)	0.5	1.8	4.3
Left Foot Acceleration (g)*	39	72	120
Right Femur Axial Force (kN)	0.3	2.7	6.0
Right Upper Tibia Bending Moment (Nm)	64	106	190
Right Upper Tibia Index	0.3	0.5	0.8
Right Lower Tibia Bending Moment (Nm)	38	137	343
Right Lower Tibia Index (Nm)	0.2	0.7	1.6
Right Tibia Axial Force (kN)	0.8	2.8	6.6
Right Foot Acceleration (g)	41	76	167

*Not available for the 1994 Saab 900

Figure 1
Adjustment of Dummy Leg Measurements
To Account for Vibration Artifacts



The Tibia Indexes were calculated for coincident measurements using the following formula:

Tibia Index = Moment/225 Nm + |Axial Force/35.9 kN|,

where the Moment was the vector sum of either upper or lower A-P and L-M moments. The maximum Tibia Indexes are the maximum of the calculated data series.

Intrusion was characterized by comparing the precrash and postcrash locations of selected points in the occupant compartment. In this analysis, intrusion of the toepan, footrest, and lower instrument panel were examined. Two locations on the instrument panel and three on the toepan were established. The instrument panel locations were 15 cm on either side of the center of the steering column and 45 cm above the floor. The height off the floor roughly corresponds to the distance between the knee pivot and foot's sole on the Hybrid III 50th percentile male dummy. The toepan locations were at the height of the center of the brake pedal and 15 cm to the left of the brake pedal, directly in front of the brake pedal, and 15 cm to the right of the brake pedal. The origin for all measurements is located in a plane that passes through the front edges of both B-pillars and is vertically oriented before the crash. Consequently, occupant compartment deformation that occurs behind the front seating area is not included in the measurements of intrusion presented in this analysis. Table 3 shows the summary statistics for various intrusion measures.

Table 3
Summary of Intrusion Measures — Longitudinal Displacement (cm)

Measurement Location	Minimum	Mean	Maximum
Footrest	7	18	34
Left Toepan	9	24	39
Center Toepan	13	25	39
Right Toepan	12	21	35
Left Lower Instrument Panel	4	10	21
Right Lower Instrument Panel	3	9	20

RESULTS

The loads on the left leg were strongly influenced by the magnitude of intrusion. They not only correlated with the general deformation of the occupant compartment but also with the intrusion of specific nearby structures. Intrusion of the footrest, which is where the left foot was placed before the crash, was positively correlated with all of the left leg load measurements. Similarly, intrusion of the left lower instrument panel was positively correlated with the left axial femur load, upper tibia bending moment, and upper tibia index. For most of the vehicles in this data set, the precrash position of the dummy's left knee was almost directly behind the lower instrument panel measurement location. Table 4 gives the correlations between instrument panel, footrest, and toepan intrusion and the axial femur forces, upper tibia bending moments, upper tibia index, lower tibia bending moments, lower tibia index, axial tibia forces, and foot accelerations for the left leg. All correlations shown in bold *italic* type were statistically significant ($p < 0.10$).

Rearward displacement of specific nearby structures was not significantly correlated with the magnitude of loads acting on the dummy's right leg. General deformation of the occupant compartment did however influence right leg loads. The correlations between the upper tibia index and the maximum rearward displacements of the instrument panel and toepan were significant ($p < 0.10$). The correlation between upper tibia bending moment and maximum toepan intrusion was also significant ($p < 0.10$). And although not significant, the relationships between toepan intrusion and lower tibia bending moments, tibia index, and axial force indicate a consistent positive correlation. The correlation coefficients involving the right femur were near zero and in some cases negative. Table 5 gives the correlations for axial femur forces, upper tibia bending moments, upper tibia index, lower tibia bending moments, lower tibia index, the axial tibia forces and foot accelerations for the right leg. All correlations shown in bold italic type were statistically significant ($p < 0.10$).

Krueger et al. (1994) suggested that the initial relationship between the foot's position on the accelerator pedal and the relative location of the toepan, or firewall, might influence the relationship between measured leg loads and measured intrusion. In the current data set, the dummy's right foot was always placed on the undepressed accelerator pedal, and the distance between the accelerator pedal and the toepan was measured. By subtracting this measurement from the rearward displacement of the right toepan target, only deformation that moved the toepan rearward of the foot's initial position is counted as

Table 4
Correlation of Left Leg Loads with Measures of Occupant Compartment Intrusion

	Rearward Displacement			
	Instrument Panel			
Dummy Leg Measurements	Left Lower	Maximum Lower	Footrest	Maximum Toepan
Femur Axial Force	***0.43***	0.36	***0.47***	0.28
Upper Tibia Bending Moment	***0.56***	***0.52***	***0.57***	***0.56***
Upper Tibia Index	***0.62***	***0.57***	***0.60***	***0.57***
Lower Tibia Bending Moment			***0.53***	***0.68***
Lower Tibia Index			***0.53***	***0.68***
Tibia Axial Force			***0.58***	***0.55***
Foot Acceleration			0.29	0.32

Table 5
Correlation of Right Leg Loads with Measures of Occupant Compartment Intrusion

	Rearward Displacement			
	Lower Instrument Panel		Toepan	
Dummy Leg Measurements	Right	Maximum	Right Instrusion Target	Maximum
Femur Axial Force	-0.28	-0.24	-0.07	0.04
Upper Tibia Bending Moment	0.19	0.37	0.26	***0.52***
Upper Tibia Index	0.40	***0.56***	0.35	***0.64***
Lower Tibia Bending Moment			0.37	0.35
Lower Tibia Index			0.41	0.39
Tibia Axial Force			0.35	0.41
Foot Acceleration			-0.06	0.28

intrusion. If the foot's initial position relative to the intruding structure changes the consequence of a given level of intrusion, then this adjustment should be reflected in the correlation analysis. The lower instrument panel intrusion measurements were adjusted in a similar manner using the knee-to-dash dummy clearance measurements. Accelerator pedal-to-toepan measures ranged from 9-15 cm. Knee-to-dash ranged 16-28 cm (left side) and 16-26 cm (right side). The correlations between these adjusted intrusion measurements and the various leg loads are shown in Table 6. Comparison of these results to the results for lower instrument panel and right toepan in Tables 4 and 5 shows that these adjustments had little effect on the correlations for both legs. Correlations that were significant without subtracting the initial position remained significant after subtracting it, and none of the nonsignificant correlations were strengthened by the adjustment.

Foot accelerations did not correlate with any of the intrusion measurements. However, the intrusion data also include measurements of the lateral and vertical displacements of the toepan targets. Table 7 shows the summary statistics for lateral intrusion measurements from the toepan, and the correlations with both foot accelerations are shown in Table 8. The most significant correlations ($p < 0.10$) for each foot are the lateral displacements of the parts of the toepan closest to the given foot's precrash position, while the lateral intrusion measurements from the furthest portions of the toepan are not significantly correlated to foot accelerations. This observation supports the intuitive notion that lower extremity loads are affected by the deformation of those structures that the extremity is most likely to contact during a crash. In addition, the positive correlations for the right foot indicate that displacing the toepan and adjacent structures to the left increases right foot accelerations, and left foot's negative correlations indicate that displacement of the footrest to the right increases left foot accelerations. In both cases, displacement of structures toward the dummy's feet, or a narrowing of the footwell area, results in higher loads. Given the strong correlations with lateral displacement of the toepan, it is curious that the foot accelerations were not significantly correlated with any of the longitudinal measures presented in Tables 4-6. However, the correlation between maximum rearward displacement of the toepan and the average of the left and right foot accelerations is significant ($r = 0.48$, $p < 0.10$), suggesting rearward displacement of the toepan also influences foot accelerations as it does other loads measured on the dummy's legs.

Table 6
Correlation of Leg Loads and Intrusion Measures Adjusted for the Initial position of the Leg Relative to the Vehicle Interior

Dummy Leg Measurements	Rearward Displacement: Instrument Panel Knee-to-Dash Clearance		Toepan Accelerator Pedal-to-Toepan Distance
	Left	Right	Right
Femur Axial Force	***0.50***	-0.19	-0.17
Upper Tibia Bending Moment	***0.49***	0.09	0.20
Upper Tibia Index	***0.57***	0.38	0.30
Lower Tibia Bending Moment			0.23
Lower Tibia Index			0.28
Tibia Axial Force			0.36
Foot Acceleration			-0.03

Note: Correlations shown in bold italics are statistically significant ($p < 0.10$).

Table 7
Summary Statistics for Lateral Measures of Intrusion (cm)

Measurement Location	Minimum	Mean	Maximum
Footrest	-6	2	10
Left Toepan	-9	1	10
Center Toepan	-8	1	11
Right Toepan	-8	2	15

Table 8
Correlation of Foot Accelerations and Lateral Displacement Of the Toepan and Footrest

Foot Acceleration	Footrest	Toepan: Left	Toepan: Center	Toepan: Right
Left	***-0.45***	***-0.49***	-0.36	-0.20
Right		0.24	***0.45***	***0.52***

Note: Correlations shown in bold italics are statistically significant ($p < 0.10$).

DISCUSSION

The foregoing analysis confirms that lower extremity loads measured by crash dummies in crash tests are strongly influenced by the magnitude of intrusion in the occupant compartment. Even axial tibia loads, which other researchers have indicated were not correlated with the extent of intrusion, exhibited significant correlations with the magnitude of toepan/footrest intrusion for the driver dummy's left leg. Correlations for the right leg measures were not significant but still exhibited trends of higher loads with more intrusion.

Loads experienced by the dummys' legs appear to be most strongly influenced by deformation of nearby structures. Rearward displacement of the footrest, which is where the dummy's left foot was initially placed, exhibited strong correlations with left leg loads. Also, foot accelerations were strongly correlated with lateral displacement of the toepan structures nearest a given foot. The correlations between foot acceleration and lateral displacement of the footwell were progressively weaker for structures further and further from a given foot's initial position.

Moving the driver and front passengers rearward, away from the intruding toepan and instrument panel, may seem to be a reasonable way to ameliorate the effects of a given level of intrusion. However, differences in the driver dummy's initial clearance from these structures among the different vehicles tested here did not appear to affect the reported relationships between intrusion and the loads measured by the dummy's legs. This suggests that if vehicle designs were changed in a way that moved the front occupants further from the toepan and instrument panel, the changes would have to provide much greater clearance than any of the tested designs.

Real world crashes indicate that greater occupant compartment deformation is associated with both higher severity and higher risk of lower extremity injuries. This analysis shows that crash tests with instrumented dummies give the same indications. Therefore, while better dummy leg designs may improve the understanding of lower limb injury mechanisms, the Hybrid III with instrumented lower legs can be used to develop and evaluate injury countermeasures

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FOOT AND LEG INJURIES IN FRONTAL CAR COLLISIONS

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Paper Number 96-S3-O-05

ABSTRACT

Attention to injuries to foot, ankle and tibia is becoming increasingly focused as safety improvements are made in other areas. As our knowledge increases, views concerning the factors that cause leg injuries, become more varied. This paper presents Volvo's view on the subject and focuses on four main factors: Geometry, Pedals, Acceleration and Intrusion. The risk of injury is believed to be an accumulation of these factors. In order to achieve significant improvements in the area of leg injuries, it is therefore necessary to address all factors.

INTRODUCTION

Overall occupant safety has improved significantly throughout the past 20 years. Leg injuries are however still frequent and when long term consequences are considered, injuries to the lower limb account for an important issue (Mc Kenzie; 1986). It is therefore important to understand the injury factors behind leg injuries, in order to know how to help reducing them.

This paper presents the main factors of foot injury in frontal impacts, namely:

- Geometry - local differences in height and width of the footwell.
- Acceleration - generation of contact forces between foot and surface through change in relative speed
- Pedals - design and behaviour
- Intrusion - in the footwell area

The factors are identified by presenting the statistic material available, discussing the results of an in-depth study of 20 injured ankles and thereafter linking this material together in a discussion, pointing out the main injury factors. Results from simulation and testing are presented, to strengthen the arguments.

Throughout this paper, the injuries are divided into foot, ankle and tibia injuries. The term leg injuries is used to, in one word, represent foot, ankle and tibia injuries.

LITERATURE SURVEY

The mechanisms of foot and ankle injuries are very complex. Several different injury factors have been identified in the literature, although there are few papers which cover more than one or two at a time. In addition, several different (and sometimes) contradictory injury mechanisms, have also been pointed out.

Mechanisms

In an extensive in-depth accident study, Fildes et al. (1995) identified the main injury mechanisms as compression of the leg, perpendicular load to the knee and crushing or twisting of the foot.

In another in-depth study, Lestina et al. (1992) found four common fracture mechanisms of the foot and ankle: inversion or eversion, direct vertical force, dorsiflexion and direct side force. The Lestina study confirmed the frequencies of the six injury mechanisms identified by Morgan et al. (1991), with the addition of identification of inversion or eversion as a prominent fracture mechanism.

In an accident study, Portier et al. (1993) identified two main mechanisms: forces acting under the metatarsal condyles, coupled with the inertial effect of a dorsiflexing foot, producing metatarsal fractures and eversion /inversion motions, caused by forces acting under the ball of the foot, producing malleolar fractures and ankle sprains.

Intrusion

The effect of intrusion has been studied by several authors. Many published studies (Gloyns et al. 1979, Portier et al. 1993, Pattimore et al. 1991, Otte et al. 1992 and more), suggest that intrusion, as well as delta-V, increase the risk of leg injuries. In a recently published study by Thomas (1995), the effect of intrusion could be separated from crash severity, stating that intrusion in the footwell increases the risk of leg injury to a greater extent than crash severity.

Pedals

Thomas (1995) also identified a higher risk of AIS2+ leg injuries to the driver than to front seat passenger. He suggests that this is mainly due to the pedals. In a computer simulation study, by Pilkey et al. (1994), a correlation was found between position of the foot on the brake pedal and the load transmitted to the heel of the braking foot, suggesting that minimised intrusion combined with a brake pedal position, which allows the heel to remain close to the toepan, would be the optimum way to limit impact on the foot.

Geometry

Otte et al. (1992) identified two characteristic mechanisms which should be regarded separately. Apart from the force mechanism, which always results from intrusion of the footwell, the study identified another mechanism, a simple supporting and slip-off mechanism of the feet, which may already occur in connection with less severe accidents, in which there is no intrusion of the footwell.

Acceleration

In a study by Crandall et al. (1995), it was found that 71% of injuries below the knee, sustained by front seat occupants in head-on collisions, occur with less than 3 cm of intrusion. The study pointed out that factors such as the vehicle's change in velocity and the rate and timing of intrusion, must be considered when examining injury mechanisms to the lower extremities.

Frampton et al. (1995) found lower limb injuries occurring under conditions of very little or no intrusion, suggesting padding in the footwell would diminish peak loads and thus, reduce injury more effectively than merely aiming to reduce intrusion.

ACCIDENT DATA

The accident data is an important source of information, for determining the injury factors. The acceleration, intrusion and geometry factors can be determined by investigating:

- Crash severity
- Exterior deformation of the car

Distribution of the impact area for driver and front seat passenger

The effect of the pedals can be determined by investigating:

- Use of brake pedal by the driver
- Distribution of injury to left and right leg
- Distribution of the impact area together with injuries to driver and front seat passenger

General Information

The following data is taken from Volvo's own data base, which contains accidents involving Volvo cars in Sweden. The present data base consists of about 25 000 accidents.

Out of the AIS2+ injuries involving belted driver in Volvo 700 and Volvo 900, about 20% are injuries to thigh and leg. The risk of sustaining long term consequences from these type of injuries, is 2.4% (Mixed model, Koch et al 1992), which indicates the significance of leg injuries.

The cases selected for the study were frontal impacts involving Volvo 200, 700 and 900 series cars, where the driver and the front seat passenger (when present) had been belted, totally 6040 accidents. From this selection, a subset of leg injuries where at least one of the occupants had sustained leg injuries of the type AIS2+, was used.

Injury Type

The database records contains fracture injuries of the leg, where the leg is divided in three parts; tibia/fibula, ankle and foot. The ankle injuries are defined as injuries to the talus, caleaneus and malleoleus. A leg injury relative frequency can be calculated from the material, according to the table below.

Table 1.
Leg Injuries where First Figure is Number of Cases and Second Figure is Relative Injury Frequency in % of All Selected Frontal Impacts in this Study.

	Tibia/Fibula	Ankle	Foot
Driver	33 / 0.5%	73 / 1.2%	56 / 0.9%
Pass	6 / 0.3%	13 / 0.6%	11 / 0.5%

The most frequently injured part of the leg is the ankle, which together with the foot represents approx. 80% of all injuries. The driver leg injury frequency (in percent), is about double that of the passenger.

Crash severity

The EBS (Equivalent Barrier Speed, Nilsson-Ehle et al. 1982) measurement was used as a measure of crash severity. With the help of a logistic regression, it can be shown that higher severity, gives a higher risk for leg injury (Figure 1).

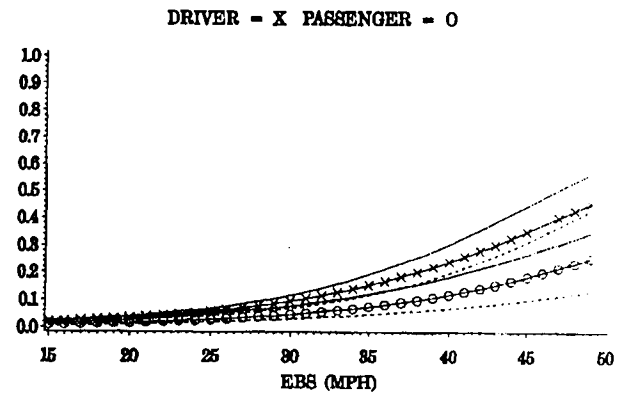


Figure 1. Leg Injury Risk for Driver and Front Seat Passenger as a Function of EBS. Risk Curves Surrounded by 90% Confidence Bands.

Figure 2 illustrates the EBS distribution between driver and front seat passenger.

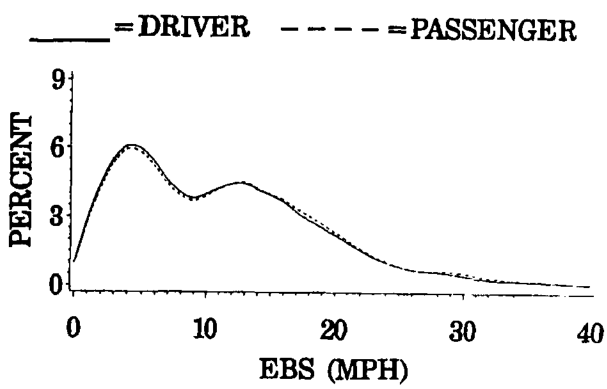


Figure 2. EBS Distributions for Driver and Front Seat Passenger in Frontal Impacts.

The small difference in EBS distribution can not explain the higher injury risk for the driver compared to the passenger. It is clear that increased severity (and thereby to some extent, increased acceleration), results in an increased risk of sustaining leg injury.

Deformation

Another way of quantifying the crash severity, is the deformation of the car. Volvo's database contains the exterior deformation of each car and with the help of laboratory data, it can be shown that intrusion in the footwell area requires an exterior deformation in excess of 50 cm.

A logistic regression, with 90 % confidence limits, shows that greater deformation depth of the car, gives a higher risk of leg injury (Figure 3). For a certain deformation of the car, the driver has a higher risk of leg injury, compared to the front seat passenger.

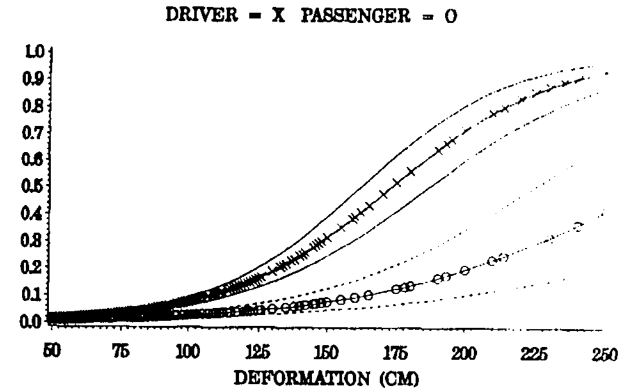


Figure 3. Leg Injury Risk for Driver and Front Seat Passenger as a Function of Exterior Deformation. Risk Curves Surrounded by 90% Confidence Bands.

Figure 4 does not illustrate any significant difference between driver and front seat passenger, regarding the distribution of the deformation.

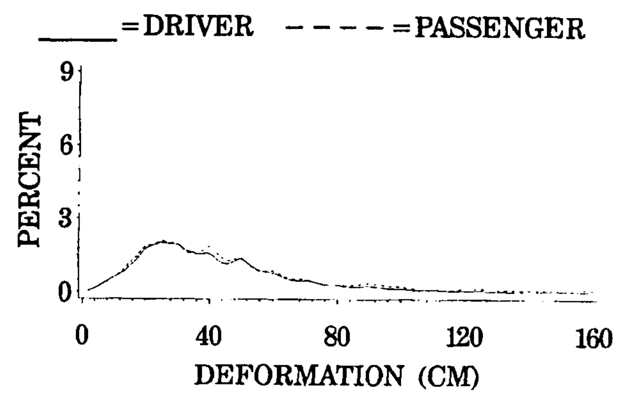


Figure 4. Exterior Deformation Depth Distributions for Driver and Front Seat Passenger in Frontal Impacts.

The higher risk of injury for the driver, compared to the passenger, can not be explained by the small difference in exterior deformation distribution. It is, however, clear that increased deformation (and thereby to some extent, increased footwell intrusion), results in an increased risk of sustaining leg injury.

Severity and Deformation

By plotting the injury cases against EBS and deformation, it is possible to distinguish, if intrusion is a significant parameter, or if there are occurrences without intrusion present. It is also possible to determine whether a higher degree of EBS or deformation, is required to produce a certain type of leg injury. Figures 5 and 6 illustrate the injuries to tibia, ankle and foot, for the driver and front seat passenger respectively, as a function of EBS and deformation of the car.

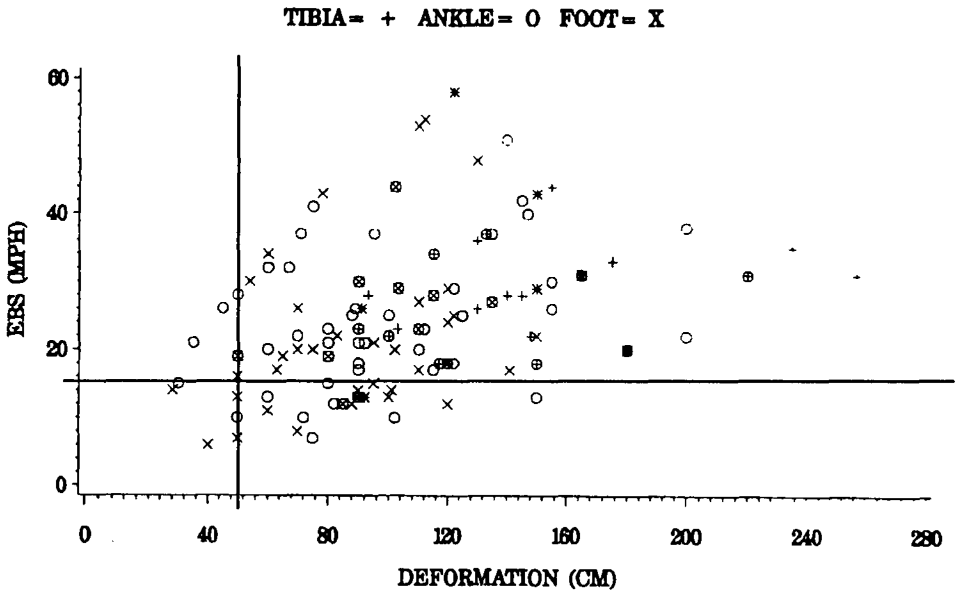


Figure 5. Leg Injuries to Driver.

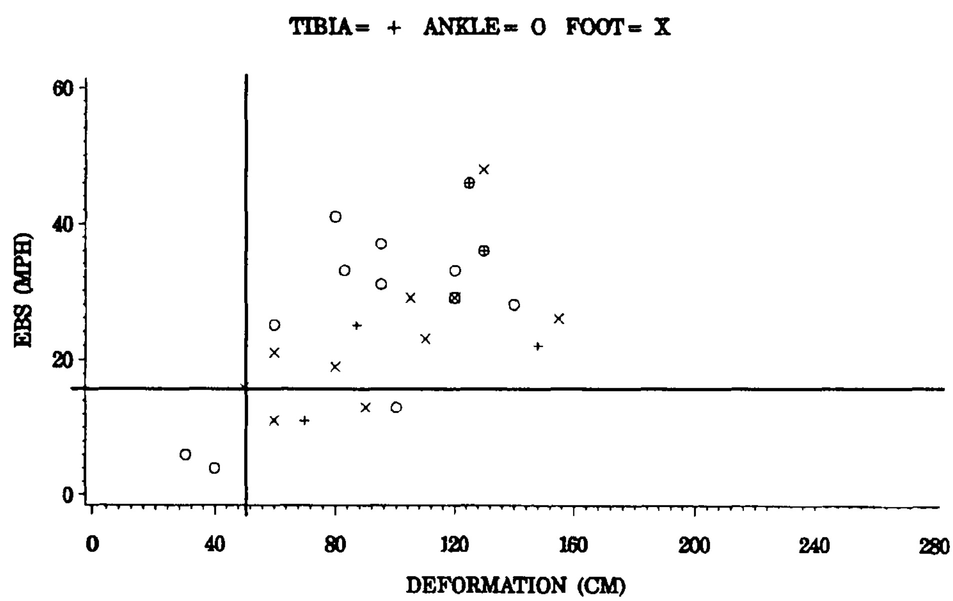


Figure 6. Leg Injuries to Front Seat Passenger.

It is interesting to note that approx. 10% of the cases are foot and ankle injuries where deformation is less than, or equal to 50 cm, i.e. no footwell intrusion is involved and severity is low (< 15 mph). These cases exist for both the driver as well as passenger. All fractures to the tibia have occurred at a deformation of 70 cm or greater and an EBS in excess of 12 mph.

Fractures to the foot and ankle, requires less deformation and EBS.

Higher severity, both in deformation and EBS, is required in order for the front seat passenger to sustain foot fractures.

Impact Area

The impact areas describe how much of the front of the car that has been engaged during the crash. The accidents are divided into three areas. Left offset is defined as all accidents involving less than 33% overlap on the left side. Right offset stands for the same definition on the right side. The remaining impacts, where the overlap is greater than 33% on any side, are defined as frontal impacts.

The percentage of offset impacts, on the left side, is greater for drivers whom have suffered leg injuries, than it is for all drivers in the selected material. See Figure 7.

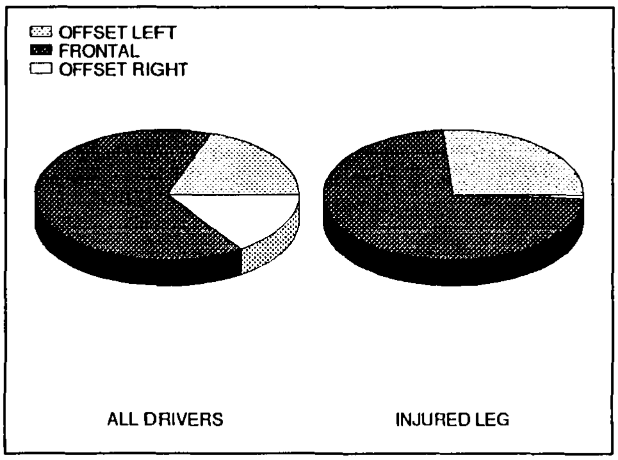


Figure 7. Distribution of Impact Areas for the Driver.

Comparing accidents involving front seat passenger leg injuries to accidents involving all passenger injuries, shows that frontal impacts are more common. It is also interesting to note that passenger leg injuries also have occurred, when the impact has been on the drivers' side.

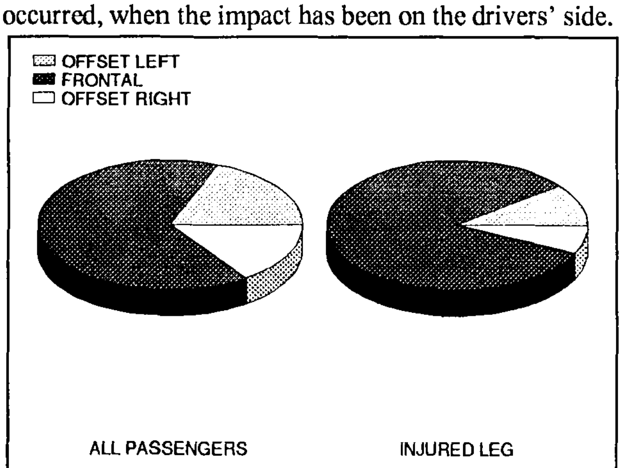


Figure 8. Distribution of Impact Areas for the Front Seat Passenger.

The injury frequencies, with respect to the three impact areas, are displayed in Figures 9-11.

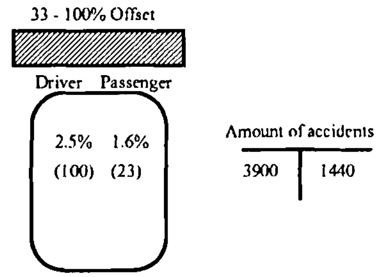


Figure 9. Leg Injury Frequency of Driver and Passenger at 33% -100% Overlap Frontal Accidents.

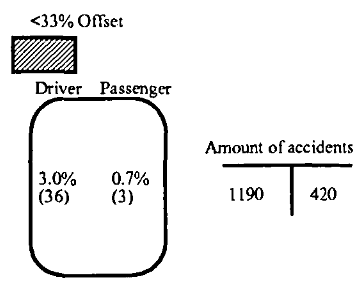


Figure 10. Leg Injury Frequency of Driver and Passenger at <33% Overlap Left Hand Offset Accidents.

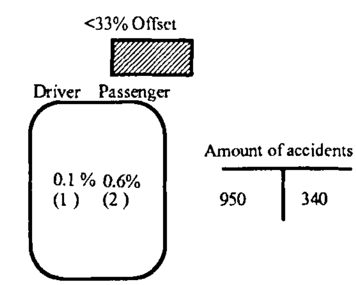


Figure 11. Leg Injury Frequency of Driver and Passenger at <33% Overlap Right Hand Offset Accidents.

Comparing similar impact situations for the driver and front seat passenger means comparing a left offset for the driver, with a right offset for the front seat passenger. There is a higher frequency of driver leg injuries in a left offset impact (3%), than there is for a front seat passenger in a right offset impact (0.6%).

The distribution of EBS and exterior deformation for left and right offset impacts and frontal collisions are displayed in Figures 12 and 13.

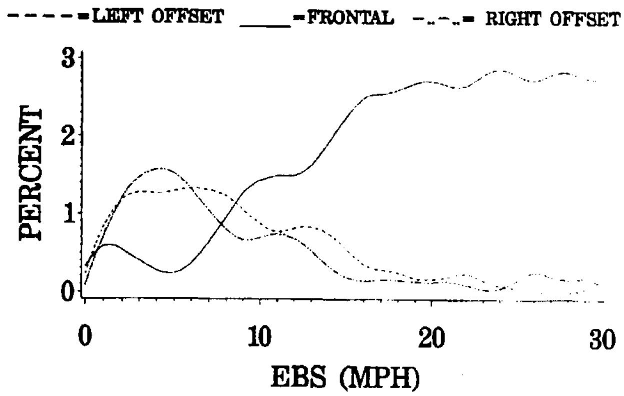


Figure 12. EBS Distribution for Left and Right Offset, and Frontal Impacts.

Offset collisions are more common at lower EBS, while frontal impacts tend to increase above 8 mph EBS.

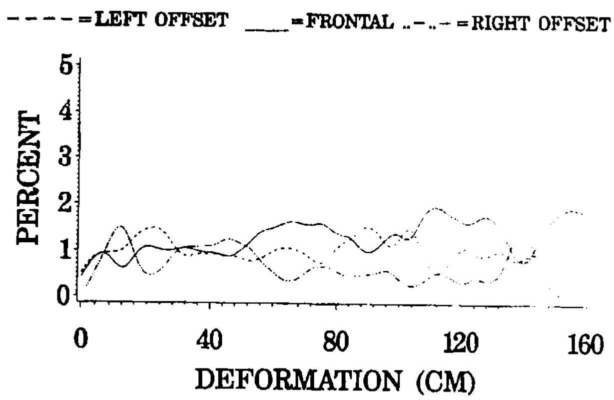


Figure 13. Exterior Deformation Distribution for Left and Right offset, and Frontal impacts.

There are no significant differences in the deformation distribution between the three impact types.

Brake Pedal Use

There is a separate indication in the Volvo database concerning use of the brake pedal. The distribution is displayed in Figure 14.

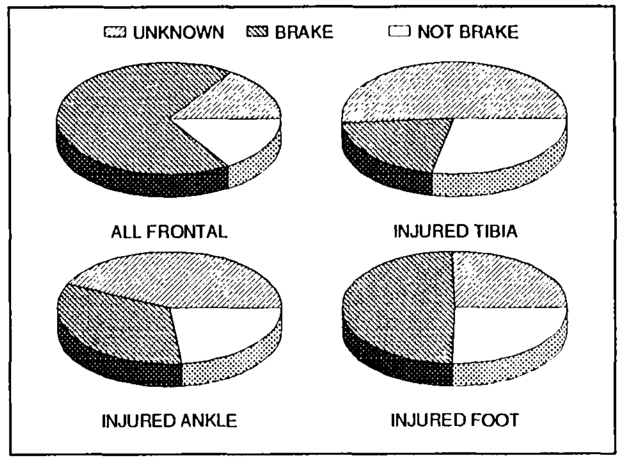


Figure 14. Distribution of Brake Pedal Use of the Driver in Frontal Collisions Including Driver Injury to Tibia, Ankle and Foot.

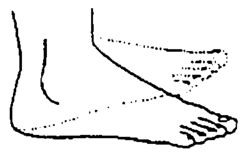
There is a higher frequency of "not brake" for those who have sustained injury to the leg, compared with all frontal accidents in this study. There is, however, a rather large portion of "unknown", making this parameter difficult to analyse.

INJURY TYPES

As a complement to the accident data, a more detailed study has been conducted on the different types of injuries, notably injuries to the ankle. As basis for the assumptions below, experiences from Volvo's accident research provide the major source of input. This input is complemented by results from externally published studies.

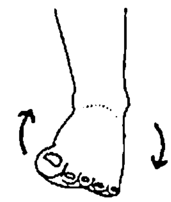
Foot Injuries

Foot injuries AIS2+, are mainly tarsal and metatarsal fractures. The fractures are probably the result of a dorsiflexion motion (Figure 15), induced by the body load, with or without combination of uneven support under the foot. There are also cases of foot injuries resulting from jamming under the pedals, e.g. case no. 10 in the in-depth ankle study described below.

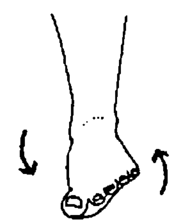


Dorsiflexion: an upward flexion of the foot.

Figure 15. Dorsiflexion.



Inversion: an inward rotation of the foot, with elevation of the medial edge.



Eversion: an outward rotation of the foot, with lowering of the medial edge.

Figure 16. Inversion and eversion.

Ankle Injuries

Ankle injuries account for the majority of AIS2+ leg injuries in Volvo's database.. The mechanisms of the ankle injuries, can be of many different types. To facilitate the understanding of the mechanisms and parameters which influence the occurrence of ankle injuries, an in-depth study was performed. This study involved 12 frontal collisions in Volvo cars (referred to as "cases"). Occupants had a total of 20 injured ankles of type AIS2+. These injuries were analysed in detail. The cases were chosen to be representative of the ankle injuries in the data and valid for the study of injury mechanisms. Each of the cases had detailed descriptions of the vehicle and the accident, as well as medical records including x-rays of the occupants. The cases were studied in a group, consisting of orthopedic experts and car crash experts. The goal was to clarify the motion of the foot and leg and to identify the cause of injury for each specific case. The cases are presented in Appendix 1.

Inversion or eversion of the foot with varying degrees of axial load through the ankle and tibia, accounted for the major injury type (Figure 16). There were 6 cases in the study, in which there was little or no intrusion of the footwell, low crash severity, but with injuries resulting from this mechanism. The bending of the foot at impact, as well as the design of the footwell area, including the pedals, were identified as the important injury factors.

Footwell intrusion and the intrusion of the pedals increases the relative foot impact speed. This often results in a more complex fracture picture (cases no. 2 and 5). If the foot is angled at impact or placed without support under the whole foot, the tolerances for fracture are very low and the inertia body load of the occupant, due to the crash, is enough to sustain a fracture (case no. 12).

Pilon Fracture - Another frequent characteristic ankle injury type in the in-depth study, was a fracture to the neck of the Talus (collum tali), also called "Pilon fracture" (Peterson 1974). This injury is a typical "braking injury" (cases 8 and 11a), but can also occur without brake pedal involvement, as in case no. 10. This fracture will occur *if* the foot is subjected to a local impact, slightly in front of the forces acting through the tibia *and* the ankle muscles are tensed, as shown in Figure 17.

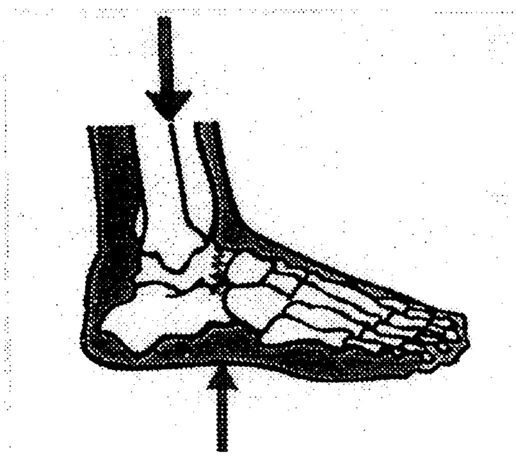


Figure 17. Collum Tali Fracture "Pilon Fracture"; the Ankle Muscles must be Tensed! (ref. Peterson 1974)

Tibia injuries

Injuries to the tibia usually occur in crashes with large intrusion where both the knee and foot are jammed, complemented with a perpendicular impact inducing bending of the bones.

This type of mechanism was also seen in the in-depth ankle study showing fractures induced by axial loads of the tibia, as well as perpendicular impact to the tibia while the leg was under axial loading. The mechanism could lead to ankle fractures, as well as fractures in the tibia and fibula. The fractures induced by axial loading occurred even with minor intrusion, indicating that the acceleration part of the severity should be studied separately from intrusion.

INJURY FACTORS

Having evaluated the statistical material as well as the different types of leg injuries occurring in the field, the main factors of leg injuries, may now be identified. The four main factors and the estimated influence of these factors as a function of speed, is schematically represented in Figure 18

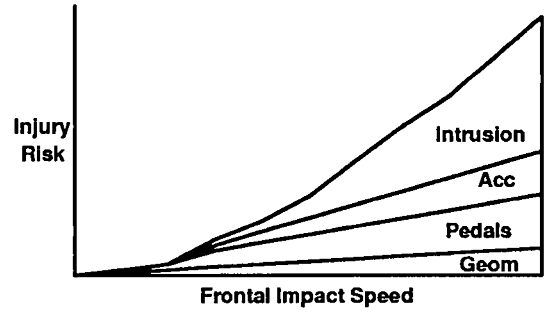


Figure 18. Distribution of Leg Injury Factors in Frontal Impact, with Respect to Impact Speed.

Although difficult to quantify, the existence of these factors may be identified from the accident data and injury types studies as follows:

Geometry

At lower speeds, the main factor is the geometry of the footwell area. There are a number of cases where foot and ankle injuries have occurred at low impact speeds (<15 mph) and where footwell intrusion is not expected. As illustrated in Figure 19, this represents about 10% of the AIS2+ injuries.

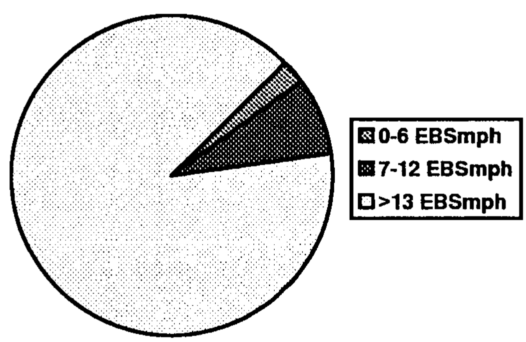


Figure 19. Leg Injury Distribution with Respect to Equivalent Barrier Speed EBS in mph

These cases are all either injuries to the foot or to the ankle. This, coupled with the fact that the foot and ankle tolerances are significantly reduced, if angled, indicates that the footwell geometry is the injury factor in these cases.

Considering the fact that the pedals may also be an injury factor, we can specifically look at the passenger side, since there is no chance of pedals being involved here. As illustrated in Figure 6, the same type of injuries are present here. This verifies the geometry as a main injury factor.

Pedals

Pedals may contribute to leg injuries in mainly two ways: Pedal geometry and Pedal intrusion. A general indication of the influence of pedals, can be seen by comparing the passenger and driver's side with corresponding impact types; left and right offset. As illustrated in Figures 1 and 3, the driver is at a higher risk of sustaining foot injuries in general, when compared to the front seat passenger. This is an indication that pedals are a source of leg injury.

A stronger indication, however, may be seen when comparing two asymmetrical impact types; the overlap situation where the intrusion is on the driver's side and where it is on the passenger's side. This comparison can be made due to the symmetrical structure and packaging of the Volvo 200 and 700/900 series.

The difference in injuries, both in the full frontal case (Figure 9) and in the corresponding offset cases (Figure 10,11), indicate that pedals are a source of injury.

Pedal geometry - At low speeds, the pedals are believed to cause injuries by twisting the ankle, the foot slipping off the pedals or being subjected to a local load combined with tense muscles from braking. The in-depth ankle study gave examples of such cases, where there was little or no intrusion and injuries still occurred (see INJURY TYPES).

Pedal intrusion - When the intrusion of the firewall starts, the brake pedal booster may also be affected. The booster is contacted by the engine, or other objects in front of the firewall, causing the booster rod to be pushed inwards into the footwell. This, in turn, means that the pedal itself will be pushed into the footwell and due to geometry, pedal intrusion will be greater than the booster intrusion.

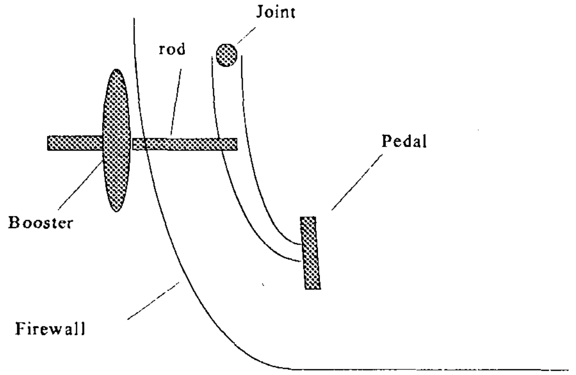


Figure 20. Typical Brake Pedal Configuration.

Acceleration

At higher speeds, the front of the car will be crushed far enough for the engine, or other objects in front of the firewall, to hit the firewall. When this occurs, intrusion in the footwell area will start. When the foot hits the firewall, the difference in speed between the foot and the firewall (delta V), is reduced to zero. Depending on the magnitude of this difference and how quick the reduction is (acceleration), a force will act on the foot.

This quick reduction of delta V (acceleration), may be shown to induce high force levels (see Simulation and Crash Test Studies) and this, in turn, increases the risk of injury.

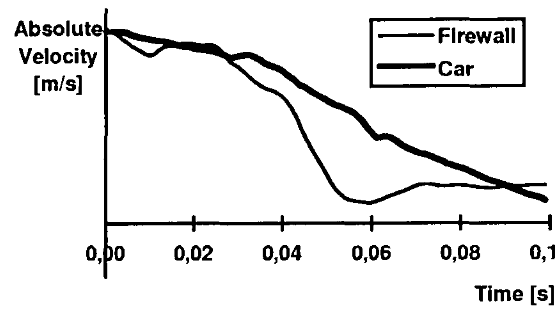


Figure 21. Absolute Car- and Firewall Velocity during a Crash Test.

Measurements from crash tests with dummies show that the contact force induced by a large delta V and acceleration, may be high (Figure 22).

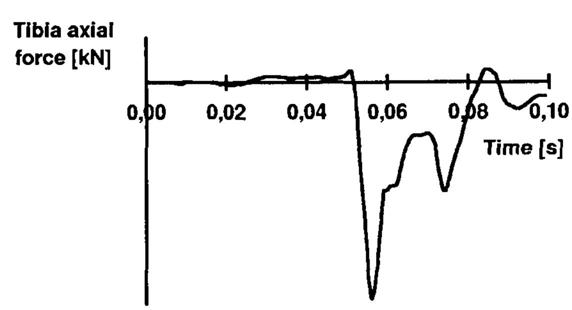


Figure 22. Typical Tibia Axial Force due to Foot Impacting the Firewall in a Crash Test.

Statistically, these acceleration related injuries are found in an area where intrusion is relatively small (<150 mm) and with reference to accident data, these represent approximately 22% of the injuries. The majority of these cases are foot and ankle injuries and the distribution is summarised in Figures 5 and 6.

Intrusion

The leg injury cases, which involve intrusion in excess of 150 mm, constitute 78% of the leg injuries. Most of these cases occur at high speeds and statistics show, that these injuries do not only involve foot and ankle injuries, but also fractures of tibia and/or fibula.

There are many possible reasons for injuries in these severe situations. The in-depth ankle study showed that jamming of feet and legs are common factors. It is also reasonable to assume that many of these injuries may be initiated at the start of intrusion due to geometry, pedals, high acceleration, etc.

If risk of injury is plotted against deformation, the curve rises along with the deformation / intrusion (see Figure 3). It could be said that this is an indication that intrusion is a main factor of injury. However, it is important to remember that as deformation increases, so does the number of factors contributing to an injury i.e. Geometry, Pedals and Acceleration. Therefore, it is difficult to determine the number of injuries which are directly related to intrusion alone.

It is important, however, that intrusion is reduced enough to avoid jamming effects, trapping of feet etc.

SIMULATION AND CRASH TEST STUDIES

Based mainly on the material from the in-depth ankle study and crash tests, there were indications that the quick velocity change (acceleration), occurring when the foot impacts the firewall, may be an injury factor. In order to verify this and also to find out how a suitable padding solution could be configured, a series of mathematical simulations and physical tests were conducted.

Since the acceleration at impact is a direct consequence of the delta V between foot and firewall, a second group of simulation runs were made, in order to find the influence of this parameter.

Simulation model

A sled test model, consisting of rigid bodies in the crash victim program MADYMO2D, was used to evaluate suitable padding characteristics and thickness. The input data such as interior geometry, crash pulse etc. represent a large Volvo car. The model was validated for a 50th percentile belted dummy, with airbag in driver position, in a 35 mph full frontal sled test. Pedals were excluded in the test and simulation.

Parameter Study with Padding and Toepan Angle

All simulations were conducted under the same conditions as the validated model. Two different toepan angles relative to the floor plane, were simulated:

- 38°, undeformed geometry.
- 63°, intruded geometry.

Five different floor characteristics were included in the study as follows:

- A reference characteristics including floor, carpet, shoe and foot deformation which represent the floor without extra padding (K0).
- Four theoretical padding characteristics, which deforms at a constant force level combined with the reference characteristics (K0), shown in Figure 23 (K1-K4).

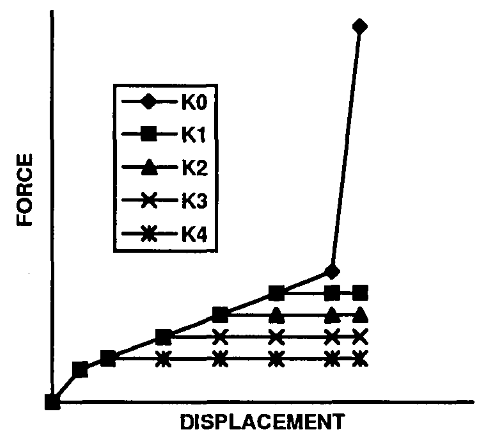


Figure 23. Floor and Padding Characteristics.

Results from Padding Study

The Tibia axial load was reduced by 35% using 15 mm of effective padding (constant force/deformation characteristics) and undeformed foot area geometry. The required physical padding thickness was estimated as the effective thickness plus 15 mm, which includes force build-up and remaining thickness at full compression.

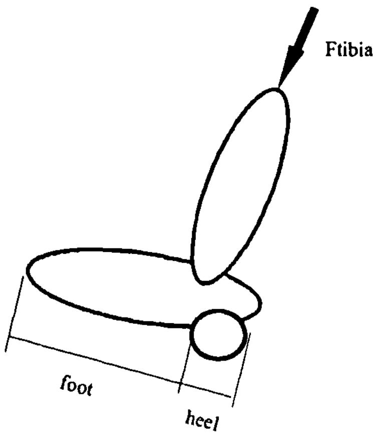


Figure 24. Madymo2d-foot. Definition of Contact Surfaces Foot and Heel in the Model.

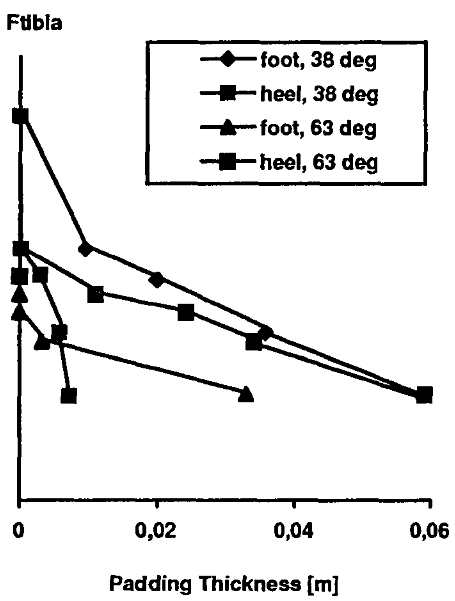


Figure 25. Tibia Load as a Function of used Padding Thickness.

Results showed that the padding was efficient even at a very low thickness. By increasing the thickness, the tibia load was reduced even further although not as much as it was by the first 10 millimetres.

Results also showed that the foot joint angle was mainly dependent on the foot area geometry. The effect of padding stiffness had a minor influence due to local deformation in the area where the foot loads the toepan

Sled Test Results

In order to verify the mathematical model, a series of sled tests were conducted, using the same padding characteristics as in the mathematical model. The tests were performed with an unbelted passenger dummy and airbag, at 30 mph. One reference test with only a steel floor and a carpet, was initially conducted. Padding materials, with a 30 mm thickness were positioned in the footwell in the two remaining sled tests.

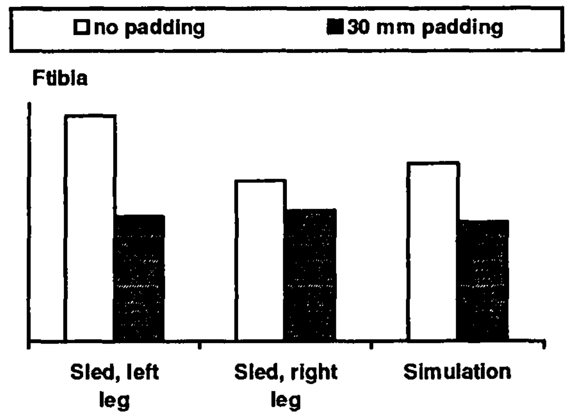


Figure 26. Tibia Load in Sled Test, With and Without 30 mm Thick Padding.

The Tibia axial load was, on average, reduced by 45% in the left leg and 20% in the right leg for the best padding selection. The improvement seen in the simulations was also seen in the tests.

Mathematical Study of Foot position and Relative Foot velocity

A second study was performed to relate the impact speed of the foot, when it contacts the toepan, to Tibia axial force. The same conditions were chosen as above, but padding was not included in these three simulations. The 63° toepan was used with three different positions translated in the x-direction:

- Nominal position.
- +50 mm in the x-direction. Moved rearwards towards the dummy so that the foot rests on the toepan.
- -50 mm in the x-direction. A distance of 100 mm between foot and toepan is obtained at the start of the simulation.

The speed of the car and foot as a function of time, is shown in Figure 27. As the speed of the car is decreased, the foot is moved forward, towards the footwell. When impact between foot and footwell occurs, the difference in speed (delta V in figure 27) is reduced to zero (the point where the lines cross), and some rebound occurs (foot speed lower than car speed).

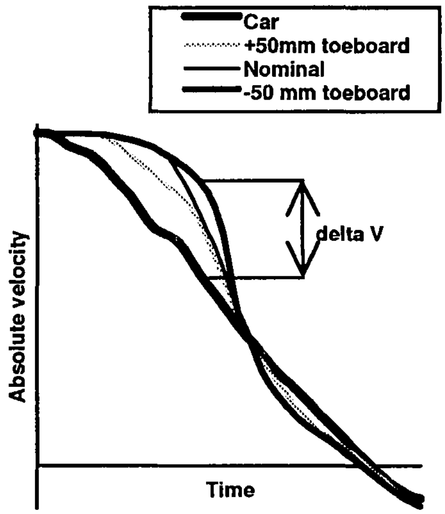


Figure 27. The Car and Foot Absolute Velocity for the Three Different Toepan Positions.

The slope of the absolute foot velocity represents the foot deceleration. The resultant foot deceleration and axial Tibia forces are illustrated in picture 28 and 29.

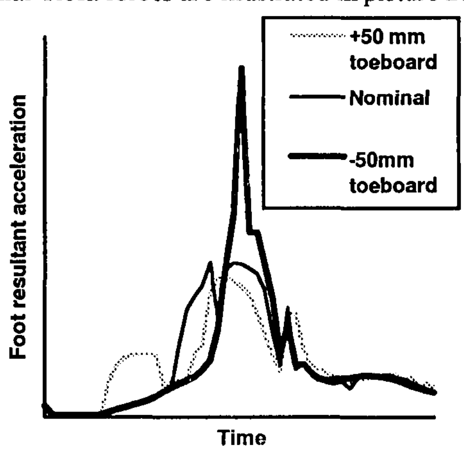


Figure 28. Foot Deceleration for the Three Different Toepan Positions.

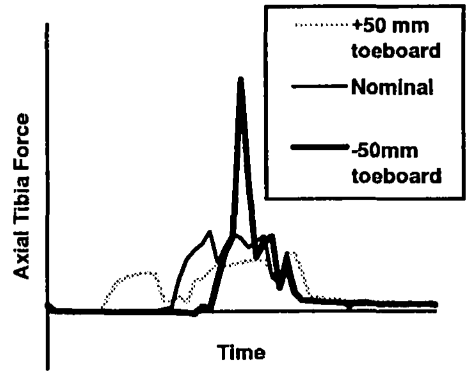


Figure 29. Tibia Axial Forces for the Three Different Toe Board Positions.

The similar curve shapes display the connection between resultant foot deceleration and Tibia axial forces. An increased distance between foot and toepan, leads to a higher impact speed. The following parameters affect the tibia force:

- Foot relative impact speed at the toepan.
- Stop-distance at the toepan.

The results show that the initial position of the foot is an important parameter since it affects the amount of load in the leg (foot, ankle and tibia). If delta V is high enough, the floor, carpet, shoe and foot will 'bottom out' and as a result, the load in the leg will rapidly increase.

CONCLUSIONS

The mechanisms behind leg injuries are complex and difficult to understand. In this paper an attempt has been made to point out the main factors by using statistical material, case study, simulation, testing and expert judgement. The main factors have been shown to be:

- Geometry - based on a number of cases occurring at low severity (< 15 mph EBS) and no footwell intrusion. Reference also to case 11b in the in-depth ankle study.
- Acceleration - based on a number of cases occurring with moderate intrusion (< 15 cm) and no pedal involvement. References also to cases in the in-depth ankle study, simulation and crash test studies, showing that impact leads to high forces in foot and tibia. Simulation and crash tests also show that delta V and padding are sensitive parameters for the resulting load.
- Pedals - based on higher statistical injury risk for the driver, coupled with the driver / passenger comparisons conducted in the injury factors' section

of the paper. Clear indications were also seen in the in-depth ankle study.

- Intrusion - based on the connection between increased intrusion and increased risk of injury. Tibia injuries due to jamming effects were also seen in the in-depth ankle study.

These factors are distributed with respect to impact speed in accordance with Figure 18. As the speed increases, a greater number of these factors contribute to the likelihood of an injury. With the knowledge that leg injuries are not the result of a single injury factor, it is easier to understand that in order to achieve any significant improvements in this field, improvements have to be made in all areas. In other words, it is not enough, just to reduce intrusion or acceleration; leg fractures will still occur due to pedals and / or geometry.

A logical continuation of this study, will be to quantify the influence of the four injury factors, as a function of impact severity.

Preventing injuries

In order to reduce the risk of a leg injury, it is important that all factors are taken into consideration. Regarding each factor, the following general design guidelines are suggested:

Geometry
- Attempt to make the footwell as smooth and flat as possible. Avoid having local differences in height and width.
- Place pedals as close to the footwell as possible. Ultimately, remove the pedals.

Acceleration
- Avoid placing solid objects in front of the footwell, which may cause increased stiffness of firewall when intruded.
- Design the footwell so that it will be shock absorbing, in order to reduce foot acceleration at impact.
- Allow for the feet to be placed close to the firewall in order to limit the delta V at impact.

Pedals
- Place the pedals as close to the firewall as possible.
- Design the pedals so that the brake booster intrusion will have limited effect on pedal intrusion.

Intrusion
- Limit intrusion
- When intrusion occurs, the footwell area should stay flat in order to avoid trapping the feet.
- Instrument panel and knee bars should be designed in such a way that the possibility of jamming the leg during impact, is reduced.

ACKNOWLEDGEMENTS

The authors wishes to thank the following persons for their invaluable contributions to this paper; Alan Dyche, Jan Ivarsson, Mats Moberg and Richard Nilsson. Acknowledgements also to the following persons for their involvement in the in-depth ankle study; Olle Bunketorp, Christer Gustafsson. Peder Johnsén and Pawel Seremak.

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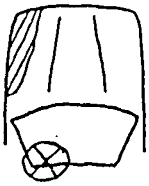
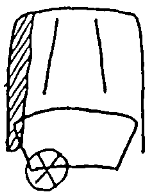
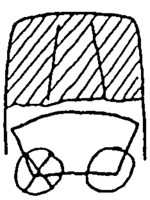
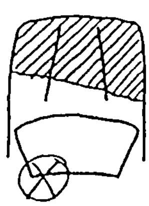
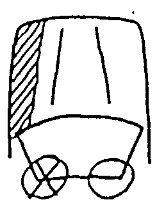
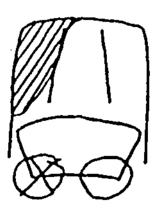
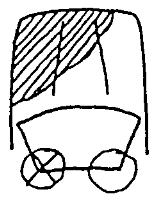
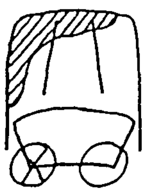
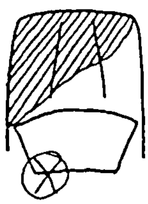
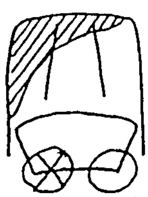
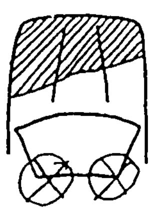
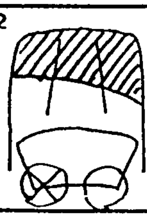
Case no	*Vehicle data*	*Injured occ.*	*left ankle*	*right ankle*	*comments*
1 	**Model:** Volvo 244 **CDC:** 11FLAN35 **Intrusion of** **wheel:** yes **side member:** no **foot well:** minor **Pedals:** deformed	**Position:** driver **Age:** 38 years old **Sex:** male **Braking:** no	**Injuries:** • ligament torn off lateral side of talus • small fracture talus lateral side **Mechanisms:** • inversion	**Injuries:** • fractures medial and lateral malleolus • (fibula fracture 5cm above ankle) **Mechanisms:** • inversion combined with axial load	• occupant jammed
2 	**Model:** Volvo 744 **CDC:** 11FLAW70 **Intrusion of** **wheel:** yes **side member:** no **foot well:** yes **Pedals:** probably deformed	**Position:** driver **Age:** 44 years old **Sex:** male **Braking:** probably not	**Injuries:** • comminuted fracture medial malleolus **Mechanisms:** • eversion combined with axial load • probably also direct impact to malleolus	**Injuries:** • no	• Occupant jammed
3 	**Model:** Volvo 745 **CDC:** 12FDEW45 **Intrusion of** **wheel:** yes **side member:**yes **foot well:** minor **Pedals:** deformed	**Position:** driver **Age:** 48 years old **Sex:** male **Braking:** probably	**Injuries:** • axial fracture of talus **Mechanisms:** • high and distributed axial load	**Injuries:** • fracture medial malleolus **Mechanisms:** • eversion	• passenger no leg injuries though larger wheel intrusion
4 	**Model:** Volvo 745 **CDC:** 12FDEW40 **Intrusion of** **wheel:** no **side member:** no **foot well:** minor **Pedals:** deformed	**Position:** driver **Age:** 45 years old **Sex:** female **Braking:** yes	**Injuries:** • fractures medial and lateral malleolus **Mechanisms:** • inversion combined with axial load	**Injuries:** • open fracture lateral malleolus **Mechanisms:** • eversion combined with axial load	• fracture on right fibula from direct impact, probably by the deformed accelerator pedal • prob. pressed clutch pedal with left foot at impact
5 	**Model:** Volvo 744 **CDC:** 12FLEW50 **Intrusion of** **wheel:** major **side member:** no **foot well:** yes **Pedals:** probably deformed	**Position:** driver **Age:** 40 years old **Sex:** male **Braking:** probably not	**Injuries:** • comminuted fracture medial malleolus **Mechanisms:** • eversion combined with axial load and transverse impact	**Injuries:** • no	• occupant jammed
6 	**Model:** Volvo 764 **CDC:** 12FLEW50 **Intrusion of** **wheel:** yes **side member:**yes **foot well:** yes **Pedals:** deformed	**Position:** driver **Age:** 71 years old **Sex:** male **Braking:** probably not	**Injuries:** • no	**Injuries:** • fracture medial malleolus, low energy **Mechanisms:** • shear forces mainly, probably oblique ankle at impact	• fracture left femur

Case no	*Vehicle data*	*Injured occ.*	*left ankle*	*right ankle*	*comments*
7 	**Model:** Volvo 245 **CDC:** 12FYEW45 **Intrusion of** **wheel:** major **side member:**yes **foot well:** minor **Pedals:** deformed	**Position:** driver **Age:** 45 years old **Sex:** male **Braking:** probably	**Injuries:** • no	**Injuries:** • horisontal fracture in talus anterior part **Mechanisms:** • load axially-medially	• no injuries to right leg despite major wheel intrusion
8 	**Model:** Volvo 745 **CDC:** 12FYEW35 **Intrusion of** **wheel:** yes **side member:** no **foot well:** major **Pedals:** deformed	**Position:** driver **Age:** 41 years old **Sex:** male **Braking:** yes	**Injuries:** • collum tali fracture "pilon fracture" **Mechanisms:** • axial impact, tensed muscles	**Injuries:** • collum tali fracture "pilon fracture" • fracture medial malleolus **Mechanisms:** • extensive axial load combined with inversion. Force at the anterior part of calcaneus	
9 	**Model:** Volvo 245 **CDC:** 12FDAW **Intrusion of** **wheel:** yes **side member:**yes **foot well:** minor **Pedals:** deformed	**Position:** driver **Age:** 20 years old **Sex:** female **Braking:** no	**Injuries:** • fracture medial malleolus **Mechanisms:** • eversion	**Injuries:** • fracture lateral malleolus **Mechanisms:** • inversion	
10 	**Model:** Volvo 744 **CDC:** 12FLEW50 **Intrusion of** **wheel:** yes **side member:**yes **foot well:** yes **Pedals:** unknown	**Position:** driver **Age:** 46 years old **Sex:** male **Braking:** no	**Injuries:** • subluxation of choupards joint • (fracture of 5th metatarsal) **Mechanisms:** • axial compression, sliding and jamming of foot	**Injuries:** • collum tali fracture "pilon fracture" • (fractures in tibia and fibula 10 cm above ankle) **Mechanisms:** • axial impact, tensed muscles • (lateral load and bending of tibia/fibula)	• fracture right femur
11a 	**Model:** Volvo 244 **CDC:** 2FDEW 35 **Intrusion of** **wheel:** no **side member:** probable **foot well:** no **Pedals:** unknown	**Position:** driver **Age:** 79 years old **Sex:** male **Braking:** probably	**Injuries:** • collum tali fracture "pilon fracture" • fractures at joint tibia/talus **Mechanisms:** • axial load when oblique ankle OR • load axially-medially	**Injuries:** • collum tali fracture "pilon fracture" **Mechanisms:** • axial impact, tensed muscles	• fracture left hipjoint • both ankles probably effected by pedals
11b	**Model:** as 11a **Intrusion of** **wheel:** no **side member:** no **foot well:** no	**Position:** front passenger **Age:** 78 years old **Sex:** female	**Injuries:** • no	**Injuries:** • fracture lateral malleolus **Mechanisms:** • inversion combined with axial load	
12 	**Model:** Volvo 944 **Intrusion of** **wheel:** no **side member:** no **foot well:** no **Pedals:** deformed	**Position:** driver **Age:** 48 years old **Sex:** female **Braking:** no	**Injuries:** • no	**Injuries:** • fracture lateral malleolus **Mechanisms:** • inversion	• right foot on acc. pedal • the load in the ankle is induced by body load only

THE REDUCTION OF THE RISK OF LOWER LEG INJURIES BY MEANS OF COUNTERMEASURES OPTIMIZED IN FRONTAL OFFSET CRASH TESTS

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Paper No. 96-S3-O-06

ABSTRACT

In the seventies injuries of the lower extremities were frequently observed in serious frontal collisions. Although these injuries were not perilous, there were dramatic personal and financial consequences.

In the late seventies Mercedes-Benz introduced an offset crash test procedure for the development of new car lines. In view of the poor biofidelity of the dummies' lower legs and the lack of knowledge about human tolerances in this body region at that time, as a first step critical load values were determined in volunteer tests. These values have been checked in crash tests with modified dummies and are still valid today, as a comparison with the latest biomechanical research in this field shows.

Various safety measures were then incorporated in all new Mercedes-Benz car lines in the eighties and have considerably reduced the risk of foot injuries particularly in frontal offset collisions.

This reduction can now be proven by a comparison of the results from the analysis of real world accidents over the last two decades. In addition, the findings show new priorities for future safety measures and the need for an improvement of the test procedure resulting in offset crash tests against a deformable barrier.

1. The History of the Offset Test

In 1973 Mercedes-Benz conducted the first offset crash test derived on the basis of accident research which started in 1969. This test was carried out at 65 km/h and an overlap degree of 50%.

The 50 km/h head-on test at 100% overlap with its high deceleration became the standard test for the effectiveness of restraint systems. As a consequence the

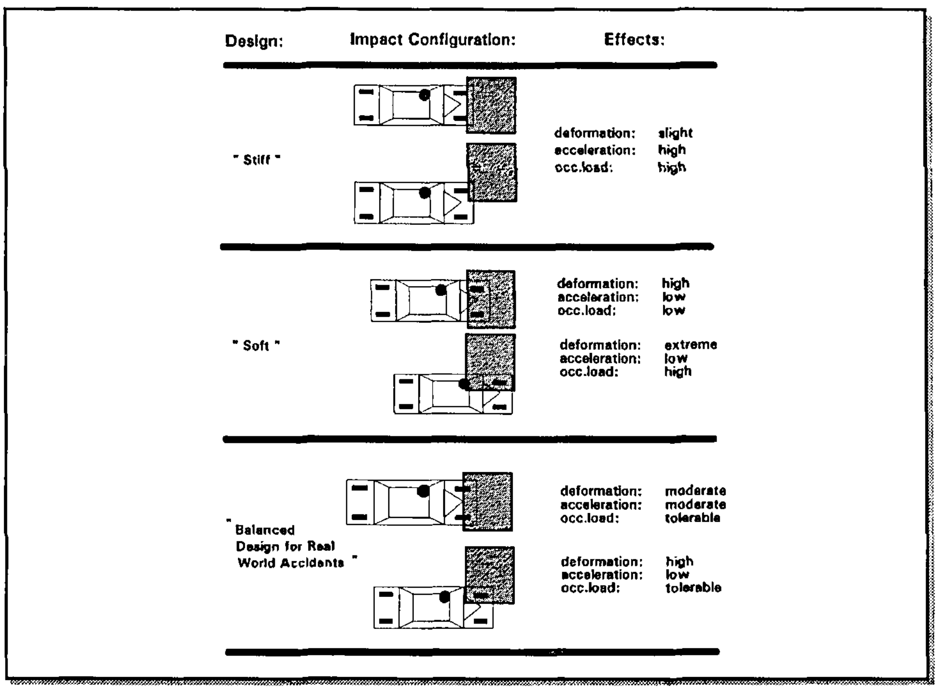

Figure 1: Comparison of a „soft“ and a „stiff“ car front structure in a full overlap and an offset frontal collision.

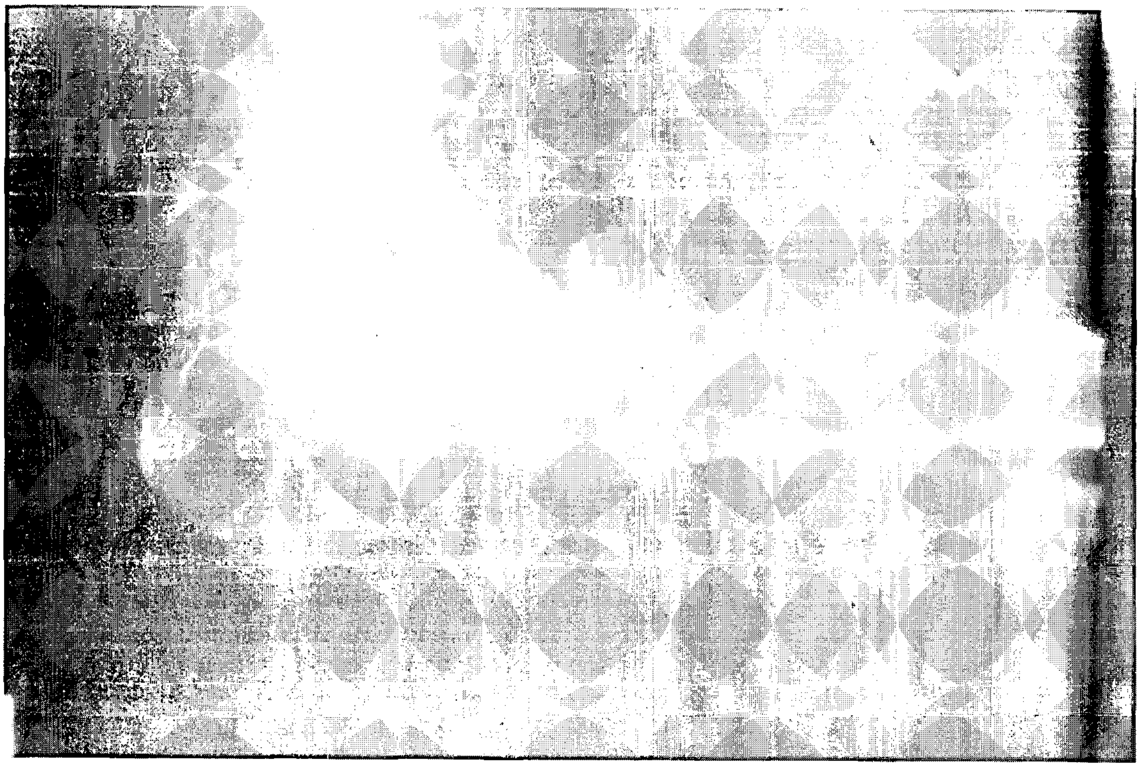

Figure 2: X-ray picture of a fracture of the heel

restraint systems were improved more and more. So the outcome of accidents gave growing significance to lower leg injuries, especially due to intrusion in offset impacts with lower overlap degrees.

This problem was discussed on the 25th Stapp Car Crash Conference, 1981 **[1]**, in combination with the introduction of a new accident reconstruction method based on the Energy Equivalent Speed (EES), which shows a significant difference to the change of velocity (Δv) particularly in offset impacts with glance-off.

Consequently, in additon to the impact against the flat barrier as a restraint system test, another test method was defined as a structural test to prove the rigidity of the passenger compartment. This test was conducted with 55 km/h against a stiff offset barrier with 40% overlap in order to test the structure and the passenger compartment as realistically as possible.

During the development of the new car lines based on these two tests it could be recognized, that the tests had contradictory demands. On that bas is the Mercedes-Benz Safety Philosophy was derived as a balanced design and with the certainty that a good offset design at that time - it was at the end of the seventies - could only be guaranteed, if the structure was not as weak as it should have been to reach best NCAP[1)] results. This can

1) New Car Assessment Program of the US National Highway Traffic Safety Administration

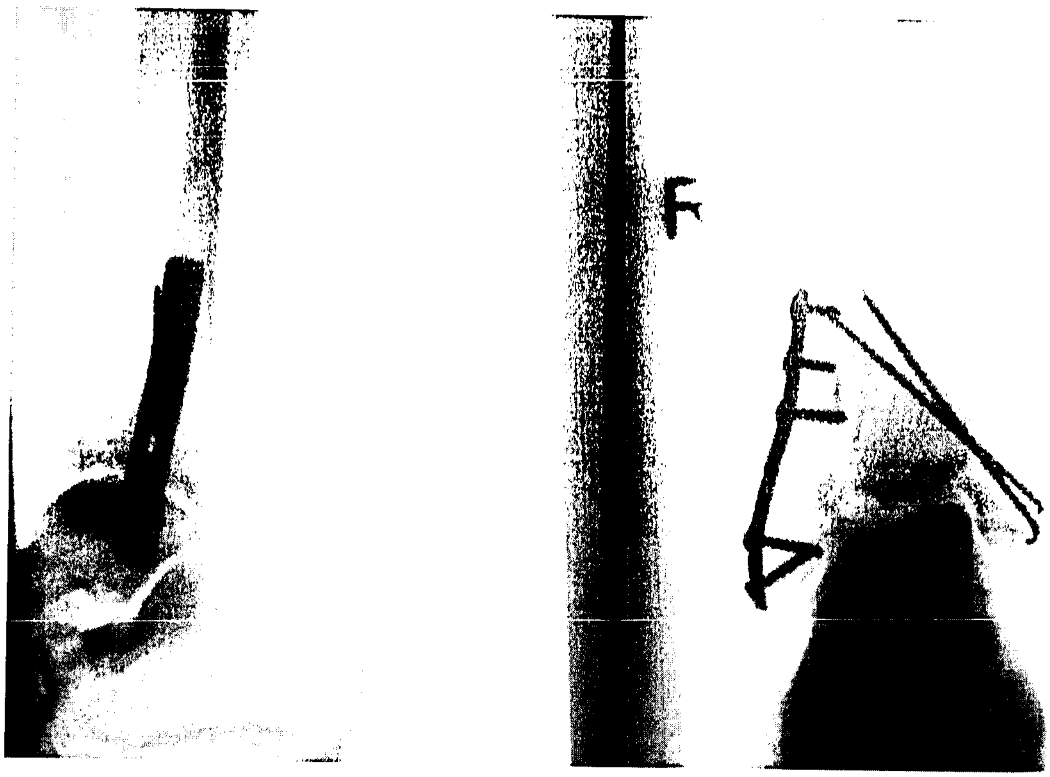

Figure 3: X-ray picture ofan ankle fracture (bimalleolar fracture after surgery) [2].

be best explained by **Fig. 1**, showing the principle difference between a soft and a stiff car with regard to the injury outcome.

So the offset test was originally introduced as a structural test to protect the lower extremities, which caused the highest amount of injury costs - if severely injured.

But at that time the protection of more life-threatening injuries to other body regions like head and chest had priority and nobody cared about injury criteria for the lower legs, and dummies were not designed to measure anything in this body region.

2. Ankle Injury Patterns

2.1. Impact Shock Syndrome

2.1.1. Injury Causation

The term "Impact Shock Syndrome" characterises an injury caused by a small intrusion of the footwell (the greater part in the elastic range of the deformation) with a very high intrusion velocity. The high velocity leads to a compression and crush fracture of the heel bone (calcaneus) and possibly of the ankle bone (talus) **(Fig. 2)**.

2.1.2. Injury Consequences

A first step investigating this kind of injury had already been achieved in 1983 [2].

As a consequence of the results of this work a research project was carried out together with four German institutes[2)] bringing up an Injury Cost Scale **(ICS)** [3], which showed that an injury of this kind results in costs to the national economy in a similar magnitude as an open craniocerebral trauma (without considering lethal craniocerebral trauma-cases - as the compared lower leg fractures rarely are lethal) **(Table 1)**. These costs are not only figures describing the burden for the economy, but they also represent the personal suffering of the afflicted persons, because consequent impairments are also included. Long term disablements are very frequent (due to displacements of the articular surfaces). This is shown in the fact that there were not enough cases with a 'fracture of the calcaneus without consequences' to give a figure for the costs of this injury.

2) Bundesanstalt für Straßenwesen (Federal Highway Research Institute in Germany), Forschungsvereinigung Automobiltechnik (FAT, research association of German car manufacturers), Institut für Rechtsmedizin (Institute for forensic medicine) in Mainz, Hauptverband der gewerblichen Berufsgenossenschaften (German Workman's Compensation).

2.2. Anklebone fracture

2.2.1. Injury Causation

Fractures of the anklebones (the distal ends of tibia and fibula) are caused by a combined compression, bending, torsion, and tension, resulting from forces applied at the foot in the case of foot well or pedal intrusion or even from the mere vehicle deceleration force of the foot against the foot well or pedal (**Fig. 3**).

2.2.2. Injury Consequences

If the articular surfaces are reset and heal in the anatomical position, there is a good prognosis for a symptom-free healing. As a consequence the injury costs of ankle fractures are by far lower than those of heel bone fractures (**Table 1**).

3. Other Skeletal Injuries of the Foot

Fractures of the metatarsal bones also occur in frontal offset collisions (**Fig. 4**), but the costs they cause are far below the heel fracture costs (**Table 1**).

A survey about foot injury patterns and their frequency is published in **[4]**.

So, the questions: "What is the objective of a simulation of a frontal collision regarding lower leg and foot injuries? Is it more important to simulate the conditons of the impact shock syndrome or those of other fractures of the foot?" are also ethical questions: which injury pattern is more weighty for the injured people and for the national economy? This problem was again addressed in **[5]**.

4. Injury Criteria and Dummy Technology

Not having any injury criteria for our car development, we started to derive a simple injury criterion. Suspecting an acceleration-induced injury mechanism for the calcaneus fractures in offset frontal collisions, because severe injuries observed in offset impacts were similar to injuries sustained in falls or even jumps out of a height of above 2 meters, this attempt based on a volunteer jump test in comparison to dummy drop tests. Accelerometers were installed in the jumping volunteer's heel of the shoe and in the Hybrid III-dummy's modified foot. Foot accelerations were measured for falling heights up to 2 meters. The result was an "injury criterion" of an acceleration of 150 g in the dummy's foot. Ten years later, we have new additional results based on impactor tests with PMTS **[10]**, and the results concerning the acceleration of the foot were not inconsistent with the older findings: Above 220 g severe fractures of the heel were found. The results of these impactor tests demonstrated furthermore that with the state of the art Hybrid III dummy a lower

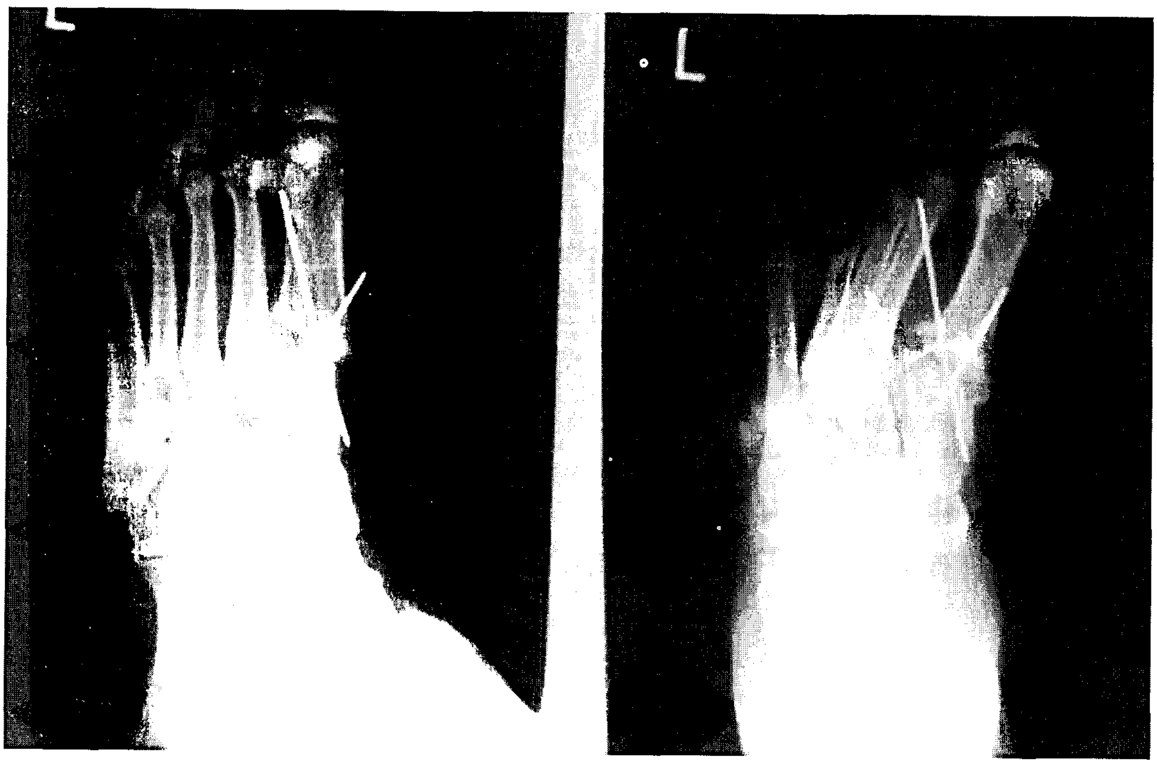

Figure 4: X-ray picture of fractures of all five metatarsal bones and the cuboid bone **[2]**.

	AIS-Code (1990)	ICS[3)] [DM]	ICSL[4)] [DM]
Fracture of the calcaneus, as a single diagnosis without consequences	851400.2	too few cases	too few cases
Fracture of the calcaneus, with functional impairment	851400.2	266.000	266.000
Closed fracture of the malleoli, as a single diagnosis without consequences	851608.2 853412.2 851612.2	47.000	47.000
Closed fracture of the malleoli, with functional impairment	851608.2 853412.2 851612.2	98.000	98.000
Closed fracture of the metatarsal bone, as a single diagnosis without consequences	852200.2	31.000	31.000
Closed fracture of the metatarsal bone, with functional impairment	852200.2	123.000	123.000
Open fracture of the vault of the cranium with loss of brain tissue	150406.4	105.000	685.000

Table 1: Comparison of injury severities and costs to the national economy of certain injuries, as given in the ICS (1 DM ≅ 0,66 US$).

leg criterion like the Tibia Index showed no better correlation with the injury outcome than the foot acceleration because of the lack of biofidelity of the instrumented leg.

The Hybrid III instrumented leg offers measuring points for M_x and M_y below the knee, for F_x, F_y, F_z, M_x and M_y above the foot joint. Today, lower dummy-legs with to some extent improved biofidelity are being developed. However, the number of the measuring instruments in the dummy grows faster than the knowledge about biomechanics of lower leg injuries.

5. Representative Test Procedure

When Mercedes-Benz in 1988 published the results of efforts with respect to an improved body structure protecting the lower extremities of car occupants, German car magazines started to conduct offset tests for safety rating. At the same time, the discussion about a European test procedure moved into the direction of an offset test [7].

Meanwhile we found that the offset test configuration should be further improved in order to simulate car-to-car collisions more realistically. This resulted in an offset test procedure with a deformable barrier. Our motive was to simulate deformation and intrusion for this kind of accident as realistically as possible, although this procedure was more severe for the car structure. But at the end, this test procedure showed the way for further improvements, because the car was tested in a more unfavourable situation [8].

The deceleration pulse of the offset test does not represent a severe test for the restraint systems, but this does not matter, because this test was a supplement to the 0° full overlap barrier test. Nevertheless, the long duration pulse beginning with a soft increase was a critical test for the airbag sensor simulating a soft crush characteristic occuring in eccentric impacts against poles or trees sufficiently well. But the main purpose was to detect weak points in the front end structure and derive suitable safety measures to enhance the protection of the lower extremities.

3) ICS = Injury Cost Scale without lethal cases

4) ICSL = ICS also considering lethal cases.

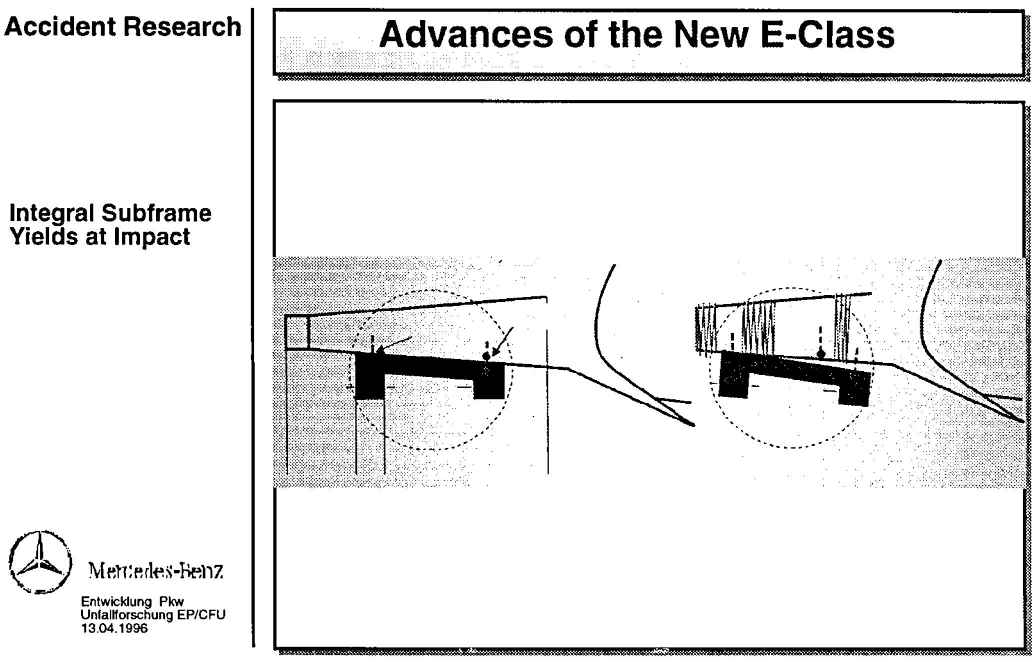

Figure 5: Disengaging integral member (filled) provides more free deformation range.

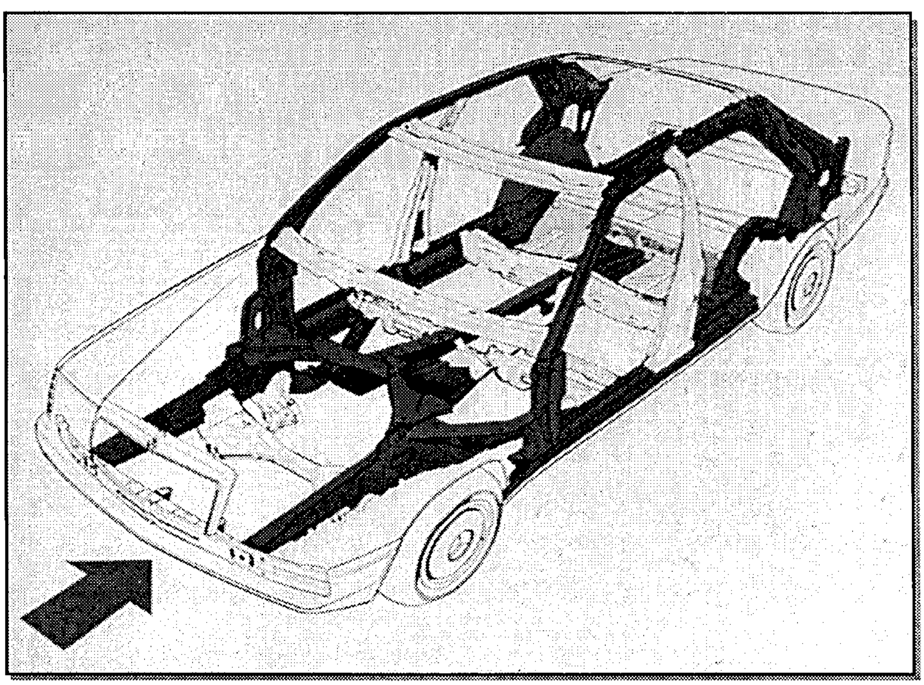

Figure 6: Forked front longitudinal member.

6. Actual Situation

6.1. Safety Measures for the Reduction of Injuries of the Lower Extremities

The reduction of the risk of lower extremity injuries was achieved by numerous safety measures meanwhile well-known through different publications [9, 10].

The most important safety measures refer to the front end structure, the structure of the passenger compartment, and the energy absorption characteristic in the foot well area of the driver. The reduction of the risk of head injuries was achieved by the introduction of the belt pretensioning retractor and by the reduction of the intrusion of the steering wheel into the passenger compartment by means of structural improvements.

6.1.1. Improvements of the Car Structure

The offset tests resulted in the development of a special structural vehicle concept, which forwards the longitudinal forces applied to the front end during offset collisions to the load-bearing components of the occupant cell (tunnel, floor, sidewall, **Fig. 6**). This vehicle concept was introduced to the market in 1980 in the Mercedes S-Class. It provides for a rigid foot well in a way that intrusions will be as harmless as possible.

The new E-Class (Car Line (**CL**)210) is conspicuous with respect to an enhanced impact energy absorption and long free deformation range of the front end, achieved by an integral member, which bears the engine and wheel suspensions of the front axle. This constitutes a stiff transverse assembly together with the front cross member optimally transferring impact forces from offset collisions to the entire front structure. At a certain load the integral member disengages from the body. This prevents an early bottoming out of the front end and advantage is taken of the full length of the front longitudinal member (**Fig. 5**). An additional transverse member at the firewall supports forces from the engine block. These improvements did not result in a heavier car body.

The effect of the optimization of the front end of the new E-Class with the deformable barrier was an improved compatibility towards lighter cars - like first internal car-to-car compatibility tests at Mercedes-Benz showed - and still preservation of safety for the lower extremities. In a renowned German car magazine the new E-Class was examined in an offset test and published as "the car with the best results ever achieved in this crash configuration" [**11**]. In our own data sample of W210 frontal collision cases no severe lower leg injury has been reported yet, but the number of

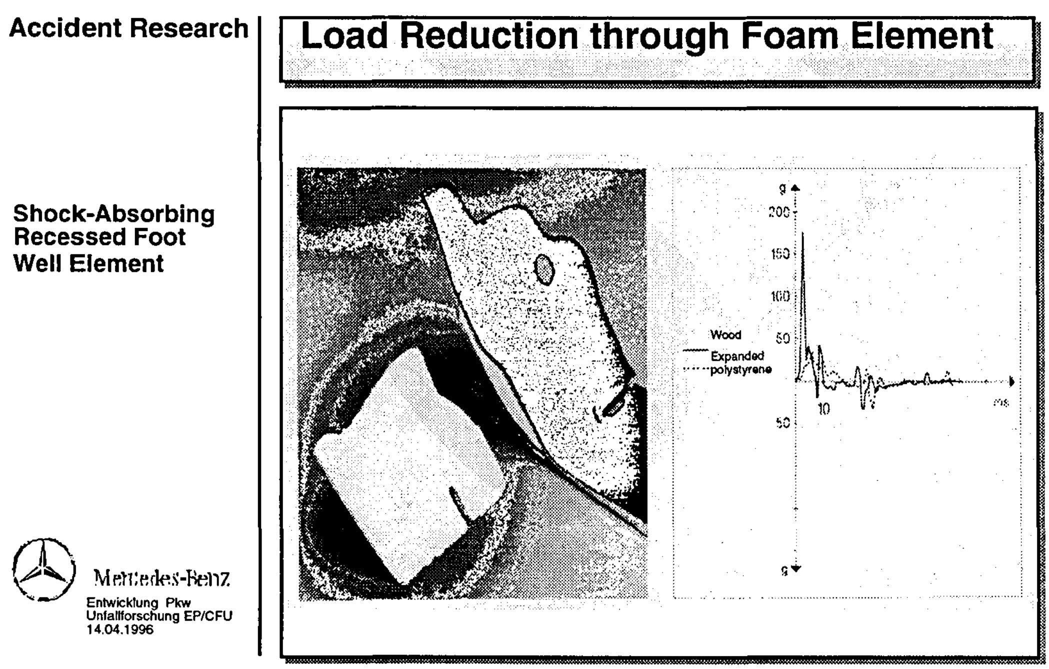

Figure 7: Foot well foam element.

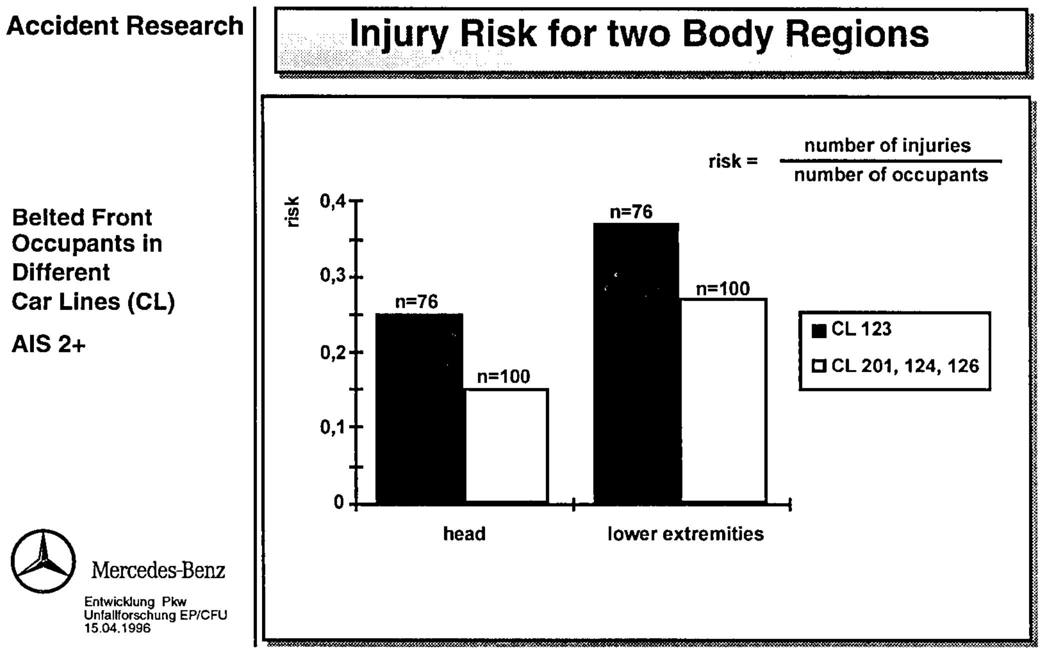

Figure 8: Injury risk for an earlier car line (CL 123) and later car lines (CL 201, 124, 126)

investigated accidents is still too small for an efficiency check.

6.1.2. Foot Well Foam Insert

Heel fractures, evoked by unavoidable small elastic intrusions of the foot well with a high intrusion speed, are prevented in a high degree by the energy absorbing foam insert in the foot well lining.

Comparing the car lines 124 and 126 (the predecessors of the present E- and S-Class) to their predecessors, the Mercedes-Benz accident research data show that the risk to sustain an AIS2-foot injury was reduced by about 20% through the improved front structure and the foot well foam insert. In the cars with the improved front structure and the foam insert we could only record one calcaneus-fracture. It was caused in a very severe frontal collision with an EES of 60 km/h.

CONCLUSION

The present situation is interesting with respect to two different aspects:

- First, we introduced safety measures on the basis of our internal demands going far beyond legal requirements at that time. The efficiency of these safety measures is shown in **Fig. 8** and **10**. The influence of the airbag - an option at that time - was eliminated in the figures for comparison reasons (the efficiency of the airbag is described in several other publications [**9, 12**]).

The overall effect of all safety measures including the belt usage and the airbag is shown in **Fig. 9**, which must be regarded with respect, because it shows the results of in depth investigations of many very severe accidents over the last two decades, which could only be collected with a huge effort.

So, even with simple injury criteria, efficient measures can be taken.

- Second, we find a confusing international situation with regard to the safety requirements:

  - There is a European test procedure, based only on the offset configuration, which therefore covers just half of the problem,
  - There are different safety rating systems based on different offset test configurations like TRL-NCAP, IIHS, AMS, ADAC. Unfortunately, some kind of competition with the unsaid slogan "The higher the test speed, the higher the level of protection" has arisen, disregarding the shortcomings with respect to compatibility. Under consideration of simple physical laws, it can be shown that a heavy weight large car has to absorb more deformation energy in terms of a higher EES-figure than a small car with a lower mass. A better solution would be a test speed in relation to the mass of the test vehicle in order to avoid incompatible and stiff front end structures for large cars.

• There is a new US-safety standard planned by NHTSA in a very complicated but realistic mode and in accordance to our safety philosophy as a supplement to the 0° barrier impact. Unfortunately, because it is a car-to-movable barrier test (which is a reasonable test configuration on principle), one can reduce the loadings to the car occupants by simply increasing the mass of the car. This side effect should be taken into account. From the point of compliance it can be expected that the repeatability of the results found under these test conditions will not meet the demands for the repeatability required for an international safety standard.

From the point of view of a car manufacturer who sells cars worldwide it would be much more sensible to harmonize the safety standards globally.

Any safety improvements which can help to save lives and prevent injuries in automotive accidents are still necessary and welcome. The offset test procedure of any kind is one important step in this direction.

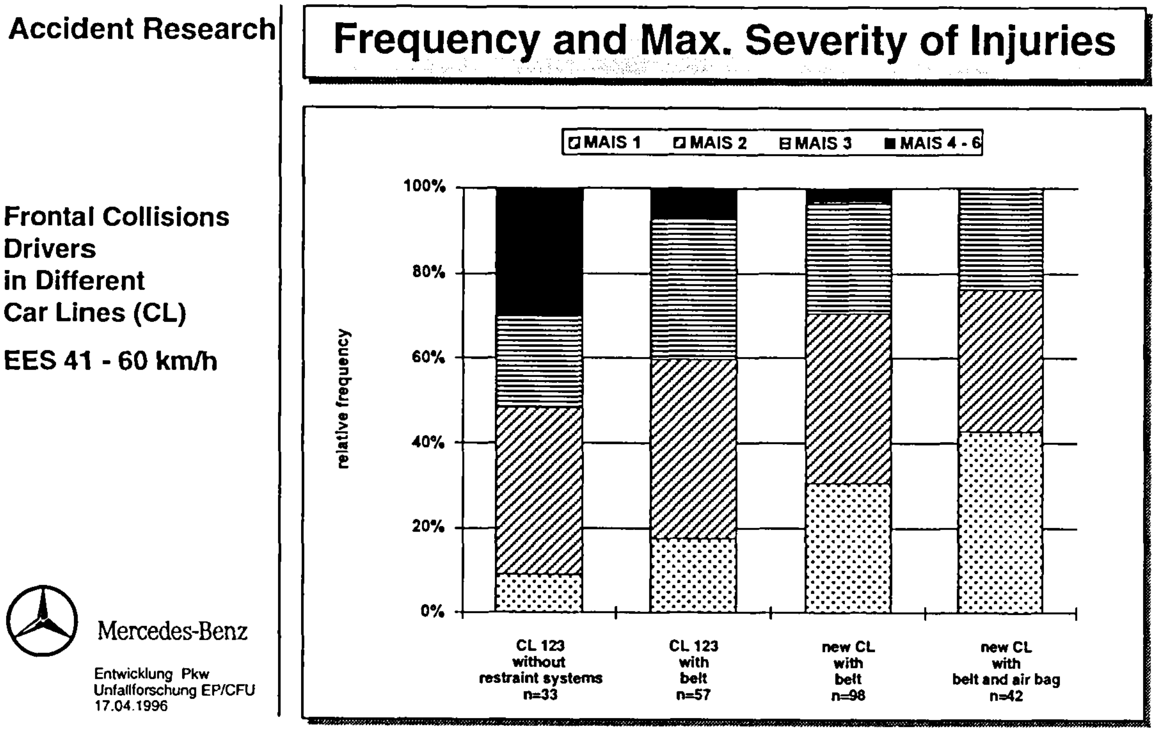

Figure 9: The overall effect of all additional safety measures since the car line 123 and before the new E-class.

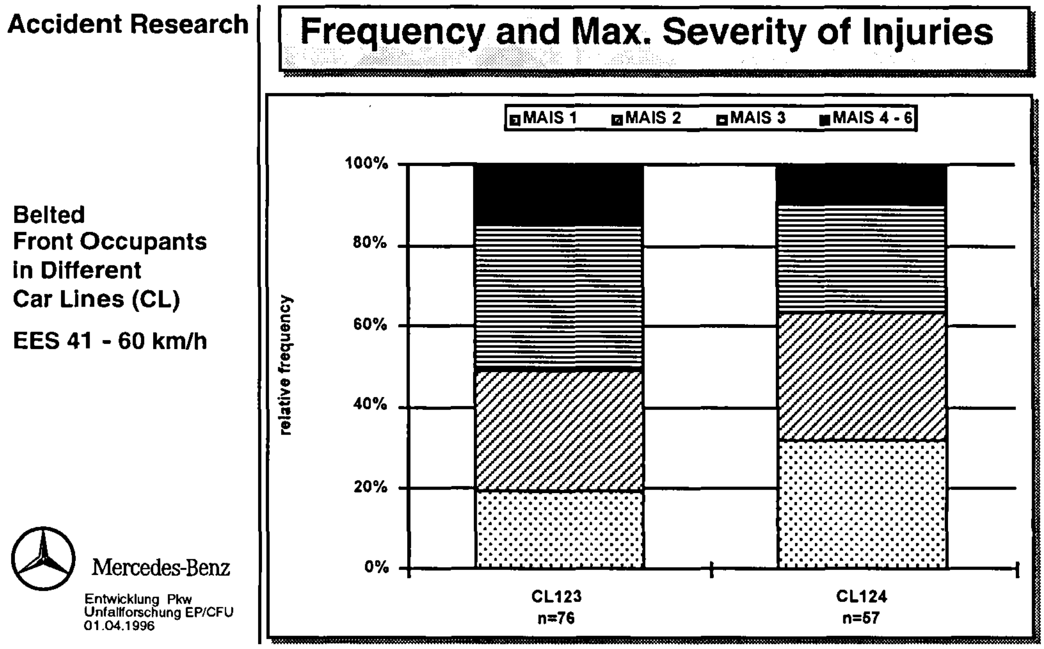

Figure 10: Comparison of the global injury severity in an earlier (CL 123) and a later car line (CL 124). (**MAIS** = Maximum Abbreviated Injury Scale value).

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OPPORTUNITIES TO IMPROVE FIRST GENERATION AIR BAGS

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Paper Number 96-S3-O-07

ABSTRACT

This paper will address current real world experience with air bags. Some adverse effects that have been encountered upon air bag deployment are the principal focus. Potential means for air bag performance improvements are discussed.

INTRODUCTION

It is reassuring to note that recent analysis of extensive claims data by State Farm Insurance has confirmed their conviction that the air bag is the single most valuable passive safety device available[1]. State Farm observes that in comparison to vehicles containing only belts, vehicles containing air bags have significantly reduced the seriousness of injuries in moderate and severe crashes. When considering medical, rehabilitation and lost wages experience per insured vehicle year, significant advantages are seen when air bags are present, according to the State Farm data. Similarly, Insurance Institute for Highway Safety has stated that their most recent estimates indicate that driver deaths are reduced 14 percent in all crashes and 20 percent in frontal crashes for air bag equipped cars[2]. Specific agency estimates of air bag performance are to be found in "NHTSA's Improved Frontal Protection Research Program" which is paper 96-S2-O-09 in this conference and in the agency's "Second Report to Congress on Effectiveness of Occupant Protection Systems and Their Use" dated February 1996.

As commented by State Farm, safety systems can have some circumstances in which their presence may worsen a situation, however, these circumstances must be infrequent and be offset by positive effects in the great majority of situations. State Farm goes on to observe that with air bag systems the positive effect on injury and fatality reduction is overwhelming, however, there are some, fortunately rare, instances in which they have seriously harmed certain individuals.

DISCUSSION

The majority of respondents to the request for comments by National Highway Traffic Safety Administration (NHTSA) in the Federal Register on November 9, 1995 generally supported the insurance industries' observations on the overall efficacy of air bags on our roadways. In this request for comments the agency acknowledges that it is aware of situations in which current air bag designs have undesired side effects which appear to have contributed to serious injuries and even death to vehicle occupants. Comments addressing these undesired air bag side effects and means for their reduction or elimination were solicited. The comments received may be found in NHTSA Docket # 74-14 Notice 97.

Air Bag Injury Mechanisms

It is generally accepted that air bag induced injuries to occupants are the result of one or more of three undesired air bag interactions with occupants. Before addressing potential improvements to current air bag systems, it would be best to clearly state and explain these adverse mechanisms which may produce threat to the car occupants for whose crash protection the air bag systems were designed.

a. Punch-out injuries may occur to an occupant who is in direct contact with, or in close proximity to, the stowed air bag protective cover at the time of air bag deployment initiation. Under these circumstances the compactly folded air bag and air bag cover constitute a significant coupled inertial mass which is driven towards the occupant by high pressure from within the air bag inflator acting on the folded air bag. Initially the high pressure and rearward motion of the folded air bag causes the pliable air bag cover to bulge rearward with high progression speed until the cover splits open along its weakened design tear seams as a result of the high tensile diaphragm stresses created by the cover bulging out of plane. The tearing of the air bag cover releases the compact mass of the folded air bag to accelerate towards the occupant under the influence of the high gas pressures generated up to bag burst-out. Injuries or fatalities may be produced if the occupant's head or chest is in contact or makes contact with either the bulging air bag cover or the folded air bag mass in the initial stage of escaping from beneath its burst cover.

If the head makes contact during this phase of air bag deployment excessive head acceleration may be produced or excessive neck hyper extension may result. In the case of chest impact during the punch-out phase of deployment, both excessive local loading of the chest and the very high rate of chest loading may result in flailed chest and/or blast type internal vital organ injuries. Punch-out injuries may be produced at any vehicle crash severity producing deployment.

It has been seen in some cases that drivers, predominantly women, have been exposed to a unique threat from contact with the decorative air bag cover as it swings open after tear seam release. In several crash investigations it has been observed that female drivers in low speed turns, experiencing air bag deployments at near design deployment crash induced velocity changes (about 11 miles per hour), suffer shattered forearms (comminuted ulna and/or radius fractures). In these cases it has been noted that the women have either had their right hand located at the approximate 9 - 10 o'clock orientation on the turned steering wheel rim in a left turn or the left hand located at the 2 -3 o'clock position in a right turn. The fractures have been located on the forearm overlapping the decorative cover at the time of air bag deployment in each case. These severe arm injuries are produced in the punch-out phase of air bag deployment.

At this point it may be well to hypothesize on means which may be employed to eliminate or ameliorate the punch-out threat. If the onset rate of gas generation currently employed in inflator designs were reduced sufficiently, we could expect that the cover rearward bulging speed would be reduced and the speed of the compactly folded air bag subsequently accelerating from confinement under its split cover could be diminished by the slower flow of gas from the inflator into the unfolding, but still compact air bag. Later in this paper this potential for reducing undesired side effects of air bag deployment by reduced gas onset flow will be further discussed in describing approaches of down loading of propellent in current regressive burn characterized sodium azide based inflators, staged inflation systems, and in use of progressive burn propellent developments in non azide inflators.

Another means for reducing the frequency of punch-out injuries is to prevent the occupant from being close to the deploying air bag. For belted occupants, the use of belt web grabbers or belt pretensioners might be expected to reduce the forward translation of the belted subjects during pre-crash braking and onset of crash. For the much more frequent subject of air bag induced injuries, the unbelted occupant, a crash sensing air bag initiation system which produced earlier air bag deployment initiation would reduce the likelihood of occupants being too close to the air bag at deployment. Occupant position sensors capable of tracking occupant forward motion during pre-crash braking could be used to disable the air bag in those situations where the occupant would have otherwise been too close to the air bag module at the initiation of its deployment. Other approaches employing occupant position sensors will also be discussed.

b. Air bag membrane induced occupant injuries constitute the second potential injury mechanism of a deploying air bag which may ensue after the initial air bag breakout phase of deployment. If an occupant encounters an air bag before the air bag has achieved its pressurized equilibrium shape, the slack air bag fabric will normally drape around and against that occupant's impinged anatomy. The extent of draping will rapidly diminish as a result of the continued flow of gas from the inflator into the air bag and the reduced air bag volumetric capacity resulting from the occupant's impinged body volume. As a positive air bag internal pressure is produced a membrane stress, a tension in the air bag fabric surface, will result to contain the positive internal air bag pressure. The rapidly increasing combination of air bag surface tension and internal air bag pressure produces a catapult like action on body components submerged below the surrounding unimpinged air bag surface. If the air bag had initially draped over the upper torso and neck of the occupant, similar to a water wing, or had it draped over head and neck or head alone, rearward catapult like forces on these engulfed regions of the anatomy may produce violent differential motions of these body parts relative to the non engulfed or more massive regions of the occupant. Thus, excessive neck moments, shears and tensions could be produced by air bag membrane loading should the occupant be in the path of an expanding air bag before the air bag has achieved its pressurized state. The arm of an adult may also be thrust off the inflating air bag by membrane action. The flailing arm may subsequently impact adjacent car components such as side door, upper A-pillar, windshield and/or its header, rear view mirror or center console at high speed causing hand or arm injuries. If the occupant is a small child undergoing the membrane loading of the air bag, the child's engulfment may produce high speed rearward propulsion into or over the seat back or upward propulsion into the upper windshield or roof area. The earlier in the air bag deployment event that the occupant encounters the unfolding air bag, the more violent the effects of ensuing membrane loading are likely to be. "Smart" air bag systems, which will be discussed later in this paper, are the most probable source of relief for reducing excess occupant loads resulting from membrane effects.

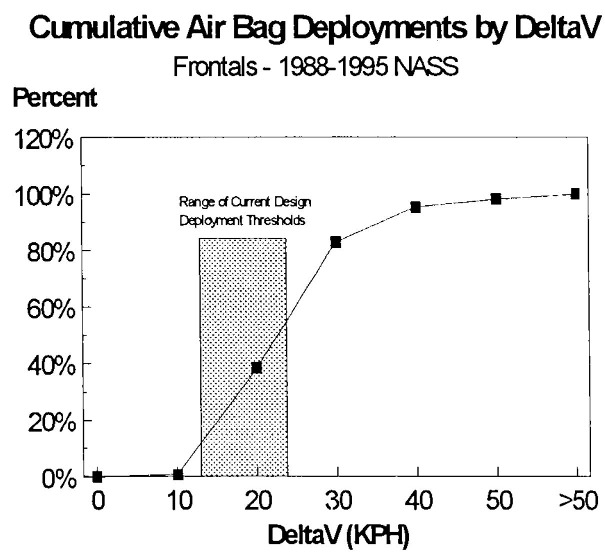
c. Bag slap injuries are created by the occupant being impacted by the air bag fabric during deployment. These injuries are normally relatively minor creating skin redness, abrasions and swelling at the site of air bag initial engagement. These injuries are most frequently located on the wrist, forearm, neck, chin or face. Although the vast majority of bag slap injuries are minor, more serious injuries have been produced when the eye socket area is impacted by fast moving fabric of the deploying air bag. The majority of bag related eye injuries are corneal abrasions which generally do not have long term consequences but some have produced more serious internal eye injuries such as retinal detachments requiring surgical interventions. From tests performed with nine different 1990 model year design driver air bag systems it was found that maximum speeds of the leading surface of the deploying air bags varied between 98 and 211 MPH among the different designs tested. The maximum speed generally occurred at or before eight inches of rearward air bag motion corresponding to about half of its maximum aftward motion prior to full internal bag pressurization[3]. For drivers it is thought that arm abrasion frequencies may be reduced by lower grip positions on the steering wheel, such as at the 4 and 8 o'clock positions on the rim when possible. Air bag internal tethering is also believed to reduce frequency of driver upper torso, neck and face abrasion.

Driver Air Bag Aggressiveness

From accident investigations in the United States there have been several circumstances which have been identified as correlated with incidence of adverse effects produced during early stages of air bag deployment. The violence of driver air bag punch-out has been identified as the probable cause in several serious injuries and fatalities. Generally the drivers who experience the most serious punch-out consequences have been of smaller stature, engaged in low speed frontal crashes producing crash velocity changes near the deployment threshold of the air bag. These smaller drivers most frequently have had their seats set in the most forward seat track position in order to reach driver pedal controls and to gain the elevation afforded at this position by many seat track designs.

Near threshold deployments are normally initiated only after significant forward translation of an unrestrained occupant. This is particularly true in low speed narrow object (pole) crashes or cases in which only limited area of frontal engagements occur. Prior to crash the smaller stature driver is located much closer to the air bag than the 50th percentile male and therefore is in greater hazard from the late deployments associated with low speed crashes. Punch-out injuries to the upper torso such as flailed chest, damage to internal organs and major blood vessels have been seen in several of these short driver fatalities. Other of the short driver fatalities have been associated with hyper extension of the neck which produced basal skull fractures, cervical spine fractures and spinal cord disruptions. The neck hyper extension may also be a product of bag punch-out if the driver's head were engaged by the air bag cover during the burst out of the air bag. In some cases the explanation may be that membrane forces acting on the head/neck region after bag punch-out and during the bag filling phase may produce a powerful differential motion of the head relative to the thorax, producing severe neck hyper extension.

Potential reductions of the incidence of these small stature driver injuries and fatalities would obviously result if the crash threshold speed for air bag deployment were raised to a level at which an unrestrained driver would be subject to only a moderate level of injury from head strikes to the steering wheel assembly. Current deployment thresholds were originally selected based on anticipated facial fractures produced in such head strikes of the unrestrained. Modern steering wheel rims and wheel hub locations softened by the undeployed packaged air bag may have increased the compliance of the modern steering wheel assemblies under conditions of head impact and now allow higher head impact speeds to match the threat of facial fractures of older steering assembly designs. With the additions of improved instrument panel padding which have been made in recent years, the unrestrained passenger may also be able to tolerate some increase in air bag deployment threshold. Figure 1 illustrates the frequency of air bag deployments experienced in the air bag fleet as a function of crash induced velocity change (delta-V). Since the frequencies of crashes as a function of crash velocity rises very rapidly at lower speeds, a significant reduction in the number of air bag deployments would follow a moderate increase in design air bag deployment threshold. Another approach to reducing the number of deployments, already employed by some car manufacturers, raises the deployment threshold when the occupant is belted.



Other methods of amelioration of the threat to short drivers are focused on reducing the onset rate of flow of inflating gas into the air bag during the punch-out phase, the earliest stage of air bag inflation. In so doing the initial energy level and speed of the cover and emerging air bag can be reduced thereby reducing the threat to a driver who is in close proximity or contact with the bag at deployment initiation.

This reduction of inflator gas onset has been demonstrated to reduce critical injury measures for dummies in contact with the air bag module at time of deployment initiation[4,5]. By reduction of the quantity of propellent in a given inflator design, a lowered quantity of gas will be provided and lower gas flow rates can be appreciated. This diminishes available punch-out energy, but at the same time, without design changes in bag volume or venting, extends the air bag's time to attain full inflation and therefore reduces the advantage of earlier imposition of occupant restraint. Another deficit of reduced total gas mass available to the air bag during driver restraint is the reduction in the maximum total kinetic energy which may be absorbed by the air bag during its venting under occupant loading. This indicates that the crash severity at which unbelted occupant protection could be provided would be reduced with the heavier drivers suffering the greatest reductions.

Other approaches to achieve reduced onset of gas flow rate involve delayed ignition of the second of two separated propellent cells within the inflator. The extent of ignition delay for the second propellant cell can be used in conjunction with designed fixed burn rate of the first propellant cell to provide a tailored gas flow rate during the initial and later stages of bag inflation. There are inflator designs which can provide only a single shaped gas flow as a function of time after inflator initiation which require the use of only one electric initiator and other designs which can provide variable flow rates by employing a separate variable ignition timing for a second initiator to ignite the second propellant cell. Similarly, hybrid stored gas systems designs exist which employ two pyrotechnic installations. The first is used to activate a canister diaphragm rupture mechanism and the hot gaseous products of its combustion are then mixed with the stored gas escaping from the high pressure canister thus adding thermal energy to this exiting gas. The second pyrotechnic is a hot gas generant located within the stored gas canister which upon its initiation supplements the stored gas and adds thermal energy. The controlled timing of this second pyrotechnic can thus be used to tailor the flow rate from the hybrid stored gas inflator into the air bag.

There are obvious drawbacks to approaches which add complexity and introduce additional components which must function appropriately to provide the desired reduced onset flow rates and subsequent controlled rates for mass flow into an air bag. Maintaining the reliability achieved in a simpler inflator requires higher individual component reliabilities when components are added to the inflator design function. It is also certain to add to the cost of the inflator.

At least one air bag system supplier has empirically demonstrated a significant reduction in air bag deployment aggressivity for driver air bag modules while still achieving current FMVSS 208 unrestrained occupant protection [5]. The reduced threat to out of position occupants who are in contact with the air bag at deployment initiation has been demonstrated in separate series of tests to evaluate two different air bag module changes. One of these changes, which alone has shown significant reductions to the aggressivity measurements in several of the SAE/ISO out of position recommended test occupant positions already tested, is a new inflator producing reduced onset gas flow during the initial 20 milliseconds after initiation. The second independent approach employed a conventional high onset gas flow rate but involved a re-engineering of the driver module. In static test the re-engineered module almost halved air bag pressure at cover tear and substantially lowered air bag speed during its emergence from the opened cover.

A non azide propellant pyrotechnic inflator which is smaller and lighter than current azide inflators has been developed for the driver air bag system of a small stiff structured car. The shortened frontal crush which characterizes a small stiff car produces a particularly difficult design challenge for an air bag to meet current FMVSS 208 required protection of unbelted occupants in frontal crash. By past practice even greater air bag aggressivity would be expected in an air bag for this small car than one designed for a larger car offering a more favorable frontal crash pulse. Tank testing of this new inflator reveal the desired low onset flow characteristic for reduced punch-out phase air bag aggressivity.

Limited static out-of-position testing with a Hybrid III 50th percentile male in the chest on module position indicate significant reductions in chest deflection, viscous criteria (V*C), and upper neck injury measures when comparing the low onset flow non azide inflator module to the conventional azide inflator driver air bag module. Limited HYGE sled testing in the chest on module position with the Hybrid 5th percentile female dummy under a moderate severity crash pulse again demonstrated reductions in V*C, chest acceleration, chest deflection, and HIC when comparing the non azide to conventional azide driver module. Comparable FMVSS 208 test performance of the azide and non azide modules were also seen in HYGE sled tests. It therefore appears that an non azide production inflator which is used in two MY 96 production vehicles offers high potential for reduced threat to small stature drivers in low speed crashes who constitute the majority of known driver fatalities associated with punch-out or membrane effect induced injuries to the basal skull, brain stem and/or upper cervical spine from hyper extension of the neck and multiple bilateral rib fractures and/or internal thoracic organ injuries from direct punch-out loads. This new inflator technology is fortunately free of added production or functional complexity as compared to current conventional inflators.

The second independent approach which reduced punch-out aggressiveness employed a prototype driver air bag module utilizing conventional high onset gas flow rate but involved a re-engineering of the air bag storage dimensions and air bag fold. In static test the re-engineered module almost halved air bag pressure at cover tear and substantially lowered air bag speed during its emergence from the opened cover while increasing the bag's radial expansion rate. Utilizing the ISO recommended out of position occupant test condition of a Hybrid III 50th percentile male dummy with his chest horizontal and in contact with the driver air bag module mounted in a horizontal stiff steering wheel, static deployment tests were conducted. These tests compared the prototype module performance to that of a current system. The results showed the prototype to provide significant reductions in chest injury measures, a 25% reduction in chest deflection and a 45.5% reduction in V*C. The improved driver air bag module is anticipated to be offered to OEMs in time for application in a number of 1998 models.

Passenger Air Bag Aggressiveness

Incidents observed and studied from the NHTSA Special Crash Investigation Program and Fatal Accident Reporting System have revealed instances of child fatalities caused by passenger air bag deployments. Several of these fatalities involved infants in rear facing child restraints located on the right front passenger seat during air bag deployment. The other fatally injured children were three or more years old and were unbelted or improperly restrained in the front right passenger seat. In nearly all cases it is known that pre-crash braking was applied to the car by the driver.

The crash induced velocity changes to the car were so low in most of these cases that the child fatalities would have been very unlikely had the passenger air bag not deployed. Passenger air bag equipped cars and rearward facing child seats post warning to parents not to place this infant restraint in the front seat. These warnings originated after NHTSA conducted tests showing the threat of fatality to children in these infant seats when subjected to a passenger air bag deployment. Similarly, NHTSA recommends all children be located in the rear seating positions of cars to insure their greatest safety. All occupants are warned to employ seat belts, and in the case of children located at a passenger position equipped with an air bag, the car seat has been recommended to be adjusted to its most rearward location on the seat track. NHTSA in no way accepts as inevitable the present situations in which children are injured fatally.

In addition to continuing its public awareness campaigns warning of the hazards, technical improvements may be necessary in the performance of passenger air bags so as to make them less of a hazard.

The injury mechanisms of the passenger air bag are essentially the same as those of the driver air bags. The punch-out phase, the membrane phase and bag slap are the threatening phenomena during deployment of passenger air bags. However passenger air bag locations and deployment trajectories vary among the car models. Upper mounts are located on the upper surface of the instrument panel and are designed to initially deploy directly towards the passenger or upward, parallel to the windshield during their expansion rearward and down towards the passenger. Mid mounts are generally located near the prior location of the glove box in the instrument panel in front of the passenger. These passenger air bag installations provide a deployment trajectory directly towards the passenger.

Child passenger fatal injuries have been produced in the brain and skull from impacts with the passenger bag covers during punch-out. Instances of hyper extension of the neck have been seen with associated basal skull fracture, brain stem damage, cervical vertebral fracture/displacement, and spinal cord transection which were likely caused during air bag cover impact to the head and head contact with the emerging air bag during punch-out or during membrane loading phases of deployment. Impacts of the head against the hard regions of the windshield and header structures have been produced during air bag deployment by membrane air bag forces driving the child upward.

Changes to the passenger air bag system which may reduce its deployment aggressiveness are similar to those proposed for driver systems. Reduced air bag break-out energies and reduction of air bag membrane effects are needed. These objectives may be obtainable through use of lowered onset flow inflators and/or sheltered air bag locations where components of the air bag module and air bag are less likely to engage the passenger during the air bag break-out phase.

As in the case of driver air bag modules, an industry air bag supplier[5], has indicated attainment of large reductions in passenger air bag aggressiveness resulting from improvements in packaging and control of passenger air bag contour during deployment. This improved passenger module is currently planned for introduction by at least one OEM during the 1998 model year.

A new concept passenger air bag module was introduced in a model year 1994 production sports car line. This module employs a two stage inflator with a single igniter squib. An approximate 15 ms delay is engineered in the pyrotechnic design before second stage burn begins. The first inflator stage incorporates larger, slower burning fuel pellets and the second stage is filled with more rapid burn small pellets. The module design incorporates a unique direct vent of a portion of the inflation gas into the instrument panel interior during initial stages of inflation as bag break-out proceeds. The combination of the slower burning first stage of the inflator and the available venting of inflator gas even before significant air bag unfolding occurs provide the desired low onset rate of flow of gas entering the air bag. After the folded air bag bursts the module cover and begins its emergence into the passenger compartment, the venting area into the instrument panel interior becomes immediately available. Thus the vent located between the inflator and air bag may serve as a pressure relief mechanism if an out of position occupant interferes with unimpeded unfolding and expansion of the air bag. In more conventional design, venting is restricted to that time when the vent ports or vent panels of the air bag have been uncovered during the bag unfolding phase. This same mid-mount passenger air bag module incorporates a special air bag folding approach which initially directs the bag expansion downward for early engagement of the lower torso followed by expansion to the chest and head regions in a further attempt to reduce the threat to many out of position passengers.

"Smart" Air Bags

A highly desirable goal is the elimination of instances of excessive driver and passenger loading produced by deployment of air bags. This might be attainable if in each air bag deployment the operating characteristics were tuned to the status of the occupant and the car crash severity for which the air bag protection was desired. Thus, the term "smart air bag" was brought into the vernacular of automotive jargon.

A smart air bag system should be able to automatically absorb and process metrics providing information on occupant size (mass), occupant position relative to the air bag, crash severity and direction of crash forces, and, in the case of passenger systems, the presence of a rearward facing child seat and based on these inputs at the time of air bag deployment initiation decision, reliably suppress deployment or tailor the air bag deployment characteristics to provide appropriate and safest occupant protection. To illustrate the harsh demands placed on a smart system: If a driver or passenger were in contact with or in too close a proximity to the air bag, either a drastically reduced inflator output or complete absence of initiation would automatically occur; If a large sized occupant was identified as being at an adequate seated distance, a more rapid gas flow and increased total gas mass would be supplied to the air bag than would have been the case with a lighter occupant; If crash severity were low, less energy would be imparted to the air bag deployment than when the crash severity threat was greater. Such scenarios imply that the smart air bag system must be preprogrammed to recognize and analyze all supplied occupant and crash information to determine and implement those deployment strategies that are uniquely most beneficial for the occupant.

Some authorities refer to a passenger air bag suppression in the absence of a passenger as being a smart air bag since it may spare the owner the cost of replacement with little or no safety benefit loss. But approaches to achieve even this goal require addition of occupant sensing. The occupant sensing systems in existence or nearing production readiness are varied. The simplest are aimed only at identifying the presence of an occupant while the more complex may identify seat occupancy to be a rearward facing child seat, an adult or child, proximity of the occupant relative to the air bag, and motions of the occupant toward the air bag.

Some occupant sensing approaches employ only the signal from a latched seat belt. A more sophisticated belt related sensor employs play out of the belt webbing as a surrogate of occupant girth and thus a metric of occupant mass. Another occupant sensing approach employs load distribution and magnitudes on the car seat to identify occupancy, proximity, and mass. Another approach to estimating occupant weight employs a capacitance measurement of the occupant providing essentially a surrogate measure of the occupant water content.

A system which has been under evaluation by OEMs employs fused output recognition from both acoustic and infrared transmissions to distinguish the presence of a rearward facing child seat, human occupancy, and proximity of occupant to the passenger air bag. Another system employs a single optical sensor type to establish human occupancy and position. At least one system in development employs low power radar in combination with ultrasonic and infra red sensing to identify occupant presence and proximity. The unique radar type sensor located in the seat back will not only track occupant location but may also provide physiological measurements relating to a drivers state of alertness for future applications in accident avoidance efforts.

Even the more complex occupant sensors described above can do no more than abort passenger air bag deployment in the cases of rearward facing infant seats, too close an occupant, or absence of an occupant unless electro-mechanical crash sensing is supplemented or replaced by full electronic crash sensing and means are provided to modulate air bag inflation. Techniques for achieving a fixed staged gas flow or a programmable gas flow from passenger air bag inflators are the same as already described in the section addressing driver air bag aggressiveness. In order to tailor the inflation characteristics as a function of occupant state requires inputs to the inflator flow control logic from the more sophisticated occupant sensors which estimate occupant proximity and weight. To add consideration of the crash severity and direction requires inputs from a sophisticated full electronic sensor system utilizing electrical output proportional to instantaneous acceleration, such as is currently produced in single point sensors utilizing piezo resistive etched silicon accelerometers. Algorithms to reflect OEM philosophy would be added to those already employed for deployment initiation based on occupant state information as well as crash severity.

Another approach to dramatic reduction of air bag induced injuries and fatalities involves implementation of crash sensor technology which could achieve an earlier determination that an impact in progress will exceed the severity at which air bag protection is desired for occupant protection. As previously discussed, current crash sensor technology provides air bag initiation late in a low speed or partial engagement crash, typically in bumper underride or pole impact. Fortunately, an economical radar technology is in prototype development which, with the current efforts of an established major producer of single point crash sensors and other electronic automotive systems, is expected to soon be employed to add an algorithm, related to measured pre-crash closing speeds, to those algorithms already employed in the sensor so as to provide earlier reliable confidence that deployment will be required before completion of the crash event. Success in these endeavors may signal economical and reliable means to allow a slower, less aggressive air bag deployment without jeopardy to the full protective potentials of air bags.

CONCLUSION

It is clear that truly smart air bags require substantial additional development efforts before substantial implementation in production cars can be anticipated. The interfacing of electronic crash sensors with occupant state sensors and controllable staged flow rate air bag inflators is a formidable task requiring maintenance of high reliability currently associated with air bags. Can it be accomplished is not in question, but rather the question is will the efforts and costs be expended which are required to achieve the full goal of smart air bag systems?

The goals to be achieved are worthy of re-evaluation. Each of us recognize the need to reduce the magnitude of injuries and the number of fatalities associated with air bag deployments. Is it now or ever to be possible to completely eliminate all such events and still appreciate the overall occupant crash protection benefits achievable with air bags? From the vantage point of today, the answer must be no. However, even today, it appears feasible, and perhaps appropriate, that we may tolerate some reduced system reliabilities if the net result could be major reductions in incidence of severe injuries and fatalities caused by air bag deployments.

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THE INFLUENCE OF FORCE LIMITER TO THE INJURY SEVERITY BY USING A 3-POINT BELT AND DRIVER AIR BAG IN FRONTAL COLLISIONS

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ABSTRACT

The number of the new registered passenger cars - with combined 3-point belt / driver side air bag restraint systems - is steadily increasing. The air bag offers an additional protection of the head, neck and torso. However, the thorax responses and kinematics by crash simulations show that a time adjustment of the shoulder belt and the air bag effectiveness is needed; furthermore correlations of the response and injury severity are required. For these aims 30 mph frontal collisions were performed with 14 human cadavers. The subjects' thorax was instrumented with the 12-accelerometer array and two chest bands and restrained with 3-point belt, driver air bag or 3-point belt - driver air bag combination.

The results show, that by using a combined 3-point standard belt (6% elongation)/driver side air bag, the thoracic injury pattern is located under the shoulder belt. The same observation was found by using belts with 16% elongation in combination with the driver side air bag. The chest contours show, that the 3-point belt causes high local compression of the chest along the shoulder belt path, which suggests there would be a high risk for thoracic injuries. On the other hand, the air bag, used by itself as a passive restraint system, distributes forces uniformly over the front of the chest.

This study asks if it is possible to obtain both the thoracic injury mitigating benefits of an air bag only restraint and the all-impact-direction benefits of the belt from a restraint system combination by adding a force limiter to the shoulder belt. For this reason, tests with force limiters were performed. The force limiter with the level of 4 kN showed, through examination of the chest band contours, a more bag like uniform compression of the chest and reduces therefore the injury risk. Non or AIS 1 injuries were found in the cervical spine and AIS 0 to AIS 1 was observed in the thorax, even in the age range 60 to 65 years. The chest compression amounts 4 to 8 cm, and the resultant vertebral accelerations were in the average 30 to 40 g's; at these levels the thorax of the most cadavers remained uninjured or an injury severity of AIS 2 was observed.

It is concluded, that when a driver side air bag is combined with a 3-point belt system that limits the torso belt loop load to 4 kN, additional injury mitigation benefits for both the cervical spine and the thorax are obtained in frontal collisions.

INTRODUCTION

The number of the new registered passenger cars with combined 3-point belt and supplemental driver side air bag (U.S. bag or EURO bag) - is steadily increasing. Real accident investigations (Huelke et al., 1992, Huelke et al., 1994, Otte, 1995) favors the 3-point belt - driver side air bag combination. Own investigations of frontal collisions with cadavers protected by 3-point belt or/and driver side air bag at impact velocities of 48 to 55 km/h (Kallieris et al., 1994) show a high local compression of the chest along the belt path when using 3-point belts; by the use of a driver side air bag the forces are uniformly distributed over the front of the chest. The local compression of the chest exists for all belt elongations available at the time (6% to 16%). Therefore, the idea was to use a force limiter. Own experiences with force limiters (torsion bar) in the 70ies (Schmidt et al., 1975) have shown a distinct reduction of the chest force, but the high head displacement and the following head impact against the steering wheel or the dash board lead to problems; similar observations were made by Foret-Bruno et al.(1978) in real accident investigations. These problems do not exist if an airbag is additionally used.

The optimization of an air bag - 3-point belt restraint was discussed by Kompass (1994), while Mertz et al. (1995) has investigated the effect of limiting shoulder belt load with air bag restraint by using Hybrid III dummies in frontal collisions.

A shoulder belt force limitation of 4 kN seemed to be suitable according to earlier cadaver investigations and computer simulations made by NHTSA (Kallieris et al.,1995) to harmonize between 3-point belt and driver side air bag for simultaneous prevention of cervical spine and chest injuries. This was tested by using human cadavers.

METHOD

Test Subjects

The test subjects were 14 unembalmed cadavers in the age range 20 to 63 years. The research content and the procedures govering the procurement, treatment, and disposition of human surrogates used in this program are conform to all requirements of University of Heidelberg's Ethics Commission.

Test Equipment

The tests were performed on the University of Heidelberg's deceleration sled. Mounted to the sled was the front part of a passenger compartment of a mid-sized car. Test subjects were positioned in the driver's seat and restrained by either a 3-point belt (6% or 16% elongation), a driver side air bag-knee bolster, or a 3-point belt (16% elongation) with supplemental driver side air bag combination; belt force limiters with the level of 4 kN were used. Figure 1 illustrates the experimental configuration. Frontal collisions were simulated with impact velocities of about 48 to 55 km/h and a trapezoidal deceleration pulse with a mean value of 14-17 g, in one test the deceleration was 20 g. Table 1 shows the test matrix.

Instrumentation

In all tests, the subject's thorax was instrumented with a 12-accelerometer array (Robbins et al. 1976, Eppinger et al.1978) and a triaxial accelerometer array at the clivus. The shoulder belt force was also measured in some tests.

To measure thoracic contours during dynamic over the time, the "chest band" was used (Eppinger 1989, Kallieris et al. 1994). The chest bands were mounted at the level of the 4th and 8th rib of the cadaver.

Autopsy- Injury Severity

For each cadaver, a full autopsy with a detailed investigation of the vertebral column, was performed. The injuries were coded according to the AIS 1990.

Data Analysis

The data were recorded in analog format and were digitized at 10,000 samples per second and subsequently filtered with a digital Butterworth filter channel class 180. The thoracic deformation contours were computed using the RBAND-PC Software from NHTSA. A program was equipped to evaluate the deformation of the front chest wall in relation to the fixed vertebral column.

To evaluate the Viscous Criterion (VC) at the level of the 4th and 8th rib, the deformation velocity was defined through differentiation of the deformation, the deformation-time-histories and the deformation velocity-time-histories were smoothed. For the evaluation of the compression, the chest depth was used (instantaneous deformation/chest depth, Lau and Viano 1986).

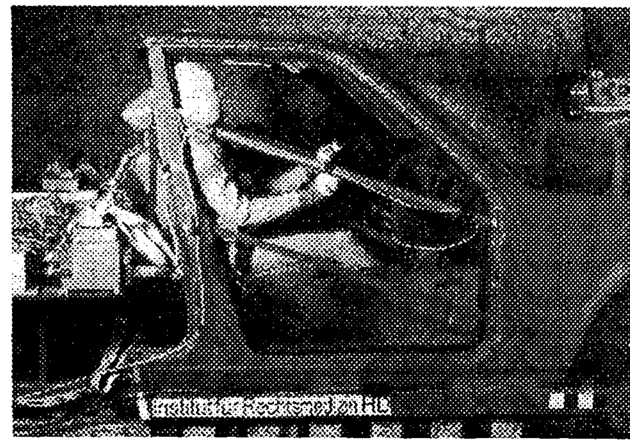

Figure 1. Test configuration

Table 1.
Test condition and specimen data (f.l. = force limiter)

Run No.	restraint system	velocity [km/h]	sled decel. [g]	age years	sex	chest circumf [cm]
1.1	3-point belt	48	16	34	m	92
1.2	3-point belt	55	17	20	m	95
1.3	3-point belt	50	20	37	m	93
1.4	3-point belt	49	15	29	m	78
1.5	3-point belt	48	14	52	f	78
1.6	driv. air bag - knee bolster	47	16	31	m	---
1.7	driv. air bag - knee bolster	49	17	25	m	88
1.8	driv. air bag - knee bolster	47	17	38	m	95
1.9	driv. air bag - 3-point belt	48	14	47	f	100
1.10	driv. air bag - 3-point belt	48	14	32	m	102
1.11	driv. air bag - 3-pt. belt - 4 kN f.l.	47	14	63	f	95
1.12	driv. air bag - 3-pt. belt - 4 kN f.l.	48	14	58	m	95
1.13	driv. air bag - 3-pt. belt - 4 kN f.l.	49	15	50	m	95
1.14	driv. air bag - 3-pt. belt - 4 kN f.l.	49	15	39	m	93

RESULTS

Mechanical Responses

<u>Shoulder Belt Forces</u>

In order to diminish the chest compression in the belt path and to keep the air bag specific compression pattern when using a 3-point belt - air bag combination the use of the force limiter was indispensable. Generally the shoulder belt force amounts 7 - 8 kN in a frontal collision with Δ v =50 km/h and a mean car deceleration of 15 g when a 3-point belt protected 50% male is used (Kallieris et al., 1982). Pretests with dummies protected by a 3-point belt - air bag combination with a force limitation of about 4 kN have shown that an air bag specific chest compression behavior is achieved (Kallieris et al., 1995); by the use of cadavers this leads to the prevention of neck and chest injuries.

Figure 2 shows force-time histories of the shoulder belt, by using a 3-point standard belt, 3-point standard belt wit force limiter of 4 kN and driver side air bag when cadavers were used. Additionally to the force-time histories, the start point of the effect of the force limiter is shown. The limiter operates with a delay of about 25 ms after the start of the belt force; at this time the shoulder belt force of 4 kN is reached. The plastic elongation of the limiter amounts between 6 to 10 cm and depends on the body weight of the test subject.

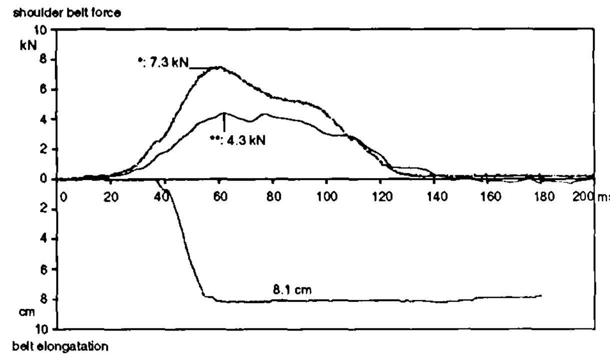

Figure 2. Shoulder belt force-time histories by using two different restraint systems
***: cadaver, standard belt without force limiter (run 1.1)**
****: cadaver, standard belt with force limiter and driver side air bag (run 1.12)**
lower part: elongation-time history by using the force limiter

<u>Accelerations</u>

Table 2 summarizes the head and the chest acceleration as single and mean values of the subjects used. The maxima of the resultant acceleration of the clivus amount between 37 and 95 g, the corresponding 3 ms values are between 34 and 69 g.

The highest maximum of 95 g was measured at a car deceleration of 20 g and the highest 3 ms value of 69 g was measured when a head impact against the steering wheel was resulted.

The lowest values (max.: 37 g, 3 ms: 34 g) were observed by using a 3-point standard belt with force limiter combined with a driver side air bag in cadaver testing.

Different observation were made for the chest acceleration according to measuring location and the restraint system used. Generally lower maxima and 3 ms values were observed at the chest when using a 3 point belt, 16 % elongation or 3-point standard belts with force limiter combined with a driver side air bag, an exception was the rib acceleration, which showed very similar mean values for the four different restraint systems used.

Table 2.
Acceleration of head and chest - neck and chest injury severity
***1.1 - 1.5*: 3-point-belt *1.6 - 1.8*: driver air bag - knee bolster *1.9, 1.10*: driver air bag - 3-point belt**
***1.11 - 1.14*: driver air bag - 3-point. belt - 4 kN force limiter (for more information refer to table 1)**

	Head Res		U. St.x-dir		L. St.x-dir		8.R.l. x-dir		8.R.r. x-dir		Th1 Res.		Th12 Res		TAIS	NAIS
	max	3ms	max	3ms	max	3ms	max	3ms	max	3ms	max	3ms	max	3ms		
1.1	41	38	31	28	32	29	20	17	40	37	31	28	50	44	0	0
1.2	55	48	---	---	---	---	---	---	---	---	58	34	---	---	2	2
1.3	95	63	43	28	---	---	---	---	---	---	48	45	---	---	3	0
1.4	71	63	45	30	39	35	47	45	32	25	40	37	32	30	2	0
1.5	47	44	37	33	35	29	57	54	42	37	35	29	49	47	2	2
mean	**62**	**51**	**39**	**30**	**36**	**31**	**41**	**39**	**38**	**33**	**42**	**35**	**44**	**40**		
standard deviation	22	11	6	2	5	3	19	19	5	7	11	7	10	9		
1.6	86	69	33	32	52	48	34	31	29	28	38	36	58	49	0	0
1.7	58	51	54	51	44	38	42	40	35	33	56	51	49	46	0	0
1.8	42	39	53	50	74	65	27	19	37	35	48	46	52	50	0	0
mean	**62**	**53**	**47**	**44**	**57**	**50**	**34**	**30**	**34**	**32**	**47**	**44**	**53**	**48**		
standard deviation	22	15	12	11	16	14	8	11	4	4	9	8	5	2		
1.9	44	41	45	27	50	16	62	57	40	37	34	32	45	42	2	2
1.10	41	38	30	21	23	19	46	42	33	30	35	32	31	30	1	0
mean	**43**	**40**	**38**	**24**	**37**	**18**	**54**	**50**	**37**	**34**	**35**	**32**	**38**	**36**		
standard deviation	2	2	11	4	19	2	11	11	5	5	1	0	10	9		
1.11	39	35	31	30	31	26	34	32	38	34	37	32	30	28	1	0
1.12	37	34	36	27	32	19	42	38	35	31	30	26	33	30	2	1
1.13	43	41	38	26	35	29	41	38	38	35	39	36	40	35	0	1
1.14	52	50	42	32	41	34	37	34	37	32	37	35	32	31	0	0
mean	**43**	**40**	**37**	**29**	**35**	**27**	**39**	**36**	**37**	**33**	**36**	**32**	**34**	**31**		
standard deviation	7	7	5	3	5	6	4	3	1	2	4	5	4	3		

Thoracic Deflections

Figures 3 and 4 show examples of chest contours, deformation-time and VC-time histories at the level of the 4th and the 8th rib of the cadaver when a 3-point belt combined with driver side air bag (run 1.9) and a 3-point belt with a 4 kN force limiter combined with driver side air bag (run 1.12) was used. By using the 3-point and air bag belt without force limiter a clear local compression at the both levels is observed (Figure 3), while by the use of the combined restraint system with force limiting devise the result was a more bag like, uniform, compression of the chest (Figure 4).

The compression at the level of the lower rib was with 7 cm higher than at the level of the 4th rib with 5.6 cm. A corresponding higher VC value was observed at the level of 8th rib with 0.41 m/s in comparison to the 4th rib with 0.2 m/s (Figure 4). Independent of the use of force limitation higher compressions were observed at the level of the 8th rib in comparison to the level of the 4th rib (Figure 3 and 4).

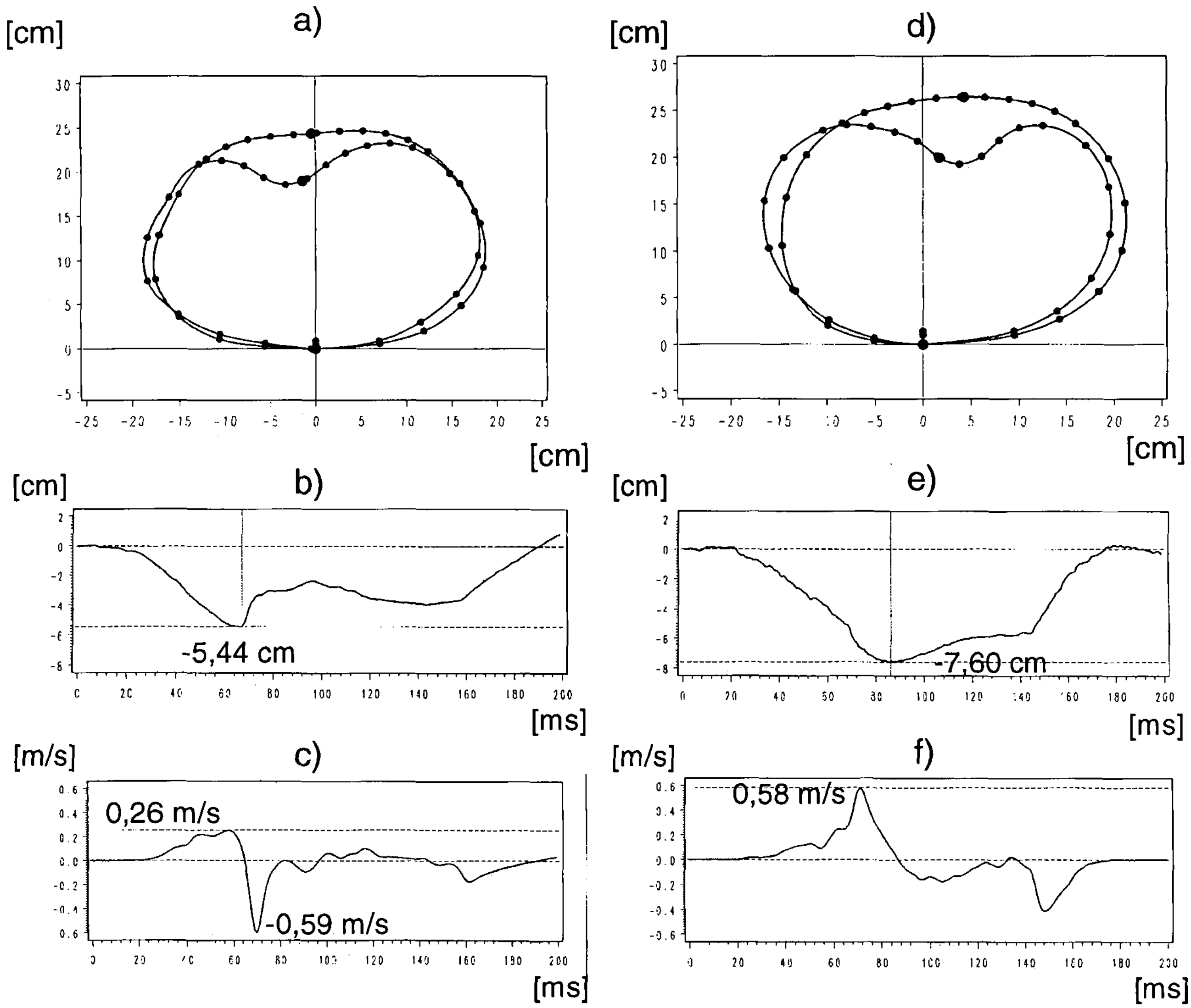

Figure 3. Cadaver chest measurements. The cadaver (1.9) was restrained by a 3-point standard belt and a driver side air bag.

a) - c): level of the 4th rib d) - f): level of the 8th rib
a), d): Thoracic deformation contours of the initially unloaded state and the state of maximum deformation.
b), e): deflection-time histories
c), f): VC -time histories

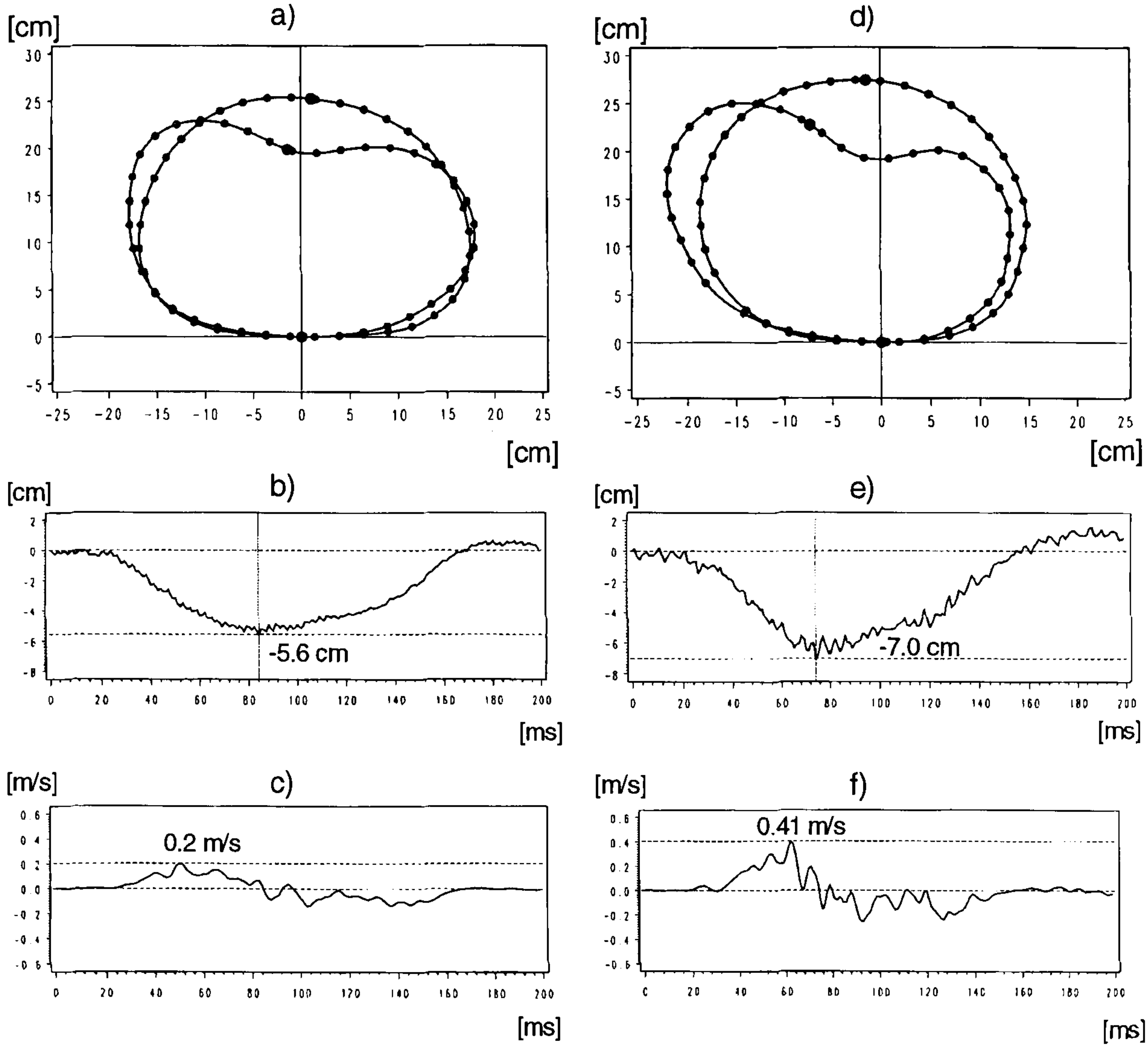

Figure 4. Cadaver chest measurements. The cadaver (1.12) was restrained by a 3-point standard belt, 4 kN force limiter and a driver side air bag.

a) - c): level of the 4th rib d) - f): level of the 8th rib
a), d): Thoracic deformation contours of the initially unloaded state and the state of maximum deformation.
b), e): deflection-time histories
c), f): VC -time histories

Medical Findings

By using 3-point standard belts skin abrasions over the shoulder and the hip bones were observed. No bony injuries were found on the cervical spine. Three cases showed lacerations of the ligamenta flava C7/Th1 and Th1/Th2 (AIS 2) with additional hemorrhages of muscles and intervertebral discs (AIS 1). Figure 5 shows an example of a lacerated ligamentum flavum at the level C7/Th1 caused by frontal flexion of the head-neck unit (run 1.2). Only two cases were found with hemorrhages (strains) of muscles and intervertebral discs with AIS 1. Nine of the 14 investigated vertebral columns remained uninjured (Table 2).

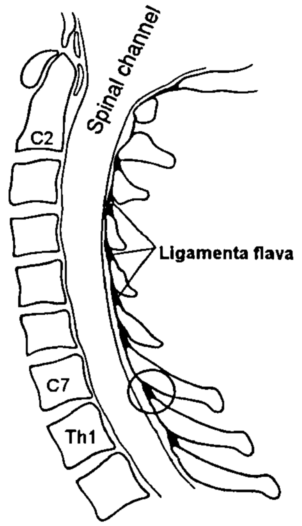

Figure 5. Middle - Sagital view of the spinal column, Run No. 1.2, with lacerated lig. flavum C7/Th1.

Six of the 14 cadaver tests showed no bony thoracic injuries. In six cases rib fractures were observed, two cases of them had additional sternum fractures. The number of rib fractures was between one to five. Two cases showed only a sternum fracture, the fractures were located underneath the shoulder belt path and were generally infractions. Typical injury patterns of the thoracic skeleton caused by the use of 3-point belt or air bag are shown in Figure 6. The injury severity amounted from AIS 1 to AIS 3 (Table 2).
Only in one case, a liver rupture with AIS 3 was observed, the impact severity was 50 km/h with a sled deceleration of 20 g (subject 1.3).

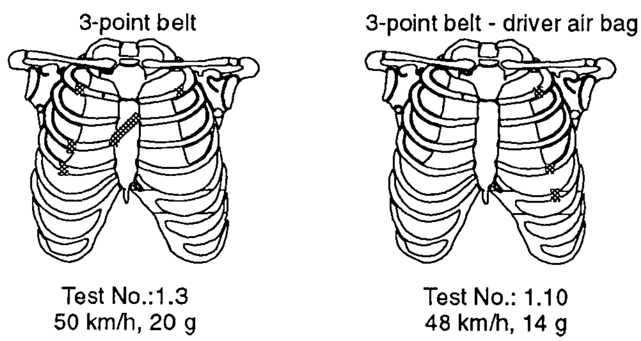

Figure 6. Example of injury pattern by using 3-point belt or air bag.

MECHANICAL RESPONSE - INJURY PATTERN AND INJURY SEVERITY

Table 3 summarizes the thorax deflections, the corresponding VC values and the thoracic injury severity. The first tests using the 16 strain gauges chest band were not completely evaluated. The thoracic deflection amounted between 2.1 cm to 10.3 cm; the highest deflection of 10.3 cm was measured in a small and fat female cadaver (167 cm, 90 kg) where the installation of the chest band caused some difficulties.
The VC values varied between 0.05 and 1.48 m/s, no clear correlations with the deflection values exist.

The thoracic injury includes uninjured cases of AIS 0 and injured cases up to AIS 3. The test with AIS 3 was performed with a higher impact severity (20 g); most of the cases showed the injury severity of AIS 2.

According to the small number of tests one can conclude, that the injury severity seemed to be influenced by the age of the cadaver, the impact severity and the restraint system / or combination used.

Table 3.
Chest deflection, VC values and thoracic injury severity
1.1 - 1.5: 3-pt.-belt (6%)
1.6 - 1.8: driver air bag-knee bolster
1.9, 1.10: driver air bag - 3-pt. belt (16%)
1.11 - 1.14: driver air bag - 3-point belt (6%) - 4 kN force limiter

Test	Deflection [cm]		Viscous Criterion [m/s]		Injury Sev.
No.	4. rib	8. rib	4. rib	8. rib	TOAIS
1.1	7.5	2.7	-	-	0
1.2	5	5	-	-	2
1.3	-	4	-	-	3
1.4	-	3.8	-	0.14	2
1.5	6.0	4.8	0.46	0.30	2
1.6	-	3.2	-	-	0
1.7	2.1	3.8	0.05	0.21	0
1.8	3.5	7.2	0.74	1.48	0
1.9	5.9	7.6	0.17	0.58	2
1.10	5.4	2.4	0.34	0.48	1
1.11	6.8	10.3	0.26	1.1	1
1.12	5.6	7.0	0.20	0.41	2
1.13	6.2	3.0	0.26	0.1	0
1.14	5.9	---	0.35	---	0

DISCUSSION

Most of the personal cars are nowadays equipped with 3 main restraint systems: the 3-point belt, air bag - knee bolster or a 3-point belt - air bag combination. The use of the chest band showed that correct belted occupants (without belt slack) suffered high local compressions of the chest front along the belt path and indirect loading of the cervical spine through the forward bending of the head.

The combination of driver side air bag - knee bolster was not suited to limit the forward movement (x-z plane), a front head impact against the header / windshield occurred (Crandall et al. 1994); however, the force distribution across the whole chest front prevented thoracic injuries.

In order to utilize the advantages of the air bag in the combination with a 3-point belt it is necessary to connect the 3-point belt to a force limiter. Earlier experimental work (Schmidt et al., 1975) and analytical simulations have shown that the force limitation to 4 kN is suited to prevent thoracic injuries if a combination of a 3-point belt / driver side air bag is used.

The reduction of the effective force at the level of about 8 kN by using the standard shoulder belt at the chest front to an average of 4 kN by using a force limiter and the 3-point belt - air bag combination clearly prevents thoracic injuries.

The balance between the 3-point belt with belt force limiter / driver side air bag combination leads to a relief of the cervical spine loading, a more translatoric forward displacement of the head is given. Only one rib fracture, at a typically loaded location when using an air bag (front axillar line) was found in a 63 years old female and no cervical spine injuries were observed, which confirms the translatoric forward displacement of the head. It means that a force limitation to 2 kN is not needed as proposed by Mertz et al. (1995) when using Hybrid III dummies.

Head impacts are typical for real accident investigations because of the generally high belt slack. The lowest average acceleration values and the smallest standard deviations of the restraint systems used are found in cases with combination of 3-point belt (16%) and driver side air bag and 3-point belt (6%), force limiter and driver side air bag in cadaver tests.

The acceleration measured at the thorax in x-direction and the resultants tend to be lower in the average when a force limiter or 3-point belt with a 16% elongation combined with a driver side air bag is used in comparison to 3 -point standard belts or air bag - knee bolster restraint systems. Similar observations are made with the deflections at the level of the 4th and 8th rib and the corresponding VC values. All these values are not specific for the restraint systems used; there exist also no deflection differences at both investigated levels.

The chest contours of the cadaver showed a clearly air bag-like compression behavior if a 3-point belt with a 4 kN force limiter with a driver side air bag was used. The effective force is uniformly distributed across the front of the chest and not locally as it would be by using a 3-point standard belt or belts with a 16% elongation.

The lower force distribution at the chest was noticeable by the thoracic injury severity; cadavers in the age range from 50 up to 63 years remained uninjured or suffered one or two rib fractures in comparison to numerous rib fractures observed by Yoganandan et al. (1993) by using air bag - 3-point belt and air bag - knee bolster systems.

The results show that the combination 3-point belt with a 4 kN force limiter combined with a driver side air bag is suited to prevent thoracic and cervical spine injuries also for older people.

CONCLUSIONS

For the impact conditions investigated in this study, the combination of 3-point standard belt with a 4 kN force limiter and driver side air bag appears to offer significant improvements in overall injury protection over the standard belt system.

ACKNOWLEDGMENT

This study was supported by a grant from the U.S. DOT, National Highway Traffic Safety Administration and by the BMW AG in Munich, Germany, with technical equipment. The authors wish to gratefully acknowledge.

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OPTIMIZATION OF AN INTELLIGENT TOTAL RESTRAINT SYSTEM

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Paper Number 96-S3-O-09

ABSTRACT

The TAKATA Intelligent Total Restraint System, or ITRS, is a controllable vehicle occupant restraint system optimized so as to cost effectively minimize the risk of injury to occupants from crashes. A model based technique is described for optimizing the performance of the ITRS. The ITRS senses occupant weight, occupant position, crash severity, and seat belt usage; and in response to these sensed measurements, controls the individual firing times of a two-stage inflator. A general form of the ITRS also controls inflator module vent area, seat belt force limit, and a seat belt pretensioning. The first stage inflator is experimentally sized so as to not injure the worst case out-of-position occupant, while the combined gas generant loading of both stages is sufficient to satisfy FMVSS-208 requirements. Given the system architecture and associated fixed parameters, occupant injury and injury assessment value (IAV) measures are each modeled as a function of the input and control variables using data gathered from an occupant simulation experiment comprising combinations of occupant size, occupant position, crash severity, and air bag inflation rate. Optimal controls are found by minimizing the IAV model with respect to the system controls subject to constraints that individual injury measures be less than corresponding threshold values. Inflation rate is dependent upon the system state, defined by occupant position, occupant weight, seat belt usage, and crash severity, where crash severity is defined as the injury producing potential of the crash. The air bag firing threshold is found as the maximum crash severity which satisfies the constraints that occupant contact velocity is less than 19 KPH and shoulder belt load is less than 3 KN. Second stage inflator time delay is found which minimizes the IAV subject to the constraint that individual injury measures do not exceed acceptable threshold limits. If an air bag is warranted for a given crash the first stage inflator is fired as soon as possible to allow for the longest possible inflation interval -- or softest possible inflation -- as controlled by the second stage inflator time delay. Results from a driver-side firing threshold study demonstrate the potential for benefit from an ITRS.

INTRODUCTION

Supplemental air bag inflation systems are widely recognized to reduce harm to occupants from frontal crashes **[1]**. These systems typically comprise a single gas generator coupled to single chamber air bag and activated by a crash discriminating sensor having a predetermined firing threshold characteristic. Vent holes in the air bag provide a means for dissipating occupant energy, and are generally tuned for best operation under maximum crash severity conditions. The air bag system supplements a lap/shoulder seat belt by distributing restraint forces over a large area of the occupant so as to reduce peak loads.

The single gas generator must be sufficiently fast and energetic so as to satisfy the restraint requirements of FMVSS 208, and the resulting deployment characteristic can be a source of injury to some occupants in relatively mild crashes. Notwithstanding the overall benefits of such a system, there is a rising concern about air bag induced injuries or fatalities to small or out-of-position occupants, or their extremities, caused by the aggressiveness of the inflator system which inherently results from the legal requirement to protect unbelted occupants in 30 MPH crashes **[2][3][4]**. Since the conventional air bag system is generally limited to a single mode of operation, this dilemma has led to suggestions for modifying the FMVSS 208 requirements to the benefit of belted occupants by enabling softer air bag inflation curves **[5]**.

Concentrated loads from seat belts, particularly shoulder belts, can also cause occupant injury, especially to elderly occupants with frail bones. Such injury can be mitigated with a force limiter -- in series with the belt -- which dissipates occupant energy as the seat belt is deployed. Mertz et al. have suggested limiting shoulder belt loads to better protect elderly occupants **[6]**.

The principle source of injury to an unrestrained occupant in a crash results from the secondary impact of the occupant with the vehicle interior. The purpose of restraint systems is to dissipate the energy of the occupant -- an inertial body having an initial velocity -- relative to a rapidly decelerating vehicle by application of restraint forces to the occupant. The magnitude and duration of the necessary restraint forces depends upon the amount of occupant energy to be dissipated, and the total unrestricted travel distance available to the occupant. Since occupant injury mechanisms are generally related to the levels of force or impulse applied to the occupant, or the corresponding amount of energy absorbed by the occupant, occupant injury can be minimized if the occupant is decelerated uniformly over the full extent of the unrestricted travel distance available to the occupant **[7][8]**.

One method for solving the dilemma of satisfying the requirements of FMVSS 208 with a restraint system which does not injure out-of-position occupants is to incorporate controllability and intelligence into the restraint system. Such a system is termed herein an Intelligent Total Restraint System, or ITRS, and is designed to minimize occupant injury by reducing restraint forces to the lowest level necessary for a given occupant -- belted or unbelted -- in a given position subjected to a given crash, thereby extending the distance over which the occupant is restrained. A general ITRS is hypothesized to require sensors for occupant size, weight, and position, crash severity, and seat belt usage; and to require controllers for air bag inflation rate, seat belt load limit and pretensioning rate, D-ring position, and knee bolster force.

Various controllable inflators, crash sensors, and occupant sensors for implementing controllable air bag inflation systems of various levels of sophistication are described in the patent and technical literature **[9][10][11][12][13][14][15][16]**. Multi-stage -- especially two stage -- inflators have been developed for controlling air bag inflation rate, whereby inflation rate is controlled through the time delay between the activation of the separate inflator stages. This technology can also be applied to seat belt pretensioners and inflatable knee bolsters. The 1974 GM production air bag system is an early example of a controllable inflation system which incorporated a two-stage hybrid inflator which filled a two-chamber air bag -- one of the chambers acting as a knee bolster -- whereby the separate inflator stages were activated by a bumper mounted crush zone sensor and a pendulous passenger compartment sensor respectively, and whereby the time delay between the activation of the crush zone and passenger compartment sensors was inherently related to the crash severity through the mechanical compliance of the car structure **[17]**.

Hypothesis

The basis for the ITRS lies in the hypothesis grounded in experience that occupant injury can be mitigated by adapting air bag inflation rate and seat belt restraint forces to crash severity, and to occupant size, weight, position and seat belt usage. The purpose of the systematic characterization and control procedures outlined herein is to quantify the relationship between the system states and the corresponding optimal controls so as to optimize occupant protection over the expected ranges of occupants, occupant positions, and crashes. One outcome of these procedures is the determination of the relative importance of the various control inputs to harm reduction, thereby enabling an orderly, phased introduction of system elements in order of increasing cost/benefit ratio.

Given a suite of ITRS sensors having sufficient observability, and ITRS actuators having sufficient controllability, the performance of the system can be optimized by finding an associated system control algorithm which minimizes a given objective -- an overall injury assessment or harm value, or IAV -- subject to the given constraint that individual injury measures are less than accepted threshold levels. The resulting optimized system should be robust, providing near optimal performance over the expected operating range of occupants, occupant positions, and crash pulses for both belted and unbelted occupants.

Purpose

The purpose of this paper is to introduce the *TAKATA* Intelligent Total Restraint System, one variant thereof comprising a two stage inflator, crash sensor, seat belt sensors, and occupant weight and position sensor. The paper describes both a philosophy and a model-based methodology for developing the associated optimal controls. The objective function to be minimized is expressed as either a linear combination of associated injury measures, or as a cumulative harm function; individual injury or harm measures are used as constraint functions. The objective and constraint functions are derived as mathematical models using occupant simulation results for a range of occupants in a range of positions subject to a range of crash pulses and a range of inflation curves. The models are used to find both inflator firing threshold and the second stage time delay for a two stage inflator as a function of crash severity, occupant size/weight, occupant position and seat belt usage. Results are presented for driver side firing threshold crash severity as a function of occupant size and position, for both belted and unbelted occupants.

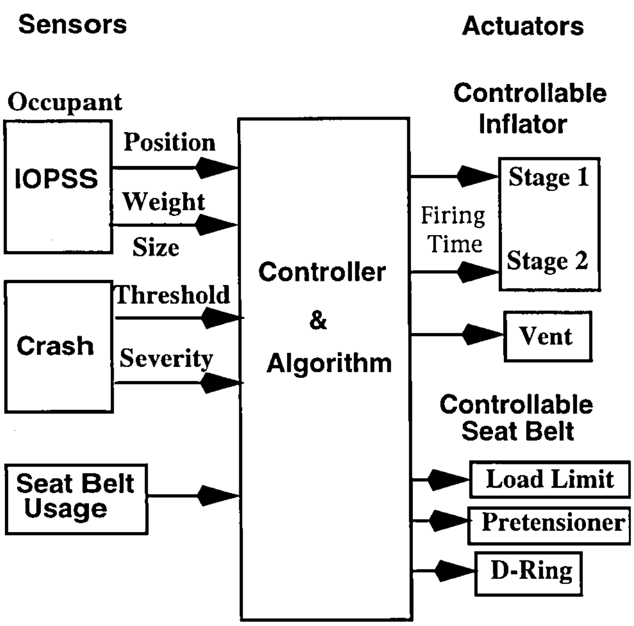
ITRS SYSTEM CONFIGURATION

The objective of the ITRS is to minimize injury to occupants from crash accidents by adjusting the operation of restraint actuators in response to the conditions of the crash, the occupant, and the occupant's seat belt usage. A general form of TAKATA ITRS is illustrated in **Fig. 1**, and includes all sensors and actuators which are expected to contribute to occupant harm reduction. The associated sensors and actuators are listed in **Table 1**, each of which has an associated level of significance towards overall harm reduction, the most significant of which can be incorporated into a reduced order system. One such reduced order system -- a two stage controllable air bag inflator based upon crash severity, occupant weight and position, and seat belt usage -- will be the focus of this paper.

Table 1: ITRS General System Architecture

Sensors (State Variables)	**Actuators** (Control Output Variables)	**System Parameters** (Model Properties)	**System Output** (Model Output)
Crash Severity	**Inflator**	**Interior**	**Head**
Occupant	Gas Generator	Geometry	HIC15ms
Size	1st Stage Firing Time	Material Properties	Accel3ms
Weight	2nd Stage Time Delay	**Air Bag**	**Neck**
Position	Module Vent	Design/Volume	Moment
H-Point	**Seat Belt**	Fold	Axial Load (T@L)
Torso Angle	Pretensioner	Chamber Configuration	Shear Load (T@L)
Seat Belt Usage	1st Stage Firing Time	Inflator Interface	**Chest**
	2nd Stage Time Delay	**Inflator**	Accel3ms
	Force Limit	Nozzle Configuration	VC
	D-Ring position	Gas generant partition	Compression
		(1st & 2nd stages)	**Femur Load**
		Seat Belt	
		Geometry	$IAV=\Sigma\ a_i * I_i / I_{threshold}$
		Webbing	or
		Seat	$IAV=\Sigma\ InjuryRisk_i$
		Knee Bolster	

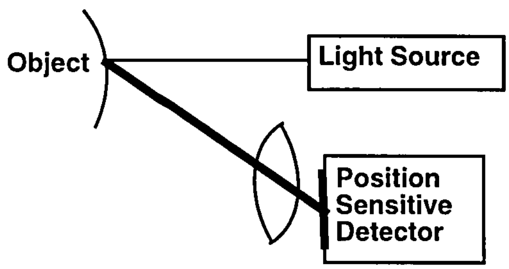
Figure 1
ITRS BLOCK DIAGRAM



Occupant Sensing

Occupant Position: Occupant position is sensed with an Intelligent Occupant Position Sensing System, or IOPSS, illustrated in **Fig. 2** comprising an infrared beam together with an off-axis position sensitive detector, or PSD. The sensed object diffusely reflects illumination beam back to the PSD, whereby the lateral position of the imaged spot on the PSD -- the sensed position -- depends upon the position of object's point of reflection along the illumination beam -- the occupant position. The sensed position signal is differentiated to measure occupant velocity, and both the velocity and distance measures are processed to calculate the likelihoods of presence of various objects, including a rear facing infant seat, a normally seated person, a person reading a newspaper, and an inanimate object. The IOPSS can be used to disable the inflator -- if there is no occupant, or if there is a rear facing child seat present -- in addition to providing information about the position and velocity of the occupant for purposes of continuous system control. Multiple beams may be utilized to sense torso angle or occupant profile.

Figure 2 IOPSS Operation



Occupant Weight: Occupant weight is sensed with a piezoresistive sensing element embedded into the seat cushion. The underlying sensing mechanism provides a continuous change of resistance with respect to weight, which can be exploited to provide an analog weight scale **[18]**.

The ITRS optimal controls are based upon results from crash simulations using models of conventional Anthropomorphic Test Devices (ATD) -- the 5th percentile female, 50th percentile male, and 95th percentile male -- for which size and weight are correlated. The relative importance of occupant size and weight is assessed with additional crash simulations using custom occupant models having similar weights but different sizes. If these variables have independent significance to the optimal controls, an occupant size sensor can be incorporated into the ITRS.

Seat Belt Usage: Seat belt usage is sensed with a switch mounted in the seat belt buckle, as is already incorporated into some production vehicles. Two sets of optimal controls are stored in the ITRS, one for belted operation, and the other for unbelted operation, which are selected in response to the state of seat belt usage.

Crash Sensing

Crash Severity: The purpose of a restraint system is to absorb the kinetic energy of the occupant relative to the decelerating vehicle in which the occupant is riding. This relative kinetic energy grows with time through the duration of the crash event until restraint forces are applied to the occupant by either the seat belt or air bag. Crash severity is a measure of the kinetic energy which must be absorbed by the restraint system, and is related to the injury producing potential of a crash. This measure depends upon both the nature of the acceleration field to which the occupant is exposed, as well as the size and position of the occupant which governs the occupant's kinematics within the passenger compartment as the vehicle is decelerated. The inflator firing threshold is defined as the crash severity level above which the air bag is fired, and depends upon occupant size, position, and seat belt usage. For a system without an occupant sensor, the crash severity measure is referenced to a normally seated occupant, and the firing threshold crash severity level depends strictly upon seat belt usage.

Crash Threshold: While the level of crash severity may be a sufficient measure of whether to fire the air bag inflator, in general, a crash threshold -- defined in terms of crash pulse parameters --enables crash events to be distinguished from acceleration producing non-crash events, such as rough roads, vandalism, or impacts with small objects. The air bag is armed for firing above the crash threshold, and is fired as soon as possible once the crash severity measure exceeds the firing threshold given as a function of occupant position, weight, and seat belt usage.

Air Bag Inflator

The air bag inflator module comprises a single plenum housing containing a two-stage Envirosure™ gas generator, to which is attached a single chamber, silicon coated, vented air bag. The Envirosure™ gas generator is advantageous because of low toxicity, small size and relative insensitivity to temperature variations.

Figures 3 and **4** illustrate the passenger and driver side 2-stage inflators respectively. These inflators are partitioned into the two stages according to the philosophy and methodology outlined below, which provides that the worst case out-of-position occupant (OOPO) is not injured when firing only the first stage inflator. The composite mass flow inflation rate of the gas generator is controlled by varying the time delay -- termed the second stage time delay, or SSTD -- between the first and second stage inflators, the first stage generally always being fired first for best protection to out of position occupants and extremities.

An alternative 2-stage passenger side inflator module utilizes two conventional Envirosure™ gas generators.

Figure 3

TAKATA 2-Stage Passenger Side Inflator

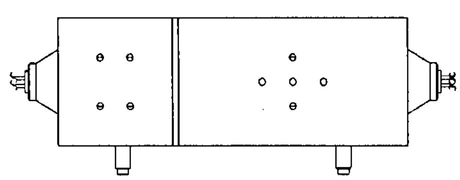
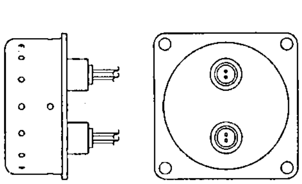


Figure 4

***TAKATA* 2-Stage Driver Side Inflator**



Seat Belt System

The ITRS system described herein incorporates a conventional mechanical seat belt system having a retractor located in the B-pillar, a force limiting energy absorber incorporated in the retractor, and a fixed location D-ring. The modeling and control methodologies described for the air bag system also apply to the problem of finding optimal controls for the controllable seat belt system illustrated in **Fig. 1**, which incorporates a two stage pyrotechnic pretensioner, a controllable force limiting energy absorber in series with the seat belt, and a D-ring positioner.

SYSTEM PHILOSOPHY

Objective

The specific objective of the ITRS is to minimize frontal crash induced injury to occupants, to avoid exceeding established injury thresholds for specific body regions, and to not cause more injury from the deployment of the restraint system than would be sustained by the occupant without the restraint system.

The optimization of ITRS controls depends upon empirical models of system responses for given states and controls. The models inherently account for the influence of the associated system architecture and system parameters. Since a considerable amount of time and effort is required to gather the data from which these models are derived, a well-founded system philosophy guiding the specification of the system architecture and parameters is an important prerequisite to obtaining effective and practical results in a reasonable amount of time.

Optimization of System Controls using Mathematical Models of Simulated System Performance

The ITRS controls are optimized using occupant injury response models which are formed from an extensive array of simulated responses to a range of crash conditions, occupants, occupant positions, and inflation rates (SSTD) which covers the expected operating range of the system. The data from which the models are derived is collected from occupant simulations using either PAMCRASH or MADYMO occupant simulation models. The data could also be gathered from Hyge sled tests, but at considerably greater expense. Occupant simulation models also have the inherent advantage of repeatability. Notwithstanding that the degree of correlation with either sled or actual crash tests is sometimes limited, occupant simulation models are still useful in predicting trends, and for showing comparative results. Models for injury measures and overall the injury assessment or harm value (IAV) are derived from the modeling database using least squares or other statistical modeling techniques.

A first subset of the independent variables of these models comprise state variables which correspond to the sensed ITRS systems inputs listed in **Table 1**. The second set of independent variables correspond to the controls. Two sets of models are formulated, one for belted occupants, the other for unbelted occupants. The modeling database is derived from a full factorial design of experiment with at least 3 conditions for each independent variable to account for non-linear effects.

Two classes of optimal controls are found, firing threshold and inflation rate control (via SSTD). Optimal controls are derived from the models by minimizing an objective function with respect to the control variables subject to a set of constraint functions, whereby the formulation of the objective and constraint functions depends upon the class of optimal controls. The optimal controls are solved for an array of state variable values, and the results are then formulated into an implementable optimal control law. The implementable inflator controls may be further modified to account for the effects of temperature on the inflator module.

One advantage of this approach is that the system models enable a limited amount of data to be interpolated to describe the entire operating space. A further advantage is that the models both filter noise inherent to the modeling database, and provide the necessary response trends. A yet further advantage is that additional data can be readily incorporated into the models to account for regions of high sensitivity or abrupt behavior.

System Performance Measures

Occupant restraint performance is typically measured by dynamic and kinematic causative factors associated with the various injury mechanisms listed in **Table 1 [19]**.

Each injury measure has an associated injury threshold level for a given sized occupant above which a person would have an associated probability (20%) of having a serious injury. The measured injury levels can be normalized with respect to these injury measures, and then collected in a weighted linear combination to form an objective function, with the weighting factors representing the distributions of serious injury by body region for a given class of restraints, e.g. no restraint, seat belt, air bag, seat belt + air bag. Within a given body region, the corresponding weighting factor can be applied to either the average of the normalized threshold levels, or -- in the preferred mode -- the maximum normalized threshold level.

Each injury measure also has an associated injury risk assessment based upon a Logist probability density function representing the likelihood of a serious injury as a function of the associated injury level. The overall likelihood of a serious injury is then given by the unweighted sum of the individual injury risk assessment values, and this can be used as an alternative objective function.

Because of the limitations of occupant simulations which limit absolute accuracy, a third possible formulation for a system objective function is given by the relative change in one of the above IAV levels between a controlled and non-controlled state of the system, thereby driving the optimization to find the controls giving the greatest improvement in performance relative to a conventional air bag system.

Motion criteria components such as restraint quotient, rebound velocity, torso angle, and pelvic restraint as suggested by Viano and Arepally **[19]** are also incorporated into the overall objective function.

Firing Threshold

The inflator firing threshold is defined as a crash severity level which if exceeded causes the first stage gas generator to be fired. This crash severity level generally corresponds to the condition of either the occupant's head impact velocity exceeding 19.3 KPH(12 MPH) **[20]** or the shoulder belt tension exceeding 3 KN **[6]**, both subject to the constraint of not exceeding established injury thresholds.

The inflator firing threshold is found by first modeling both head impact velocity and shoulder belt load each as a function of crash severity, occupant size/weight, occupant position, and seat belt usage; and then solving to find the maximum crash severity for which the both the head impact velocity and shoulder belt loads do not exceed their respective limits. These solutions are found for a range of occupants and occupant positions, and from them the resulting firing threshold crash severity is expressed as a function of occupant size/weight, occupant position, and seat belt usage.

For a system without an occupant sensor, two different firing thresholds values are defined, one for unbelted occupants, the other for belted occupants. Each respective value is either set for the normally seated 50th percentile male, or for the minimum crash severity for the range of occupants and occupant positions.

Firing Time

The ITRS fires the air bag inflator as soon as the estimated crash severity level exceeds the firing threshold. By firing as soon as possible, the air bag inflation rate can be minimized to minimize restraint forces and thereby minimize the risk of injury to the OOPO or to occupant extremities **[21]**.

Inflation Rate Control

The inflation rate of the 2-stage inflator is controlled by varying the time delay between the firing of the first and second inflator stages, i.e. the second stage time delay (SSTD). The SSTD is adjusted to satisfy the ITRS objectives of minimizing occupant injury or harm subject to the constraint the no individual injury measure exceed its accepted injury threshold limit.

The optimal inflation rate is found by first modeling the objective -- either injury assessment value or injury risk -- and individual injury measures each as a function of crash severity, occupant size/weight, occupant position, and SSTD. Separate models are formed for belted and unbelted conditions. The optimal SSTD is then found by minimizing the objective function with respect to SSTD subject to the constraint that none of the individual injury measures exceed their corresponding injury thresholds. These solutions are found for a range of occupants and occupant positions, and from them the resulting optimal SSTD is expressed as a function of crash severity, occupant size/weight, occupant position, and seat belt usage.

For a system without an occupant sensor, the SSTD function is found by minimizing the objective for the normally seated 50th percentile male ATD subject to the constraint that the individual injury measures are not exceeded for all occupants over the range of seating positions. These solutions are found for a range of crash severities, and combined to express the optimal SSTD as a function of crash severity and seat belt usage.

APPROACH

The optimal ITRS controls are determined through a systematic characterization and control procedure. Given a specification of the system architecture, preliminary experiments are conducted to establish the system parameters. Total propellant load and air bag vent area are found for FMVSS-208 conditions. The partitioning of the propellant load is found from static deployment tests under both cold conditions without an occupant and under hot conditions with occupants placed in worst case OOPO positions. A simulation experiment is then designed and conducted, and the results are modeled with respect to state and control variables, and the controls are then optimized and implemented. Details of various aspects of this approach follow:

Simulation Experiment

The system responses to the conditions of the experiment are measured with an occupant simulation model, either PAMCRASH or MADYMO. ATD models with deformable chests are utilized for conditions with either air bag or seat belt restraint. The simulations are subject to several necessary, simplifying assumptions, including the elimination of the module cover and establishing a minimum distance of about 100 mm between the OOPO and the inflator module.

A post processor has been developed to process and manage the large volume of data generated from the simulation experiment, and to prepare a modeling database for purposes of system characterization. The individual injury and motion measures are calculated for each simulation condition. The IAV is formed as a linear combination of these measures using associated weighting factors representing the likelihood of serious injury by body region. An overall injury risk value is also calculated by summing the individual injury risks **[19]**.

System Characterization and Control

The modeling and optimization procedures for finding optimal controls for firing threshold and SSTD have been described above. The resulting optimal inflator controls are implemented in either functional or tabular form, may be further modified to compensate for the effects of ambient temperature on either the gas generant burn rate and or the deployment characteristics of the inflator module cover and air bag.

Validation

The resulting implementable optimal controls are compared with a conventional inflation system using occupant simulations. Configurations of interest showing the greatest levels of improvement are then selected and compared in Hyge sled tests.

RESULTS

Fig. 5 illustrates the range of driver-side occupants and initial occupant positions that were used in a PAMCRASH crash simulation experiment conducted for purposes of determining the driver-side inflator firing threshold. These occupants include: 1) the 5th percentile female (05F), 2) the 50th percentile male (50M) and 3) the 95th percentile male. The 95th percentile used in these simulations was approximately 10 cm shorter that the corresponding ATD due to a defect in the dummy database.

For the unbelted conditions, each occupant was initially placed in its normal H-point seating position with three different torso angles: 1) normal seating, 2) vertical (shoulder joint relative to the H-point), and 3) forward leaning. Additionally, the 50M occupant was positioned in the 05F and 95M H-point locations for both the vertical and forward leaning postures. The seating positions are identified by the lower case letters printed beneath each of the associated illustrations in **Fig. 5**.

The belted conditions were all simulated with the occupants in their respective normal seating position. The seat belt system incorporated a simulated 2900 N force limiting energy absorber located in the vicinity of the seat belt retractor.

The purpose of these simulations was to determine the highest crash severity level satisfying the following conditions: 1) occupant head impact velocity less than or equal to 19.3 KPH (12 MPH) **[20]**, 2) shoulder belt load less than or equal to 3000 N **[6]**, and 3) occupant injury measures less than or equal to their respective threshold levels. This was accomplished by performing a series of simulations for each initial seating position and posture using crash pulses of three to four different severity levels. The crash severity level is indicated herein by the corresponding maximum integrated velocity of the acceleration crash pulse.

The results of the simulations are presented in **Figures 6-8** for the controlling injury measures, overall injury risk, and injury assessment values respectively, each plotted as a function of torso rotation and crash severity, and grouped according to occupant size, H-Point position, and seat belt usage. The belted configurations are only plotted as a function of crash severity since the torso angle was limited to the normal seating position. Specific occupant positions are indicated on these plots with the corresponding alphabetic codes from **Fig. 5**.

The injury measures pertinent to firing threshold control are plotted in **Fig. 6**. Contours of head contact velocity are plotted as a function of torso angle and maximum crash velocity for the unrestrained cases. The 20 KPH firing threshold limit contour is highlighted with a bold solid line. The bold dashed lines superimposed on some of the plots represent specific injury threshold limit conditions. Maximum shoulder belt force is plotted as a function of maximum crash velocity for the belted cases. Simulation conditions for which an injury or the velocity threshold is exceeded are highlighted with a "square" symbol around the corresponding position code.

The injury risk functions plotted in **Fig. 7** are given by the sum of the individual injury risks based upon the Logist probability function and associated parameters given by Viano and Arepally **[19]**. Facial injuries are not included in this sum. The parameters (α and β) for those injury measures for which biomechanically based parameters have not been given were estimated based upon mathematical similarity to the parameters which have been specified.

The weighting factors for the IAV plotted in **Fig. 8** were estimated from the associated distribution of injuries by body region for unrestrained occupants as referenced by Viano and Arepally [19]. The weighing factors are applied to the worst normalized injury measure for the associated body regions.

DISCUSSION OF RESULTS

One of the objectives of this study was to determine the extent to which inflator firing threshold depends upon occupant position, occupant size/weight, or seat belt usage.

For the unbelted cases, the head impact velocity is the principal determinant of crash firing threshold. However, this measure is not sufficient to determine the threshold, as illustrated in **Figs. 6a** and **6e**. In **Fig. 6a**, for example, neck flexion torque appears to be a limiting factor for the forward leaning 05F occupant.

For all of the occupants in their normal seating positions -- **Figs. 6a, 6c**, and **6e** -- the head impact velocity threshold increases with increasing torso angle beyond the vertical position, and is generally a minimum near the vertical position. The sensitivity near the vertical position appears to be due to the kinematics of the occupant, and specifically the relative times at which the lower and upper portions of the body make contact with the vehicle interior. As noted previously though, the neck flexion or compression constraints become active for large initial torso angles. The firing threshold crash severity appears to increase with decreasing occupant size.

The overall injury risk measure in **Fig. 7** is relatively flat with respect to torso angle for the 05F and 95M drivers, but increases with decreasing torso angle for the 50M driver. This measure however does not account for facial load which appears to be the limiting factor for low torso angles.

For the 50M driver, comparing **Figs. 6b-d**, the velocity threshold is generally greatest for the normal (mid) H-Point position, and decreases for positions both fore and aft of this point.

For the belted cases illustrated in **Fig. 6f**, the shoulder belt load limit is reached at approximately the same crash velocity for each of the three different occupants. As seen in **Fig. 5**, the relative location of the D-Ring to the occupant is most effective for the 05F occupant, and this effectiveness decreases with increasing occupant size. High neck forward shear loads in the 05F driver likely result from the early engagement of the shoulder belt because of the D-Ring geometry. The seat belt energy absorber did not function except at the most severe (53 KPH) crash pulses in the simulation because of friction between the shoulder belt and the D-Ring. The shoulder belt load threshold was assumed to be 3000 N for this study. If a higher load limit were chosen, then a higher crash severity level could be chosen for the 50M and 95M occupants. None of the occupants contacted the vehicle interior except for the 53 KPH crash pulse.

SUMMARY AND CONCLUSIONS

A systematic procedure and associated philosophy has been developed for optimizing the controls in an Intelligent Total Restraint System. Occupant simulations are used to gather system responses to an array of control inputs, and the response data is mathematically modeled. While simulations are useful tools in understanding trends, the results must be confirmed with sled or crash tests. The occupant kinematics throughout the crash have a significant effect on the results, and this influence is complicated by the inherent non-linearities of the system as well as the difficulty in robustly simulating contacts between system elements.

The results presented in this paper represent a relatively small part of the overall control optimization process. However, they offer encouragement to the possibilities of improved occupant safety with an ITRS. Inflator firing threshold appears to depend upon occupant size, weight, and position. Each of these elements would be sensed in an ITRS, and can be used to reduce risk of injury to the occupant by the air bag by improving the ability of the system to more precisely determine when the air bag is needed. An ITRS having control of the seat belt load limit would further enhance safety for belted occupants in moderate severity crashes.

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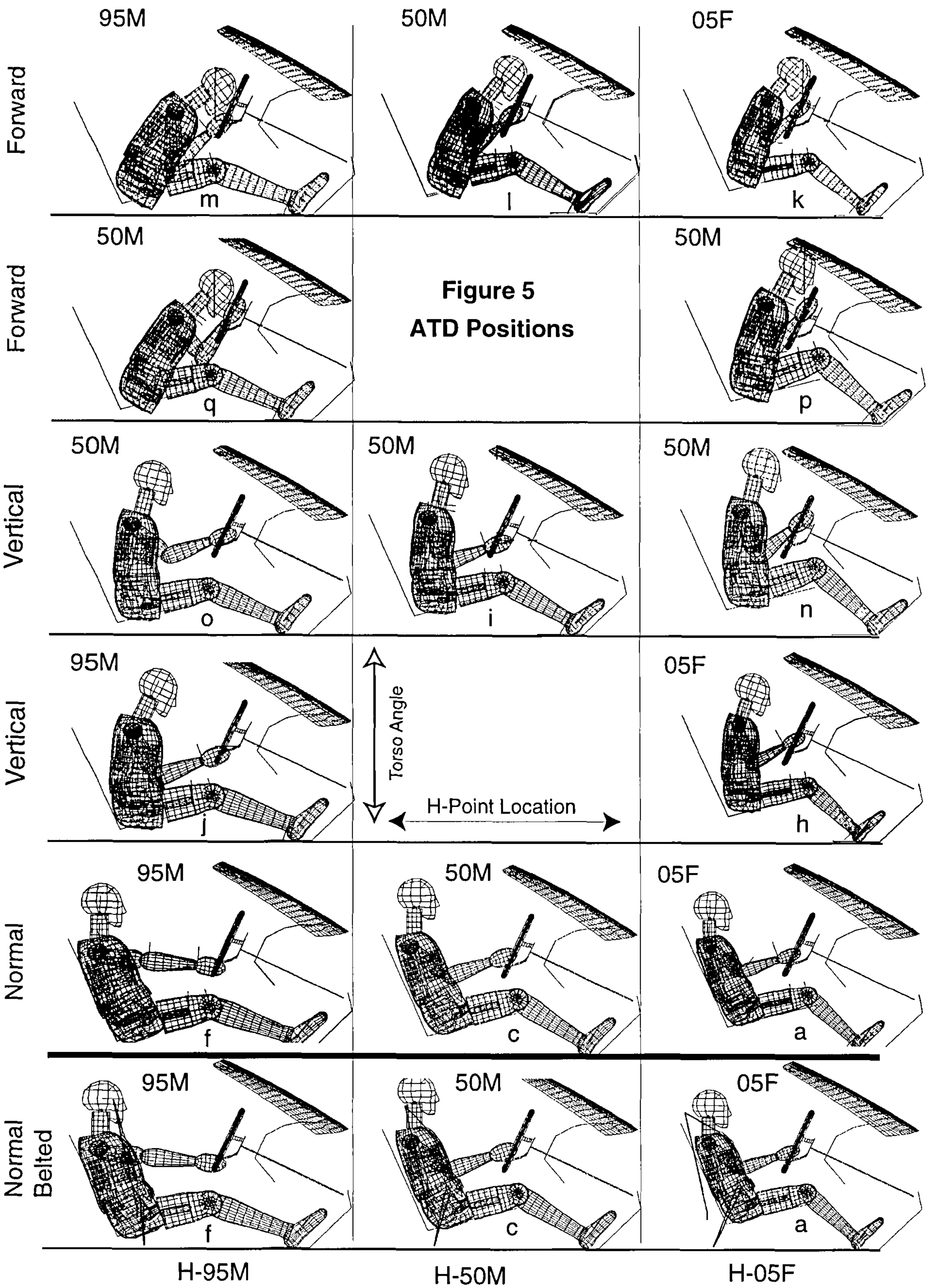


Figure 5
ATD Positions
95M
50M
05F
Forward
m
l
k
q
p
Vertical
o
i
n
j
h
Torso Angle
H-Point Location
Normal
f
c
a
Normal Belted
H-95M
H-50M
H-05F

Figure 6: Occupant Injury vs. Torso Rotation & Crash Severity

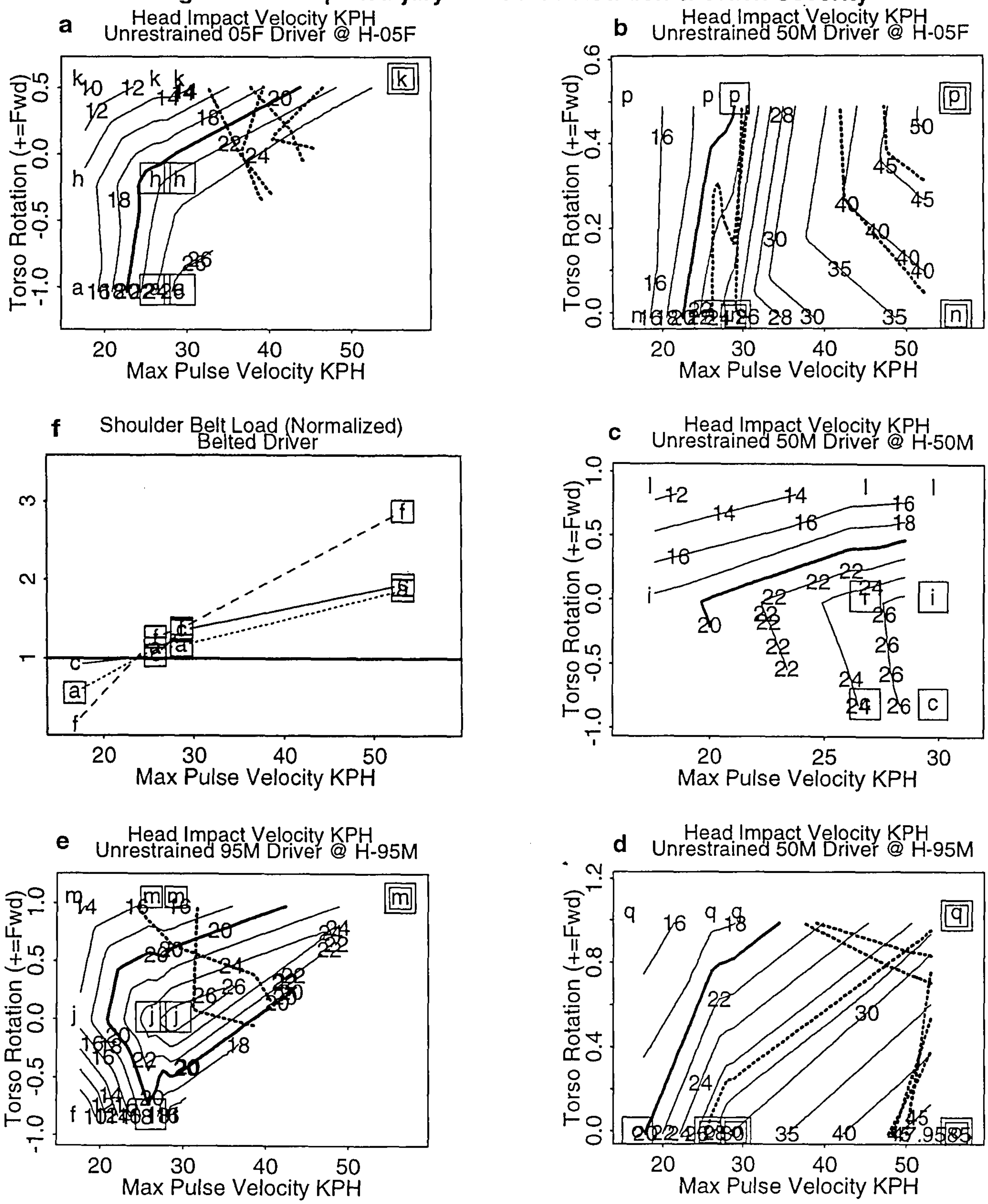


Figure 7: Occupant Injury Risk vs. Torso Rotation & Crash Severity

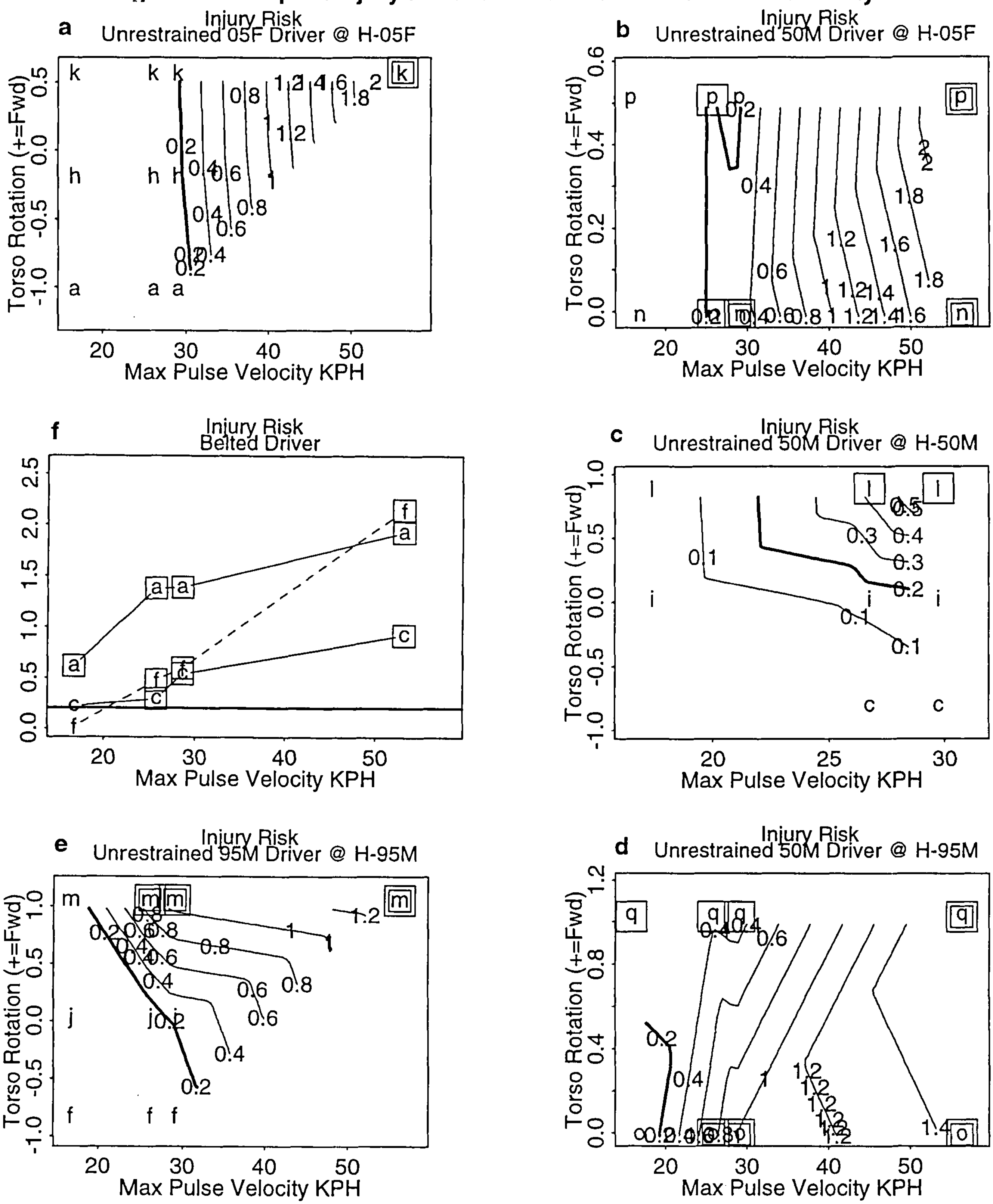
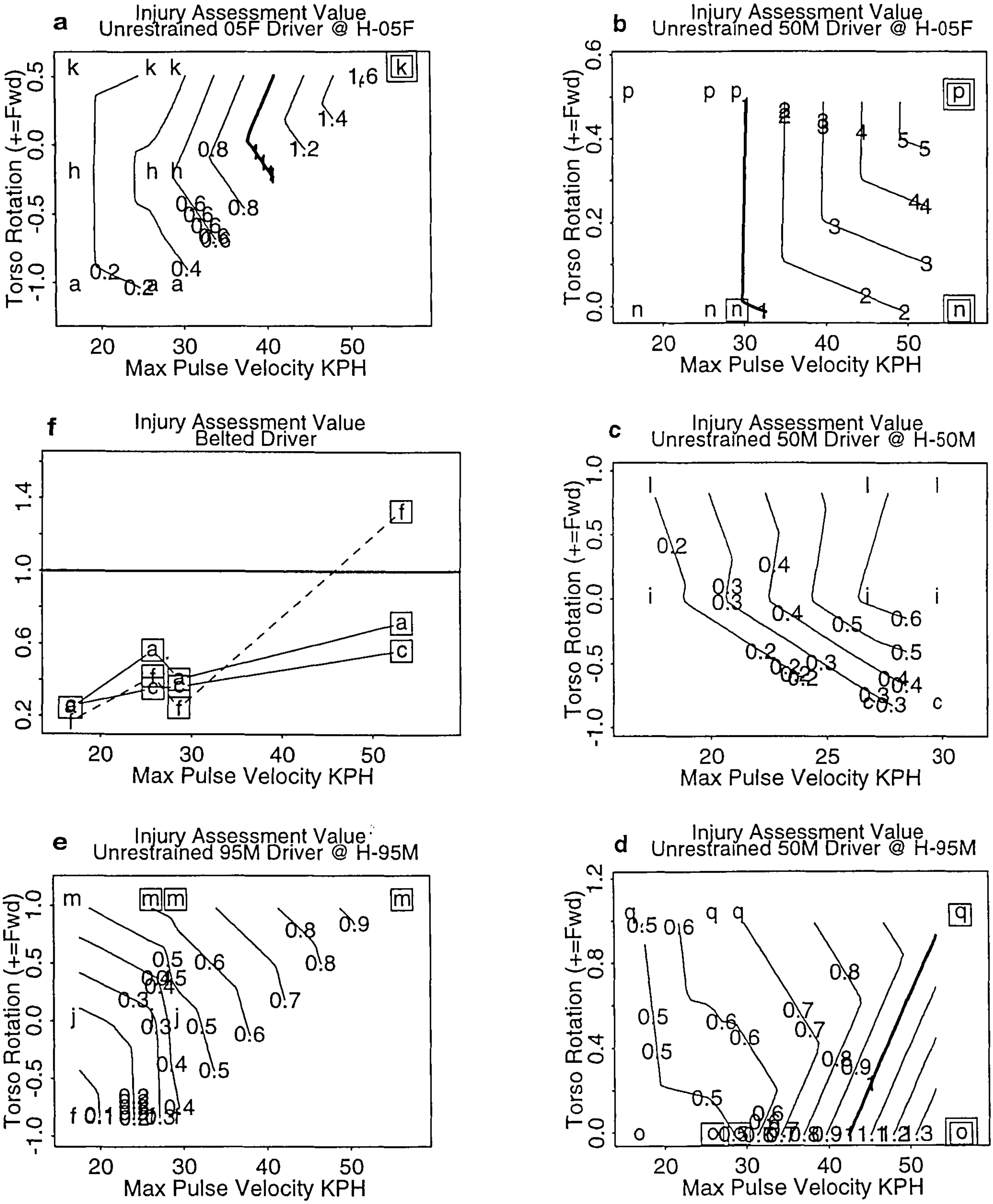


Figure 8: Occupant Injury Assessment Value vs. Torso Rotation & Crash Severity



VSR PROGRAM - VEHICLE AND SAFETY ON ROAD - STATE OF THE ART AT MIDWAY

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Paper Number 96-S3-O-15

ABSTRACT

The Research Program on road safety proposed for the 1993 to 1998 period aim to combine the most appropriate approaches under the following aspects :

- Avoid accidents and reduce frequency and importance of injuries (primary and secondary safety);
- Combine measures of improvement on vehicles and road;
- Mix technical and educational measures.

VSR Program has been allowed by the European Commission on June 2, 1993 for a total amount of research reaching 695 millions of FF. At the present time, we are near of the program midway and this paper gives a first range of conclusions.

A first step of 235 millions of Francs has already been engaged and 175 more are on the way corresponding to more than 50 % of the research.

Involving driver, vehicle and road system, VSR is organized round four main axis:

- Increase the knowledge on accident causes and analyze in depth their consequences;
- Prevent situations leading to accident;
- Avoid that these situation be transformed in an accident;
- Protect the occupant when the impact is ineluctable.

On the first item, INRETS, PEUGEOT and RENAULT have set up 4 teams of accidentologists working on one's ground simultaneously with police and rescue. Moreover, police and medical reports of 9 000 fatalities have been analyzed and a first evaluation of primary safety counter measures effectiveness has been established.

In the frame of the second approach, a Method by Gradual Experiment - said MAP : « Méthode par Apprentissage Progressif » - has been developed. It can detect changes in driver behavior due to drowsiness. Other changes in behavior can also be detected as those caused by :

- internal state of the driver : fatigue, exasperation, ...
- other actions than driving activity : drink, eat, smoke ...
- driving environment : loudness of traffic, road work, intersection, ...

Car fittings industry has initiated low cost component development, for mass production, able to be used in driving assistance operations (i.e. radar for AICC).

For occupant protection, detailed studies of material laws and breaking mechanisms have led to improved performances in computer modeling. In some cases, improvements over 40 % are obtained for frontal and side impact.

Research concerning trucks and buses and analysis of road safety sociological and economical aspects are also integrated in VSR

INTRODUCTION

Thanks to the freedom that it brings, automobile impress our age and our way of life. But his development and the way we use it give a challenge to the human community. One of these major challenge is the improvement of road safety. This worry is not new and, up to now, considerable progress has been obtained. In France, from 1970 to 1994, the number of fatalities was reduced by near 50 % as in the same time the number of car was doubled. This reduction is the result of regulations, road and vehicle improvement and transfer between several type of transportation. From 1985, the progress is slower despite number of campaign for enhance motivation of the drivers and the succession of actions.

One explanation is the fact that elementary rule of safety are not respected : not adapted speeds, insufficient rate of wearing for safety belts, alcoholism when driving. We must add that the control and sanction system is not able to reduce significantly the violations of safety rules. From an other point of view, some kinds of behavior which are not definite violations of these rules contribute to make the balance heavier : drowsiness, bad reaction in critical situation, wrong appreciation of the real capabilities of the vehicles in relation with the road network.

At the present time, on basis of accident statistics, it is well known that, in France, if the seat belts were worn systematically at 100 %, at the front and rear seats, by adults and children using convenient restraint systems 29 % of death in passenger cars might be avoided.

More over, in 51 % of the fatal collisions, the violence of the impact is at such a high level that we cannot afford to save the occupants otherwise than avoiding the accident to occur.

The remaining 20 % of fatalities is then the maximum that the passive safety can improve, even we bring the protection quality of cars up to an unrealistic level.

To sum up, the French Research Program entitled « Vehicle and Safety on Road » V S R, proposed for the period running from 1993 to 1998 want to be realistic and efficient. It aims to obtain effective results on the basis of convenient approach combining :

- accident avoidance and reduction of injury frequency and severity;
- cooperative countermeasures between vehicle and road;
- enhancement of educational dispositions (information and education of the driver).

V.S.R. - AN INNOVATIVE GREAT RESEARCH PROGRAM FOR 1993-1998 PERIOD

In front of the slowing down of safety progress during several years, a new approach appeared to be necessary associated to ambitious target.

The aim of Vehicle and Safety on Road French program - VSR - is to contribute to a significant reduction of number and gravity of accidents technical and administrative dispositions leading to a quieter way of driving, without reduction of the quality use of the vehicle.

This plan is the issue of an approach that is comprehensive, rigorous, innovative, determined and coordinated.

- Comprehensive, because it no longer focuses separately on the three components of road safety (driver, vehicle and road system) but views these as an integrated system whose various elements can and must interact.
- Rigorous, in that it affords the most cost-effective solutions based on a scientific analysis of the various types of accidents, their frequency and the conditions in which they occur, and on an assessment of the devices that could be used to avoid them.
- Innovative, since it draws upon the vast potential of the emerging electronic and telecommunications technology.
- Determined and coordinated, because the proposed solutions involve many different partners and must necessarily be European in scope.

This national French Program , elaborated in the frame of PREDIT, is made of four main panels called KNOWLEDGE, PREVENTION, AVOIDANCE, PROTECTION :

- KNOWLEDGE to understand better accident causes and consequences,
- PREVENTION to avoid critical driving situations to occur,
- AVOIDANCE to act such a way that a critical driving situation don't run into an accident,
- PROTECTION to reduce at the maximum level the severity of the accident consequences for occupants.

The program as a whole is built upon a strong cooperation between car manufacturers, suppliers of the automotive industry, managers of road network and scientific laboratories and especially INRETS.

VSR has been notified by the European Commission on June 3, 1993 for a total research amount of 695 million of Francs as detailed in Table 1.

Table 1
Financial framework and state of expenses

	Total amount (MF)	Notified (MF)	To be allocated (MF)
VSR PROGRAMM	**695**	**272**	**423**
KNOWLEDGE	80	55	25
PASSENGER CARS	410	205	205
Prevention	57	22,5	34,5
Avoidance	73		73
Protection	80	54,5	25,5
Demonstrators	200	128	72
BUSES AND TRUCKS	140		140
SOCIO-ECONOMY	45	11	34
SIMULATORS & TEACHING	20	1	19

The most important part of the program (59 %) is devoted to passenger cars, followed by buses and trucks (20 %). Thanks to the growing importance of human factors in the road accidents, a significant part of the program applies in the kingdom of socioeconomy and research on accident causation.

In the middle of 1996, near 40 % of the expenses has been engaged and has led to a first range of result which will be presented in the suite of this paper.

KNOWLEDGE

The most adapted solutions to the given target have to be selected after a scientific approach able to evaluate the perspectives of earnings and to calculate the cost in order to obtain the best cost benefit ratio.

For this evaluation, it is necessary to acquire knowledge and data on accident causation and to study in depth all the implications. These actions have to be held in the field of socioeconomic and accidentology

Socioeconomic

A Commission gathering representatives of the French Departments of Transportation, Research and Industry, together with INRETS and Car Manufacturers has selected the following research subjects for which a call for research proposal has been circulated.

Under the coordination of CEESAR [1], six main themes shall be operated.

The research on the handicap and the future of road casualties shall contribute to the evaluation of the medical after-effects gravity and propose a methodology for economical road morbidity evaluation.

The identification and the measure of the risk and the models aiming to describe and explain the road insecurity will make more accurate the risk calculation thanks to a better knowledge of the mobility. The program takes into account the internal risk (vehicle occupant) and the external risk (pedestrians, 2-wheelers, occupants of collided vehicle) and will introduce comparisons with geographic and international parameters.

The development of new technologies, especially electronics in driving assistance, shall introduce stronger interactions between car manufacturers, road project managers, traffic operators ...Research integrating driving situation characteristics, drivers psychology, consumer aspirations, needs and limits will help to understand better what can promote or slow the acceptability of new technologies.

Everywhere in the world, the risk incurred by the young drivers is much more higher than the average risk. This fact underlines the necessity to improve the effectiveness of safety educational methods as well for initial training of the drivers than the maintaining of an up to date level of knowledge during all the driving activity. The use of training road simulators will be possibly studied. Socioeconomic research on the functioning and financing conditions of initial and permanent training systems will be encouraged.

In many fields of the social life (health, people and good protection, work...), safety become a more and more important value. It must be balanced with other criteria as comfort, pleasure and cost. The research aims to analyze the place taken by safety in the value systems of road users, understand the psychological mechanisms and explain their inconsistencies with the behavior.

At last, the sixth theme propose to study what can be the contribution of institutional management organisms in the road safety performances. Research in sociology of the institutions must analyze role, ambition, means, implication levels of various institutions including concrete conditions of functioning and identification of braking factors.

CEESAR received approximately 25 answers on the six above themes which can be split between definition contracts allowed to little groups of researchers and research contracts to be subscribed by pluridisciplinary teams with a well-known scientific environment.

Accidentology

Measures proposed by the French laboratories of accidentology as LAB Peugeot Citroën/Renault, INRETS, CEESAR have widely participated to the progress that has been realized during last 15 years as well for use of restraint systems (obligation to bear safety belt at front and rear seats, best conditions of seat belt use, ...), than for the optimal deformation mode for vehicle structure.

For more than five years, it is now demonstrated that more than one half of fatalities cannot be saved other than by avoiding accident. Consequently, VSR program has oriented research toward accident causation with three main themes : detailed study of accident on the ground, exploitation of fatal accident statements, on-board electronic recorders.

Detailed studies of accidents on the ground

These studies are made by pluridisciplinary teams who work in real time on the accident location. Accidentologists are alerted by the rescue immediately after the collision occur and they operate in the same time. Each team is composed by 3 specialists :

• one expert of the vehicle who store 230 parameters relative to active and passive safety;

• one expert of the road configuration at the accident time who store 130 parameters relative to the shape and the surface characteristics of the road environment

[1] Centre Européen d'Etudes de Sécurité et d'Analyse des Risques

including meteorological conditions. He is also in charge for the mathematical accident reconstruction.

• one psychologist who store 135 parameters on the conditions of the pre-collision phase as felt by the drivers.

Typically one docket for 2 cars on 2 crossing roads contains usually more than 1000 informations.

Presently, 4 team have been installed. Two of them by INRETS work at Lyon and Salon-de-Provence near Marseille. The 2 other one have been installed by CEESAR approximately 100 km West and North of PARIS in the frame of a contract with the LAB Peugeot Citroën/Renault.

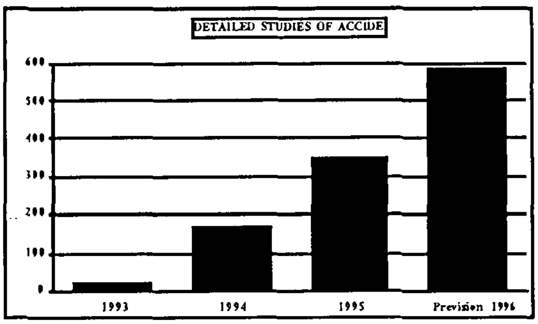

Figure 1 Evolution of cases of detailed accident from 1993 to 1996 (prevision).

The number of cases increases regularly. 300 cases were available at the end of 1995 and the total will reach 600 in 1996. Some improvements of the method are going on especially by a better knowledge of the road characteristics through a specially fitted car with image numerical analysis equipment, thanks to the work in connection with other European teams, the use of research road simulators and taking into account the new technologies able to improve the stability control of the vehicle.

The information obtained to day from the data analysis must to be used carefully due to the low level of statistic significance, the real scientific complexity of this domain and the subjective expectations that can lead to wrong estimation of the responsibility share between car, road and driver. Nevertheless, one can give some tendencies as the poor effectiveness of ABS, the growing importance of road parameter in accident causation and the low effectiveness of driving training, before the licence, of younger escorted by an adult.

Fatal Accident Statements

The raw material of this study is made by all the policeman reports for fatal accidents in France during the year 1990. The total number of these accident reports reaches 8 419 shared between passenger vehicles (3 926), trucks and multipurpose vehicles (1 291), pedestrian and two-wheeled vehicles (3 203).

The coding of each case must be done very carefully: for each parameter the operator takes care to make an action of inquiry and analysis in order to be sure to affect a convenient value to the variable.

At the beginning of February 1996 almost all the statements have been coded (87%) and particularly in the case of the passenger cars (95%).

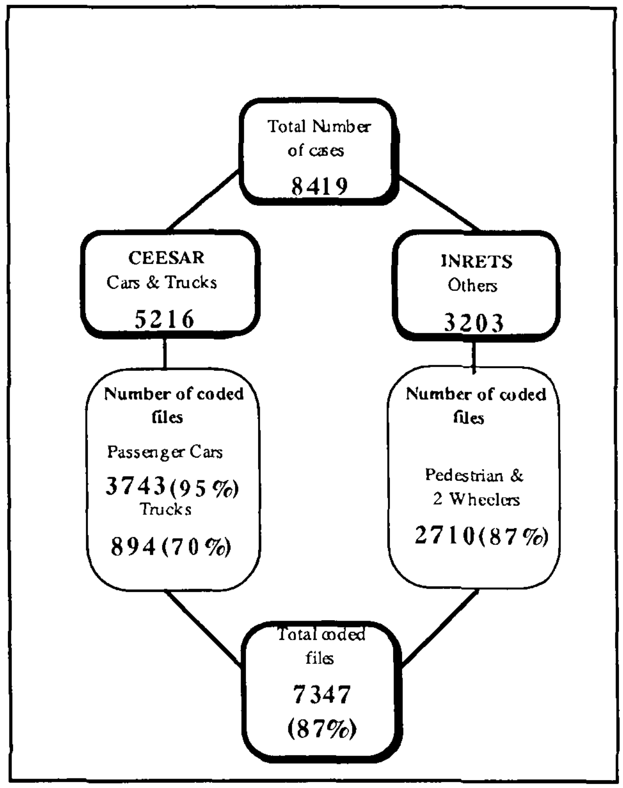

Figure 2 Number of statement analyzed at the beginning of 1996.

For an accident with a single car, each file contain 155 variables shared between general information (16), accident location and road characteristics (14), vehicle (81) and car users (44). If 2 cars are involved the variable number grows up to 280.

The first results are cheerful and we can quote sevaral thematic studies results coming from this data base.

The lane keeping system developed by PROMETHEUS offers a certain capability in any case where lane demarcation is clear ; the number of fatalities would be reduced by 45%.

In 30% of cases, the fatal accidents have been cause by a driver under alcoholic influence.

The young drivers having the driving licence for less than one year represent 10% of drivers responsible of fatal accidents.

The risk of accident is twice when the weather is wet.

Knowledge of human being

The main actions aiming to understand better the behavior of human being have been oriented to physiology, biomechanic and mathematical modeling.

For physiological purpose, ECIA has developed in connection with the LPPE Strasbourg (Pr MUZET) a fundamental study about the effect of hygrometrical and thermic conditions on comfort and driving attention.

INRETS and the « Laboratoire de Biomecanique Appliquée » at MARSEILLE University have performed an anatomical study leading to 3D reconstruction of human in seated position. With this model it is possible to show the interrelation between the seat belt and the internal organs.

Human body modeling

Several actions are on the way for a better numerical representation of dummies and human being. In fact, we need to have at our disposal two kind of tools. On one hand, it is necessary to have an accurate and reliable representation of the mechanical dummies used commonly during crash tests devoted to check the compliance of cars with safety regulations and with the internal specifications of car manufacturers in the area where regulations does not exist. On the other hand, an human model with a high biomechanic representativity have to be developed with the purpose to understand better the mechanism of injuries and increase the real safety of the vehicles.

A first action is engaged outside of VSR program through a BRITE contract gathering PSA RENAULT VOLVO and FIAT in order to build numerical models of Hybrid III Frontal and Eurosid Lateral dummies. This action involve sophisticated experiments on foam and rubber material laws.

VSR paid his attention principally on human being models. The project is composed of three steps :

• in the first step, obtain a reliable 3D representation valuable for displacements and forces in cinematic and dynamic behavior in order to understand better injury mechanism; this implies that medical personnel in connection with the project experiment material law from the human;

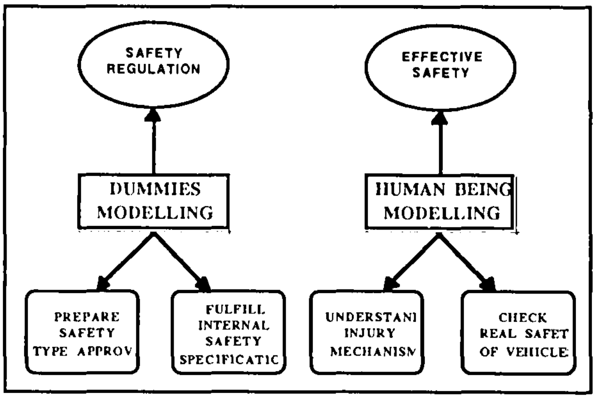

Figure 3 Application of dummy modeling compared to human being modeling

•in the second step, reach the stage where the model is able to furnish predictive capability expressed in term of injury threshold under a given deceleration law;

• in the third step, have at our disposal a numerically steady model that can be connected with a vehicle model, this model could be, then, integrated into a commercial existing crash computation program.

At the present time, neck and head are correctly calibrated. For thorax, in addition to the seat belt representation, tolerances versus air bags have to be reevaluated. For pelvis, experimentation on tolerances are going on, but some problems are encountered with bony structure representation. Nevertheless, the final first result for a global applicable model of human being is expected for year 1997.

ACCIDENT AVOIDANCE

Several directions have been taken into account for measures aiming to avoid accidents. They are principally attached to improve the visibility, define ergonomic conditions for driving especially with the growing rate of information coming to the driver with the increase of electronic in the car, make a more accurate diagnostic of driver behavior, enhance the conditions to obtain a still better road holding with active suspensions.

New Technologies for displays

The project had for purpose to make easier the reading of information by the driver. New display systems, using Liquid Crystal Display have been tested in order to obtain an effective and optimized use by the driver.

Two running mock-up have been built on the basis of a VAN multiplex architecture and a LCD display at the basis of the windshield.

The technical feasibility have been established and the technical characteristics of the multiplexed network are clarified.

This work open the way to an evaluation in terms of ergonomy and safety, the development of other display systems as Head Up Display or Tactile Display and on-board computers of new generation.

On-Board System for Driver Behavior Study

For several years, a lot of work has been engaged in France to diagnose the driver's loss of vigilance. The method consisted usually the seek a correlation between the actions exerted by the driver on the command actuators of the vehicle (i.e. steering wheel angle) and physiological measurements, as Alpha waves, checked with invasive sensors.

The research in VSR was initiated on similar basis but demonstrated that it was practically impossible to obtain such a correlation even though the number of sensors was great and not only steering wheel angle, but accelerator position, car speed, pressure sensors on different part of the seat, position of the car on the lane measured by video camera, ...

This first methodology has been replaced by a gradual experiment method, developed with Compiègne Technical University, where the initial driver situation is recorded through a neuronal network computer as a starting reference. Other driving situations are diagnosed by fuzzy forms recognition where one neuronal network correspond to one driver state.

A number of real world test performed on highway by LAA (Pr COBLENZ), Ecole Centrale PARIS Laboratories and Peugeot Citroën have demonstrated that the system is convenient, can adapt to the specific driver's characteristics and works in real time on vehicle.

The future steps of this program plan to continue the evaluation on a great number of subjects and equip a mini fleet of vehicles with sensors and computer.

PROTECTION

The VSR projects devoted to passive safety were composed of the several following aspects.

A first range was constituted by theoretical approach on car and occupant modelisation, improvement of our knowledge on material laws, a theoretical and practical approach on computed rupture prediction of mechanical parts, a general study on a new rear safety seats generation and several safety sub-systems relative to frontal and side impact.

Most of these activities will be described in more details in Session 2 with the paper entitled "PSA Experimental Safety SubSystems in VSR".

A New Safety Seat Generation

The purpose of the program carried out by CESA (Compagnie Européenne de Sièges pour l'Autommobile) was to create a fast conception tool for the rear seat definition applicable to the passenger cars of the years 1995/2000 offering new functionality and answering to a stronger safety request.

A number of hygrothermic test have been performed on 8 groups of 10 male and 10 female volunteers up to ambient conditions of 30°C and 57% of relative dampness in order to have a better knowledge of the thermic exchanges at interface seat/occupant.

Taking into account these first elements of comfort, a study in-depth of the use of rear seats by car users led to the SAFI (Siège Arrière à Fonctionnalités Intégrées) concept where the 3 seats are independent, the child restraint systems are integrated, the 3 seat are at the same safety level - what mean 3 point-seat belts and head rests everywhere - and the seat backs give a protection against luggage intrusion.

The passenger installation have been checked with MADYMO computation in order to define the seating area shape without any submarining risk and obtain the loads applied by the dummies and the luggage on the rear seat structure.

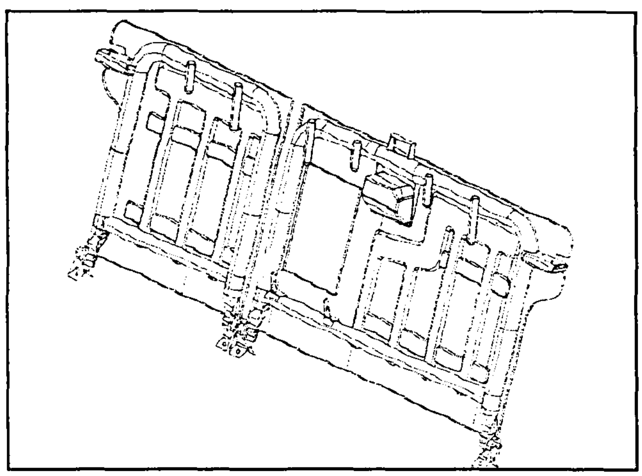

Figure 4 Rear seat back structure

Several technologies reaching the technical specification have been compared :

- complete aluminum structure;
- hybrid steel and aluminum;
- composite aluminum and foam
- TRE
- aluminum honeycomb showing that the integral aluminum solution is lighter but more expensive.

Basic studies

The fundamental crashworthiness studies of VSR are orientated in three main directions : influence of deformation speed on material characteristics, prevision of mechanic parts breaking, simplified fast models for crash and occupant.

The material behavior during impacts is not yet completely controlled; particularly, the sensibility of XES steel vs. deformation speed have been experimented between 0,01/s and 450/s. These test have underlined a significant variation with deformation speed of maximum force and energy of deformation.

A detailed study of the breaking material laws has been initiated for AS7G03 aluminum alloy. This alloy is commonly used for linking parts between engine and engine compartment structure with a great importance in crashworthiness.

On the third subject, project engineering need to use simple models easy to operate and giving a fast answer at the beginning of new projects in order to check the main principles of the new car architecture. Using a Hybrid III dummy model of 2200 finite element associated to a simplified car model of 8000 finite elements, it has been possible to represent the relative displacement of the steering column and the dummy with an accuracy better than 5 %.

Applications to demonstrators

(See for more details paper 96-S2-O-10)

On those basis, Experimental Safety SubSystems (ESSS) were developed for frontal and lateral impact.

ESSS Frontal impact studied more realistic impact conditions against deformable barrier with offset in order to simulate car to car collision. In this frame, a special attention was paid to the submarining problem. This action run into new specifications for embossing and various seat fitting.

On the other hand, ESSS Lateral Impact has been realized by using a schematic global model of side impact deformable barrier made from 5000 finite elements instead of 15000 usually. The final result led to a reduction from 11% to 43% of the car side and values for thorax deflection (30 mm) and viscous criteria (0,9 m/s) far under tolerances.

TRUCKS

The research results on passenger cars, especially for active safety can be transferred partially on trucks. Three main targets for industrial vehicles have been defined :

- lost of steering or rollover of an articulated vehicle as a consequence of strong braking or bad road holding behavior;
- research and development of front, rear and side structures in order to reduce the aggressiveness against other road users;
- development of driving assistance systems for bad conditions and diagnostic control of the mechanical and dynamic behavior of the vehicle.

In addition, these main axis must be based on statistical information coming from the real world of accidents. From accidentology, it appears that :

- mass ratio with trucks is unfavourable for passenger cars;
- the front face stiffness of truck is much higher than passenger inducing enclosure deformation not compatible with occupant survival;
- the ground clearance in front of trucks is generally too high and make easy the embedding of passenger cars.

These consideration have directed the trauck safety program to energy absorption anti embedding device. The first tests have shown that simple disposition can be effective.

SIMULATION

Improved driver education goes through the study and development of effective driving simulators for initial formation and maintenance purposes. Two kind of simulators have to be developed, one for trucks, one for passenger cars.

From another point of view develop efficient simulators for a reasonable cost would allow driving school to enjoy the benefit of a large diffusion of such tools with a significant progress in the teaching quality.

CONCLUSION

At the midway, VSR program begin to give the first results specially with the introduction of new tools aiming to know better the human behavior in the accidents in order to understand more accurately how their occur.

From the technical point of view, VSR has brought new tools adapted to the conception of safer vehicles, modern fittings able to increase the conspicuousness, to give a greater protection in case of accident.

The French national program on ground transportation PREDIT has been closed in February last year and a new program is starting now in which VSR will continue to be developed.

Through the axis to be priviledged one can mention vision enhancement, improvement of road holding models to be used on road simulators, Radar and Lidar for AICC and anti collision, on-board software reliability, effectiveness of new safety device, protection devices against fixed obstacles and protection of other road users as children, 2 wheelers and pedestrians.

ACKNOWLEDGEMENT

The author would like to express his best thanks to the members of the PREDIT Commission members in charge of Safety and enhanced car functions whose the held has been decisive to allow VSR Program to live. The author would also thank the administration representatives and manufacturers who took part into the program.

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MODELING OF ADAPTIVE PASSENGER AIRBAG SYSTEMS IN CAR FRONTAL CRASHES

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Paper Number 96-S3-W-10

ABSTRACT

The objective of this study was to evaluate whether the effectiveness of a passenger airbag can be increased by making the passenger airbag adaptive to crash conditions, such as severity and type of impact, belt use, and passenger size. Such an adaptive passenger airbag will better cover a wider range of crash situations than a conventional airbag system.

In this study, mathematical modeling was performed for the optimization of an existing passenger airbag restraint system. The final solution can be used as reference for the design of an adaptive airbag system. The present study was divided into four steps: (1) a FEM airbag model was developed by using the MADYMO 3D program based on an existing prototype of a passenger airbag; (2) the airbag model was validated against airbag static deployment tests, airbag drop tests, and sled tests; (3) the performance of passenger airbag systems of different volumes was evaluated in terms of injury criteria for front seat occupants of different sizes; (4) a parameter study with different impact conditions and varying levels of variables was carried out to find an optimized design of an adaptive passenger airbag restraint system.

Three variables selected for passenger airbag optimization are the airbag volume, the bag pressure, and the size of the ventilation hole. Four input factors that varied in different crash conditions are the dummy size, belted/unbelted, impact speed, and rigid/deformable barrier. The HIC, chest acceleration, chest deflection, neck moment, and axial femur force were calculated in each simulation. The effectiveness of the passenger airbag models was evaluated in terms of the injury criteria of FMVSS 208 and other relevant injury parameters. The influence of the airbag variables on the dummy response was analyzed and discussed.

The size of the ventilation hole was found to have the dominating influence on the performance of the airbag. The best effect of the passenger airbag system could be obtained either by using a small airbag with high initial pressure or by using a big airbag with low initial pressure. The findings in the present study indicate that a midsize volume (120-liter) airbag with variable venthole can give front seat occupants of different sizes better protection over a range of crash severities than a conventional big (150-liter) airbag with only one size of venthole. The results from this study can be used for the development of a future adaptive passenger airbag system.

INTRODUCTION

The concept of an air-filled cushion to protect people in crashes was described as early as 1941, and patents for airbags began to be issued in the 1950s. Airbag restraint systems first came into use in passenger automobiles about twenty years ago. Now millions of cars have been equipped with airbags as standard equipment. Airbags are superior life savers in the most common type of vehicle impact - the frontal crash. Analysis of car crashes indicates that airbag systems reduce many of the serious and fatal head and chest injuries which unrestrained occupants can sustain. The combination of the airbag and the seat belt is the best protection available against serious and fatal injuries (Evans, 1991).

.Proper airbag deployment and the effectiveness of the airbag is dependent on many factors. Generally speaking, these factors can be categorized into two groups. One group is related to the performance of the airbag during its deployment in an accident. The other group is related to vehicle crash conditions. The first generation of the airbags was designed to meet the passive restraint requirement of the FMVSS standard 208. Most existing restraint systems are optimized for the Hybrid III 50th percentile dummy in two crash conditions. One is for an unbelted dummy in a full frontal impact into a rigid barrier at 48 km/h (30 mph). The other is for a belted dummy in full frontal impact to the rigid barrier at 56 km/h (35 mph) (the so called NCAP test). Thus, cars and protective systems are designed to perform optimally in these impacts. However, real world accidents are often different, and the majority of the accidents are of more moderate severity. Most accidents correspond to an impact speed of less than 48 km/h against a barrier that is deformable and positioned off-set or oblique to the vehicle. Due to the US tests at the impact speeds of 48 km/h and 56 km/h against a rigid barrier, the restraint systems are usually unnecessary stiff to the occupants at lower impact speeds.

The benefit with an adaptive airbag restraint system is that the maximum protection for the occupants will be achieved over a range of crash situations by controlling some airbag parameters. New sensor techniques (Jost, 1995) will enable the design of protective systems that adapt to various impact conditions such as impact speed, severity and type of impact, belt use, and passenger size. Which of these parameters that are significant, and how the system should adapt to offer adequate protection in as many types of accidents and for as many types of occupants as possible still remain open questions.

The aim of this study was to evaluate whether the effectiveness of the passenger airbag can be increased by making the passenger airbag adaptive to crash conditions such as severity and type of impact, belt use, and passenger size. For this purpose, it is necessary to find the relevant airbag parameters to control the adaptive airbag system. It is also necessary to find proper values of the parameters that are suitable to keep constant, such as airbag volume/shape. The results from this study can be used as a guidance for the design of future passenger airbag systems.

METHOD AND MATERIAL

The MADYMO 3D program (TNO, 1992) was used to simulate sled crash tests with a passenger airbag and a Hybrid II or III dummy. In this study, the passenger airbag was simulated with the Finite Element Method (FEM). The test rig, car compartment and the occupant were modeled with Multi Body Systems (MBS).

Development of Airbag Models

For evaluation of the passenger airbag restraint system, three FEM models of the passenger airbag were generated, including one reference airbag model and two airbag models with smaller volumes. The reference airbag model had a volume of 150 liters at static deployment. The other two airbag models had the volumes 120 and 90 liters at static deployment.

150-liter Reference Airbag Model

A reference airbag model was created based on a passenger airbag which is equipped on a mid-size passenger car.

The airbag was modeled using the three-node triangle membrane element. The mesh layout of the FEM passenger airbag model consisted of 2248 membrane elements as shown in Figure 1.

The reference airbag was made of high-tensile-strength polyamide 66 (Nylon 66) fabrics. A linear elastic isotropic material model was used in the present simulations of the passenger airbag. Characteristics of the fabrics used in present study is described as follows.

Young's modulus	$E = 24.8 \times 10^7$ Pa
Poisson's ratio	$\nu = 0.3$
Density	$\rho = 682$ kg/m3
Permeability factor	$\eta = 0$
Thickness	$t = 0.00034$ m

Two tether strap fabrics are used to connect the front and the back of the reference passenger airbag. The strap was modeled with a massless linear spring element between two nodes. Each tether strap fabric was represented by two spring elements. The length and stiffness of the spring elements were defined based on characteristics of the tether strap fabrics.

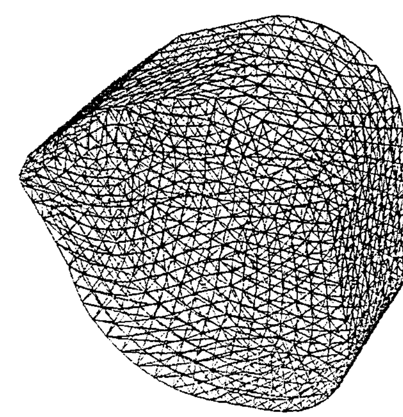


Figure 1. The FEM mesh layout of the 150-liter passenger airbag model (2248 elements, 1126 nodes).

Venthole - There is one venthole on one of the sides of the airbag for ventilation. The original airbag has a venthole diameter of 55 mm. In the MADYMO airbag model, a number of elements were defined to represent the exhaust area of the venthole. In order to investigate the influence of venthole size on the performance of the passenger airbag, additional venthole diameters ranging from 30 mm upto 80 mm were used in the airbag models.

Inflator - The airbag inflation process was simulated by defining the inflator in the MADYMO. The inflator module supplies gas, mass, and heat into the airbag. The mass inflow of the inflator was specified by means of mass flow rate as function of time and the temperature-time function. The thermal properties of gases are described by an ideal gas law equation of state for pressure and volume changes with temperature.

The inflator input data were obtained from a tank (147 liters) test (Figure 2).

Inflator Triggering Sensor - Inflator triggering sensor can be defined in the airbag model. In simulations of the airbag deployment, the inflator is activated by the sensor.

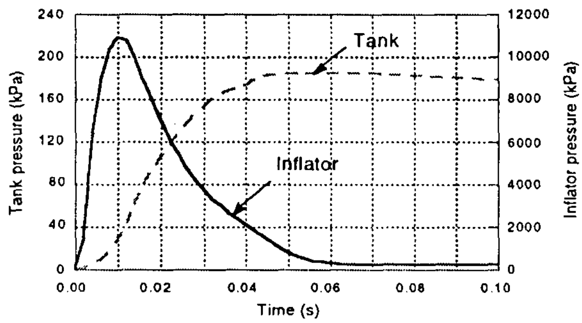


Figure 2. Time history plot of the tank pressure and inflator pressure.

Folded Airbag - Folding of a finite element airbag model, using available mesh generator, is hardly feasible. As an alternative a crumpling technique was used in this study.

120-liter and 90-liter Airbag Models

The reference airbag model of 150-liter volume was scaled down into two additional sizes of 120-liter and 90-liter airbag models (Figure 3). The scaling was only done in the XZ-plane, in order to keep a similar shape of the airbag models in the cross section (in the XZ-plane, X-axis is oriented in the backward direction and Z-axis in the upward direction in the car). The airbag width 350 mm was the same in all three models. The airbag volume was calculated at 60 ms in a static deployment test by using the MADYMO.

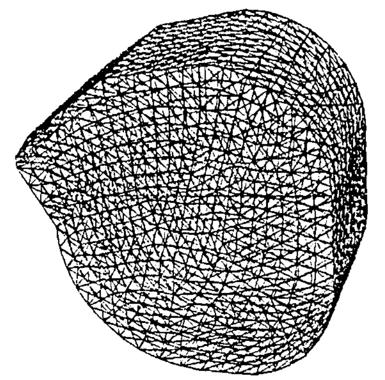
The massflow produced by the inflator was adjusted in the different airbag models, so that the same pressure time history was achieved with a maximum pressure of 20 kPa. The massflow was then scaled for each airbag model so that the same pressure time history curve shape was accomplished with maximum pressure of 10 and 30 kPa, respectively.

VALIDATION

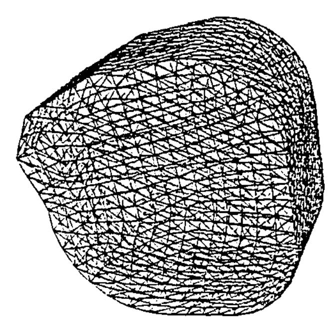
The validity of the 150-liter airbag model was evaluated with data from static deployment tests and drop tests. The airbag model was then verified against a sled crash test. The sled test was carried out using the reference passenger airbag and an unbelted Hybrid-II 50th percentile dummy at an impact speed of 48 km/h.

Validation against Static Deployment Test of the Passenger Airbag

The static deployment test of the passenger airbag was done with a test rig which was constructed based on the geometry similar to that of the car compartment. The test rig consisted of instrument panel with a passenger airbag assembly, and a windscreen.



(a)



(b)

Figure 3 The FEM mesh layout of (a) 120-liter airbag model with 2368 elements and 1186 nodes, as well as (b) 90-liter airbag model with 1944 elements and 974 nodes.

A comparison of the time history plots of the airbag over-pressure from test and simulation is shown in Figure 4.

Validation against Drop Test of the Passenger Airbag

The passenger airbag drop test was performed with the same test rig as in the static deployment test. A dynamic loading was applied to the airbag during its deployment by a chest-form impactor. Simulation of the 150-liter airbag drop test was carried out with a chest form impactor at a speed of 6.5 m/s.

The validity of the airbag model was evaluated by comparing the results from the tests and the computer simulations. The time history plots of the airbag pressure and the deceleration of the chest form impactor is shown in Figure 5. The kinematics from the simulation and the drop test is shown in Figure 6 (Appendix).

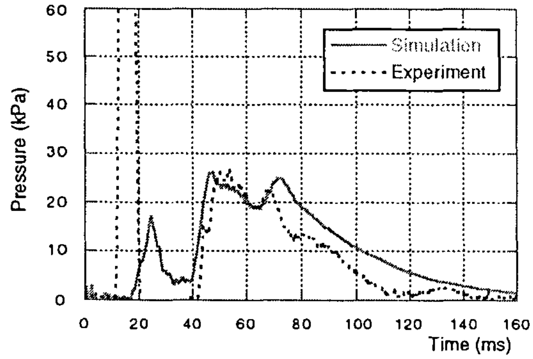


Figure 4. The time-history plot of the airbag over-pressure from the static deployment test and the computer simulation.

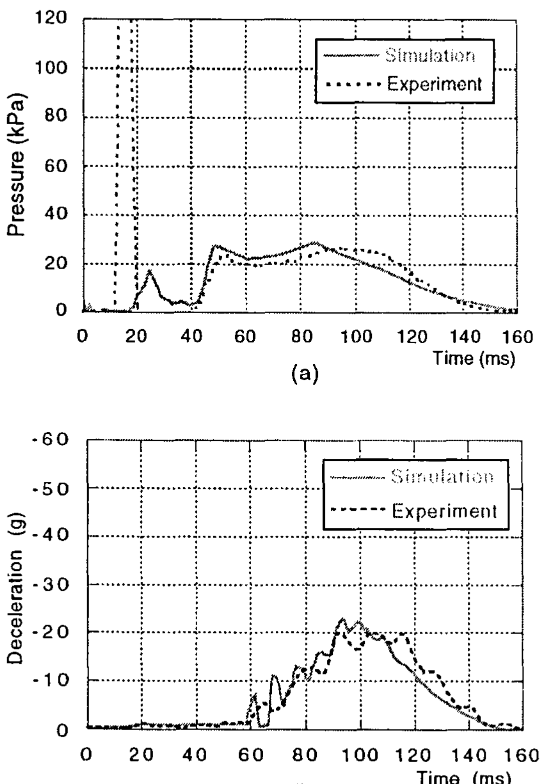


Figure 5. The time-history plot of (a) the airbag pressure, and (b) the acceleration of the chest-form impactor from the drop test and the computer simulation.

Modeling of Sled Test

The 150-liter airbag model was validated against a sled test with an unbelted dummy. The sled test was reconstructed using the MADYMO 3D program.

The configuration of the car compartment model was based on a mid-size car equipped with a passenger-side airbag. The geometry of the passenger-side compartment was modeled. The car compartment model consisted of the instrument panel, the knee bolster, the windscreen, part of the roof, the footwell, the floor pan, and the passenger seat with belt restraint system.

A standard Hybrid-II 50th percentile male dummy model from the MADYMO database was used in the simulations to represent the dummy in the sled crash test.

The configuration is shown in Figure 7. The 150-liter airbag model in its initial state was mounted on the plane element representing the instrument panel. The Hybrid-II dummy was located and adjusted in a normal seating position (mid-rail).

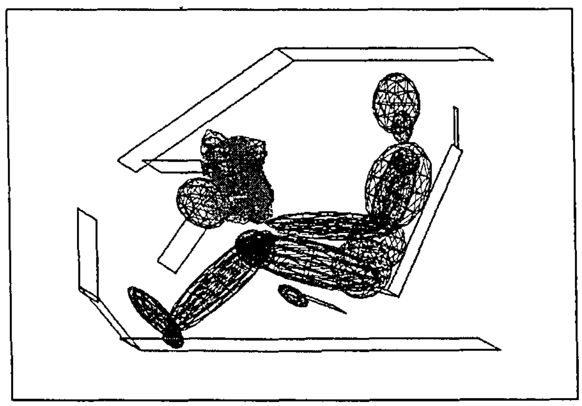


Figure 7. The configuration of the sled test model with an unbelted Hybrid-II dummy and a passenger airbag.

Load Pulse - The recorded acceleration pulse from the crash test was used (Figure 8) as the acceleration field for the Hybrid-II dummy model in the simulation. The sled crash test was performed at an impact speed of 48 km/h.

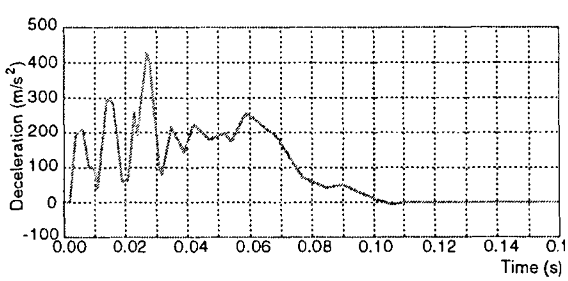


Figure 8. The acceleration pulse from the crash test with a Hybrid II dummy at an impact speed of 48 km/h.

Triggering Time of the Inflator - The triggering time of the passenger airbag is derived from the crash pulse based on the following equation:

$$\Delta v = \int (a - 3g)dt \geq 0.65(m/s) \quad (1)$$

The injury-related parameters: head acceleration, HIC, chest acceleration, pelvis acceleration, and axial femur force were calculated in the computer simulation.

Figure 9 and 10 show the time history plots of head acceleration, chest acceleration, pelvis acceleration, and axial femur force from the computer simulation and the sled crash test.

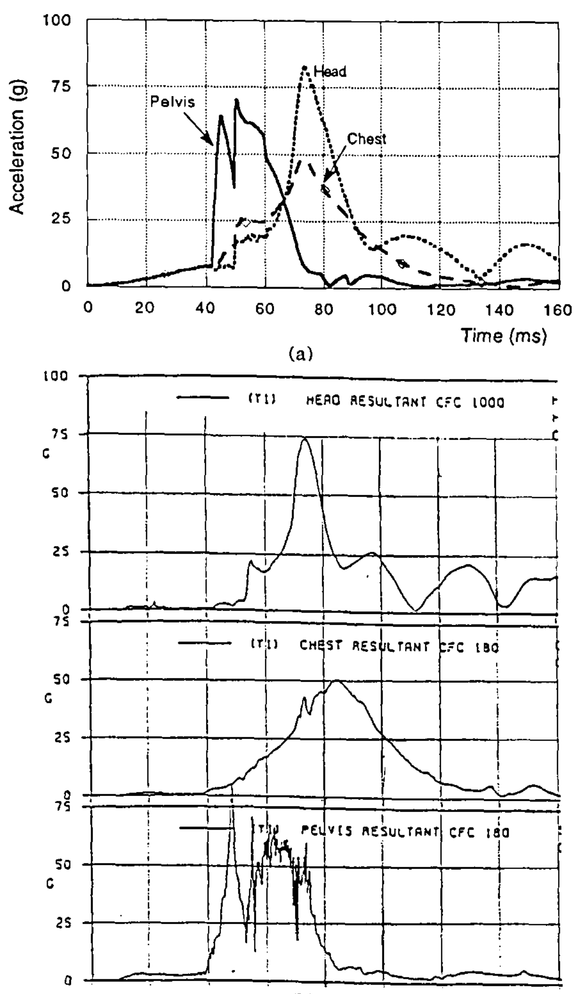


Figure 9. The time-history plots of the injury-related parameters: head acceleration, chest acceleration, pelvis acceleration from (a) simulation and (b) frontal impact test with an unbelted Hybrid-II dummy and a passenger airbag.

In the simulation of the sled test with a passenger airbag and a Hybrid-II dummy, the kinematics of the dummy during the airbag deployment is shown in Figure 11 (Appendix).

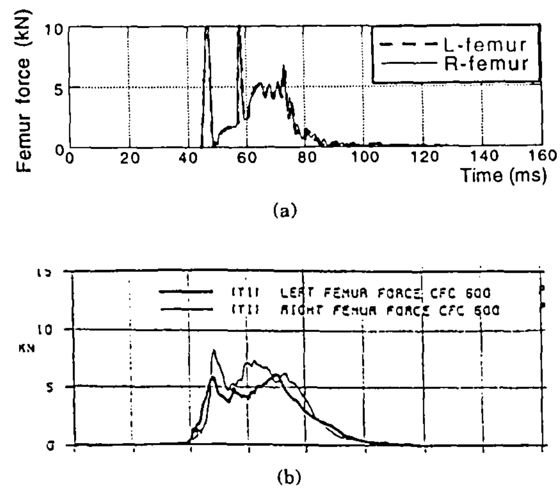


Figure 10. The time-history plots of the injury-related parameters: axial femur force from (a) simulation and (b) frontal impact test with an unbelted Hybrid-II dummy and a passenger airbag.

PARAMETER STUDY

The validated FEM 150-liter passenger airbag model and the volume-varied airbag models were used to carry out a parameter study which consisted of three parts:

- sensitivity analysis of the airbag models,
- analysis of airbag performance with factorial tests,
- investigation of the potential for an optimized adaptive airbag.

In the parameter study, the selected airbag variables and crash conditions are described as follows.

Airbag volume - The original airbag volume of 150 liters was scaled down into two additional sizes: 120 and 90 liters.

Airbag pressure - The mass flow produced by the inflator was adjusted to the different airbag volumes by scaling, so that the same pressure time history was achieved, with a maximum pressure of 20 kPa. The massflow was then scaled for each airbag volume to a maximum pressure of 10 kPa and 30 kPa, respectively.

Venthole size - The original airbag has a venthole diameter of 55 mm. Two other venthole diameters of 30 mm and 80 mm were also used in the airbag models. The varying venthole sizes were implemented in the MADYMO by change of the venthole coefficient.

Dummy size - Three Hybrid-III dummy models, the 5th percentile female, the 50th percentile male, and the 95th percentile male, were used.

Belt system - Both belted and unbelted occupants were considered in the study. A conventional 3-point seat belt system for a 4-door car was used in the simulations.

Impact speed - In the first sensitivity analysis, the simulations were performed for an impact speed of 48 km/h. In the following parameter studies, the simulations of frontal crash tests were performed for impact speeds of 56 km/h (35 mph) and 32 km/h (20 mph). The corresponding pulses can be found in Figure 12. The 56 km/h impact (into a stiff barrier) represents a severe frontal crash, while the 32 km/h impact (into a deformable barrier) represents a "medium" frontal crash.

Barrier - Two barriers were chosen: full frontal rigid barrier and deformable offset (40%) barrier.

Triggering time of the airbag - The triggering time of the airbag inflator was calculated for each crash pulse based on Equation (1). For the rigid barrier at the speed of 56 km/h, the calculated triggering time is t = 4.5 ms. For the deformable barrier at the speed of 32 km/h, the calculated triggering time is t = 19 ms.

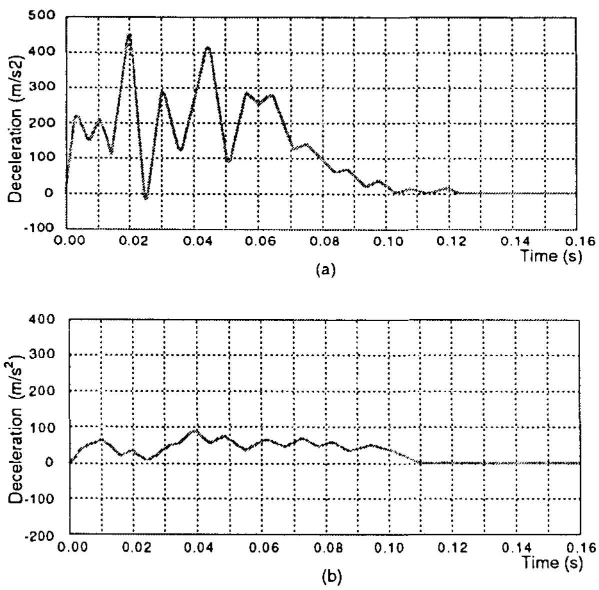


Figure 12. The acceleration pulse, used for the parameter study, from (a) the impact speed of 56 km/h against rigid barrier, and (b) the impact speed of 32 km/h against deformable barrier.

Injury related parameters - The protective capacity of the simulated systems was evaluated by means of the criteria in the FMVSS 208, together with various neck and chest criteria. These criteria and tolerance levels are listed as follows:

HIC: Head Injury Criterion (15 ms)	< 1000,
Chest-g: Chest acceleration (3 ms)	< 60 g,
Chest-D: Chest deflection	< 50 mm,
Femur-F: Femur axial force	< 10 kN,
Neck-M: Neck moment	< 59 Nm.

Sensitivity Analysis of the Airbag Models

The sensitivity and performance of the airbag models were evaluated with three variables: bag volume, bag over-pressure, and venthole size (Table 1). These simulations were carried out with an unbelted 50th percentile dummy against a rigid barrier at an impact speed of 48 km/h.

Table 1
The matrix for simulation with an unbelted dummy (at 48 km/h, full frontal rigid barrier)

Run	Volume (liter)	Pressure (kPa)	Venthole (mm)
1			30
2		30	55
3			80
4			30
5	150	20	55
6			80
7			30
8		10	55
9			80
10			30
11		30	55
12			80
13			30
14	120	20	55
15			80
16			30
17		10	55
18			80
19			30
20		30	55
21			80
22			30
23	90	20	55
24			80
25			30
26		10	55
27			80

Approach for Evaluation of Airbag Effectiveness

The overall risk of whole body injury is determined by summing the individual risks for each body region. This approach enables injury to each body region to be assessed by the available biomechanical criteria and tolerance levels.

The injury criteria for different body region were combined into a single injury criterion index I in the present study. The index I can be calculated in terms of the injury tolerance levels and the related calculated values from each simulation by using the following formula:

$$I = [(\frac{HIC_{15}}{1000}) + (\frac{Cg_{3ms}}{60}) + (\frac{Cd}{50}) + (\frac{M_{neck}}{59}) + (\frac{F_{femur}}{10000})] / 5 \qquad (2)$$

where

HIC_{15} - the 15 ms head injury criterion,
Cg_{3ms} - the maximum chest deceleration (>3ms) (g),
Cd - the maximum chest deflection (mm),
M_{neck} - the maximum moment on the upper neck joint (Nm),
F_{femur} - the maximum axial force on the femur (N).

The results from each run were compared with a reference case in which the injury-related parameters were calculated for the original airbag. The original airbag has a volume of 150 liters, a venthole diameter of 55 mm, and an over-pressure of 20 kPa under static deployment. A quotient (Q) can be calculated to compare the results from each run with the reference case, and to evaluate the overall safety performance of the airbag system. .

$$Q = \frac{I_s}{I_r} \qquad (3)$$

where

I_s the index for the simulation run,
I_r the index for the run with the reference airbag.

Quotient $Q < 1$ means that the airbag is better than the reference airbag for that particular crash situation, if none of the tolerance limits for the various injury criteria are exceeded.

Parameter Study with Factorial Test

An analysis was carried out with a factorial test to investigate the influence of the airbag parameters to the response of the dummy in different crash situations. Five factors were taken into account in the statistical study, including three *airbag parameters* and two *crash situation parameters*. The factors with two levels for each factor are presented in Table 2.

Table 2
Selected factors and levels
for the first part of the parameter study

Factors	Levels –	Levels +
A = Volume	90-liter	150-liter
B = Pressure	10 kPa	30 kPa
C = Vent-hole	40 mm	80 mm
D = Dummy	5% HIII	95% HIII
E = Belt	No	Yes

These 5 factors with two levels were tested at two impact conditions: (1) rigid barrier at a speed of 56 km/h, with triggering time t = 4.5 ms; (2) deformable barrier (off-set) at a speed of 32 km/h, with triggering time t = 19 ms.

The factorial test was conducted with 16 runs at the rigid barrier impact speed of 56 km/h and also 16 runs at the impact speed of 32 km/h with the deformable barrier.

Parameter Study with Selected Variables

In order to exploit the maximum safety potential of a passenger airbag restraint system with one bag volume, an attempt was made to find one size of airbag with variable venthole, which gives a better occupant protection over a wide range of crash situations than the reference bag (150-liter, 55 mm venthole, 20 kPa initial pressure). Further simulations were therefore performed by choosing parameters between the levels used in the former study, such as a middle airbag volume of 120-liter, and a venthole diameter of 70 mm. The selected factors and levels in tests are presented in Table 3.

Table 3
Selected factors and levels
for the second part of the parameter study

Factors	Levels 1	Levels 2	Levels 3
A = Volume	120 liter		
B = Pressure	20 kPa		
C = Vent-hole	40 mm	70 mm	
D = Dummy	5% HIII	50 % HIII	95% HIII
E = Belt	No	Yes	
F = Speed	32 km/h (def. barrier)	56 km/h (rig. barrier)	

Table 4
Factors for the reference airbag test runs

Factors	Runs 1	Runs 2	Runs 3	Runs 4
A = Volume	150 liter		150 liter	
B = Pressure	20 kPa		20 kPa	
C = Vent-hole	55 mm		55 mm	
D = Dummy	50% HIII		50% HIII	
E = Belt	No	Yes	No	Yes
F = Speed	56 km/h (rigid barrier)		32 km/h (deformable barrier)	

For 6 factors and selected levels as shown in Table 3, a simulation test matrix with 24 runs was carried out. To compare with the reference airbag (Table 4), 12 simulations were performed. The performance of the simulated airbags with selected variables was assessed with calculated index I (Eq. 2) and quotient Q (Eq. 3) as well as injury related parameters.

RESULTS AND DISCUSSION

Sensitivities of the Airbag Models

Table 5 shows the results from 27 simulation runs with an unbelted 50th percentile Hybrid-II male dummy at an impact speed of 48 km/h (full frontal rigid barrier). In each simulation the injury-related parameters HIC, chest acceleration, chest deflection, neck moment, and axial femur force were calculated.

The sensitivity of the airbag performance to the change of the airbag variables can be derived from the dummy responses in the simulations.

1. The HIC value is very sensitive to the change of the venthole size for the big (150 liter) and the small (90 liter) airbag. The effect of the venthole size to HIC is not significant for the middle (120 liter) airbag.

   The big airbag with the small venthole was relative stiff to the head, and resulted in a higher HIC value than that with the larger venthole. The small airbag with the large venthole has low capability to absorb the kinetic energy of the head as well as of the upper torso, and resulted in higher HIC value than that with the small venthole.

2. The chest acceleration and the chest deflection are very sensitive to the change of the venthole size for the big and the middle airbag. The effect of venthole size is smaller for the small airbag.

   The distance between the dummy chest and the fully deployed airbag is an important factor for the chest acceleration and chest deflection. It seems that the smaller the distance, the more sensitive is the chest response to the change of venthole size.

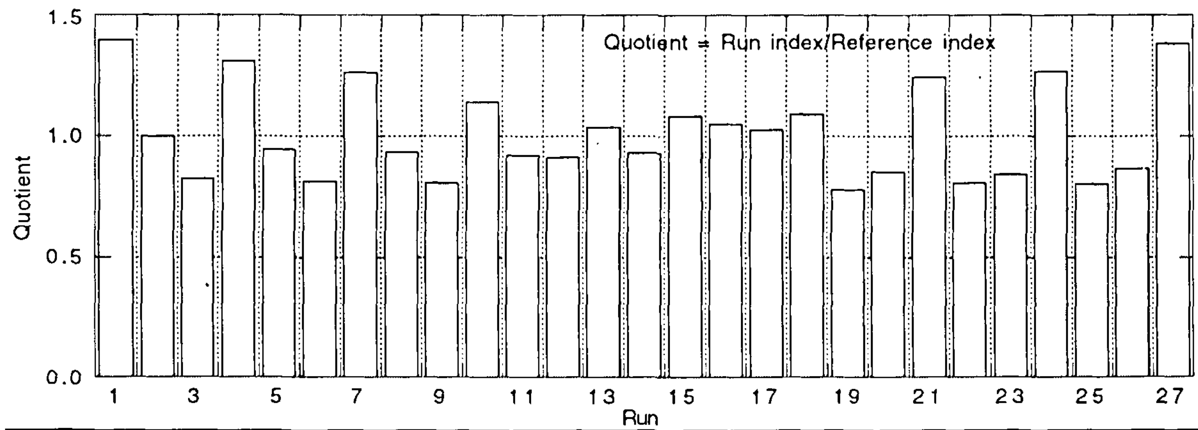
3. The neck moment is very sensitive to the change of the bag volume if the small venthole is used.

   A small venthole increases the stiffness of an airbag. With increased bag volume the duration of contact between head and airbag increases, which increases the extension of the neck joint.

4. There is almost no effect of the different airbag parameters to the femur force for the unbelted dummy.

Table 5
Injury parameters calculated in the first parameter study with an unbelted Hybrid II dummy, impact speed of 48 km/h (full frontal rigid barrier)

Run	Volume (liter)	Pressure (kPa)	Venthole (mm)	Chest-g (g)	HIC	Chest-D (mm)	Neck-M (Nm)	Femur-F (N)	Index	Quotient
1			30	69	635	32	95	8827	0.984	1.40
2		30	55	54	310	25	62	8792	0.728	1.04
3			80	40	112	20	63	8804	0.625	0.89
4			30	64	638	31	91	8845	0.950	1.35
5	150	20	55	53	319	23	57	8827	0.702	1.00
6			80	37	170	14	71	8725	0.629	0.90
7			30	62	568	30	90	8788	0.921	1.31
8		10	55	51	353	24	57	8828	0.706	1.01
9			80	38	175	16	65	8782	0.622	0.89
10			30	60	440	30	77	8814	0.845	1.20
11		30	55	44	317	19	72	9023	0.711	1.01
12			80	30	424	13	70	8937	0.653	0.93
13			30	55	383	29	65	8857	0.773	1.10
14	120	20	55	40	407	18	75	8922	0.719	1.02
15			80	39	620	15	79	8979	0.761	1.08
16			30	52	450	29	69	8941	0.792	1.13
17		10	55	40	537	17	89	8933	0.789	1.12
18			80	40	613	15	79	8995	0.764	1.09
19			30	43	162	23	44	8747	0.592	0.84
20		30	55	30	290	12	67	8949	0.612	0.87
21			80	54	734	19	79	9252	0.856	1.22
22			30	45	321	25	43	8821	0.636	0.91
23	90	20	55	33	463	15	65	8922	0.661	0.94
24			80	55	919	20	74	9027	0.879	1.25
25			30	42	282	23	49	8905	0.633	0.90
26		10	55	33	574	14	65	8970	0.681	0.97
27			80	59	1134	23	80	9049	0.968	1.38



Run	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27
Vh	30	55	80	30	55	80	30	55	80	30	55	80	30	55	80	30	55	80	30	55	80	30	55	80	30	55	80
Pr	30			20			10			30			20			10			30			20			10		
Vo	150									120									90								

Figure 13. The calculated quotients for 27 runs in the first parameter study with Hybrid II dummy, at impact speed of 48 km/h (full frontal rigid barrier). Vh = the size of venthole (mm), Pr = the pressure of the airbag (kPa), Vo = the volume of the airbag (liter).

Figure 13 illustrates the quotient calculated based on simulation results in the first part of the parameter study. The quotient diagram gives a global view of the overall airbag performances.

Influence of the venthole size - Analysis of the quotients in this figure indicates that the venthole is the parameter that gives the largest variation in the results. The result for the 150-liter airbag is much better with the largest venthole than with the smallest venthole. For the 120-liter airbag there is almost no correlation with the venthole size. The 90-liter airbag provides worse results with increasing venthole size.

Influence of the airbag pressure - The low pressure airbag is slightly better than the high pressure bag for the 150-liter volume. The difference between the various pressures for the full size bag is largest with the 30 mm venthole. There is almost no difference with the 80 mm venthole. The results show the opposite tendency for the 90-liter airbag.

Effect of the airbag volume - With high pressure (30 kPa) and small venthole (30 mm), the 90-liter airbag appears to have the best performance. With low pressure and large venthole, the 150-liter airbag appears to have the best performance.

Analysis of Airbag Performance

The absence and presence of interaction between two factors (either airbag variables or/and seat restraint parameters) are analyzed with mean response diagram based on the results from the factorial test runs.

Figure 14 shows HIC mean response diagrams of two-factor interactions between (a) airbag volume and pressure (AB); (b) airbag volume and dummy size (AD); (c) airbag pressure and venthole (BC); (d) airbag venthole and dummy size (CD).

There is no interaction between airbag volume and airbag pressure (Figure 14a). The HIC value goes up with increase of both bag volume and bag pressure. The small airbag with low pressure has the lowest HIC value in this simulation matrix. The HIC value goes up by 15 - 20% when the airbag initial over-pressure increases by three times. The HIC value goes up by 40 - 50% when the airbag volume increases by 67% (from 90 to 150 liter). It is necessary to point out that in present study the airbag width was kept constant as the airbag volume was changed.

Figure 14b shows the interaction between airbag volume and dummy size. For the small dummy, the HIC value goes up quickly with the increasing bag volume. For the big dummy, the HIC value is not very sensitive to the change of volume. A small occupant requires less energy to be absorbed by the airbag, thus allowing the use of a smaller airbag. A large size occupant has more energy to be absorbed by the airbag, thus allowing the use of a big airbag.

Figure 14c shows a small interaction between airbag pressure and venthole size. The HIC value goes up about 20% with decreasing venthole size.

Figure 14d shows the interaction between venthole size and dummy size. For the small dummy (D-), the HIC value goes down quickly with increasing venthole size. The HIC value is not very sensitive to the change of the venthole size for the big dummy (D+).

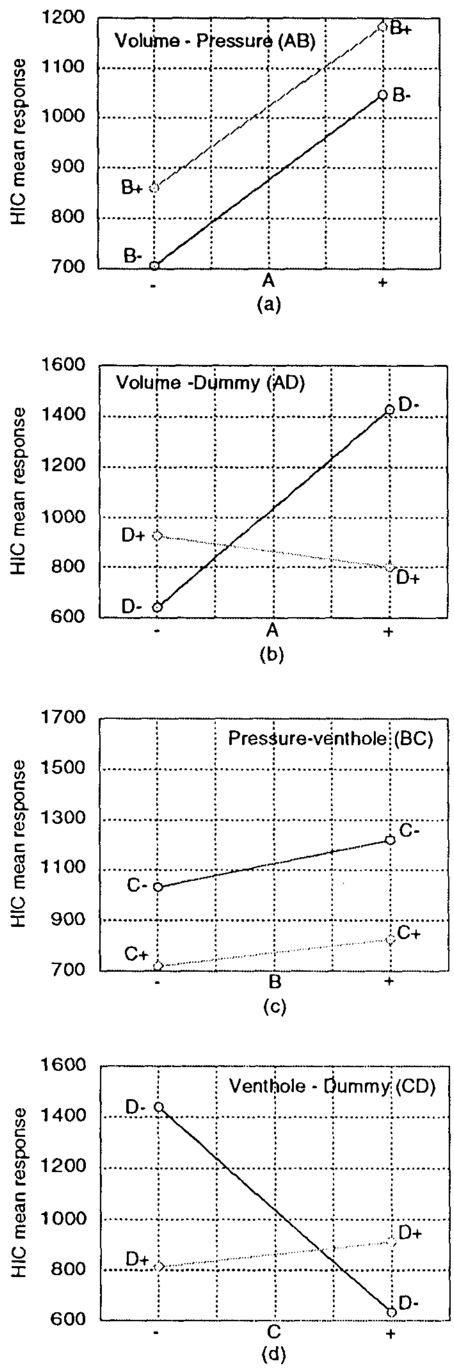


Figure 14. The absence and presence of interactions between: (a) airbag volume and pressure (AB); (b) volume and dummy (AD); (c) pressure and venthole (BC); (d) venthole and dummy size (CD); (rigid barrier at 56 km/h).

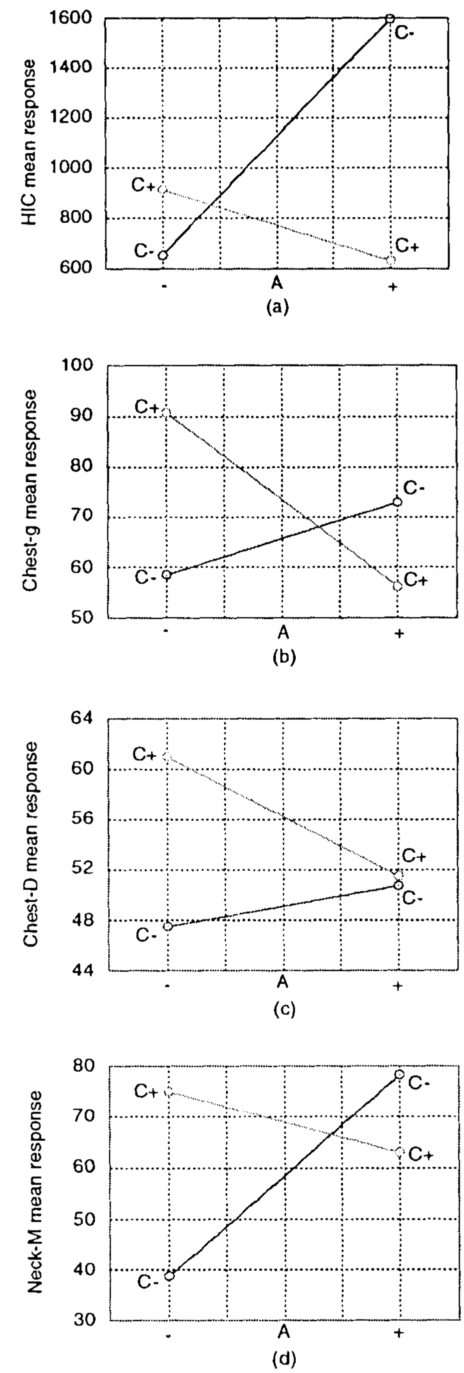


Figure 15. The absence and presence of interactions between airbag volume and venthole (AC) for the responses: (a) HIC; (b) chest-g; (c) chest-deflection; (d) neck-moment; (rigid barrier at 56 km/h).

Figure 15 shows the diagrams of interactions between two factors (AC = the airbag volume A and the venthole size C) in terms of the calculated mean responses for all selected injury criteria: HIC, chest acceleration, chest deflection, and neck moment. The strong interactions between the airbag volume and the venthole size for HIC, chest acceleration, and neck moment can be seen from Figure 15a, b, and d. For the small venthole (C–), the HIC value and neck moment increase quickly with increasing bag volume (Figure 15a and d). For the big venthole (C+), the chest acceleration and chest deflection decrease rapidly with increasing bag volume (Figure 15b and c).

Table 6
Ranking of 2-factor interactions between the airbag volume and the venthole size

Injury parameter	2-factor (Volume[liter]/Venthole[mm])							
	90/40		90/80		150/40		150/80	
	56r	32d	56r	32d	56r	32d	56r	32d
HIC	1	2	3	1	4	4	1	2
Chest-g	1	2	4	1	3	4	1	2
Chest-D	1	2	4	1	2	4	2	2
Neck-M	1	2	3	3	3	3	2	1

56r = Severe crash against rigid barrier at 56 km/h.
32d = Medium crash against a deformable barrier at 32 km/h.

In an alternative way, an analysis of the interactions between factor A and C (the airbag volume and the venthole size) was made in terms of the calculated mean responses. In this analysis, the mean responses were ranked in four levels coded 1 to 4. The code 1 represents the best (lowest) response, and the 2, 3, and 4 represent good, worse, and worst, respectively (Table 6). This analysis indicated that:

- the small airbag with small venthole has the best effectiveness in a severe crash;
- the big airbag with a big venthole gives much better responses for HIC and chest g, and the same or better responses for chest D and neck M in severe crashes than with a small venthole;
- the small bag with a big venthole is a good solution in a medium crash.

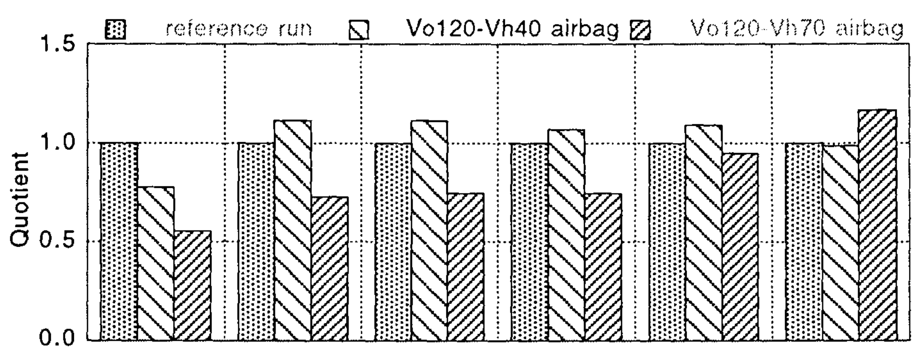
Potential for Optimizing the Passenger Airbag

The results from the simulations with selected variables are shown in Figures 16 and 17 as quotients (Eq. 3). The calculated quotient for each run is presented in Figure 16 for the results from the simulations at the impact speed of 56 km/h with the rigid barrier as well as in Figure 17 for the results from the simulations at the impact speed of 32 km/h with the deformable barrier. The simulation runs were divided into 6 groups in terms of the dummy size and usage of the seat belt. For the sake of comparing the results between simulation runs and reference runs, the results from corresponding reference runs are presented in the same figure. The quotient for the reference bag is equal to 1.

As presented in Figure 16 at the severe impact condition, the quotients for the different sizes of occupant substitutes were on average lower with the 120-liter airbag and dual size venthole compared with the original 150-liter airbag system.

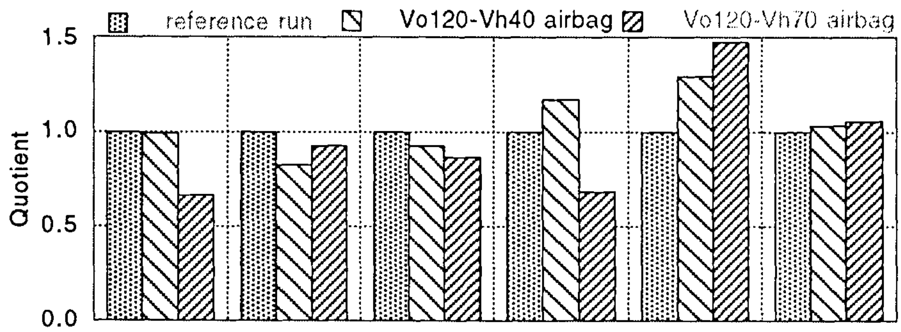
The potential of reduction of the occupant loading for each occupant substitute is analyzed based on the calculated peak values of HIC and chest-g shown in Figure 18(a) and 18(b). The reduction of HIC values is achieved most significantly for belted dummies.

For the 5th percentile female belted dummy (Figure 18, group 1) the injury values are significantly reduced, by 28% for chest-acceleration, and 87% for HIC.



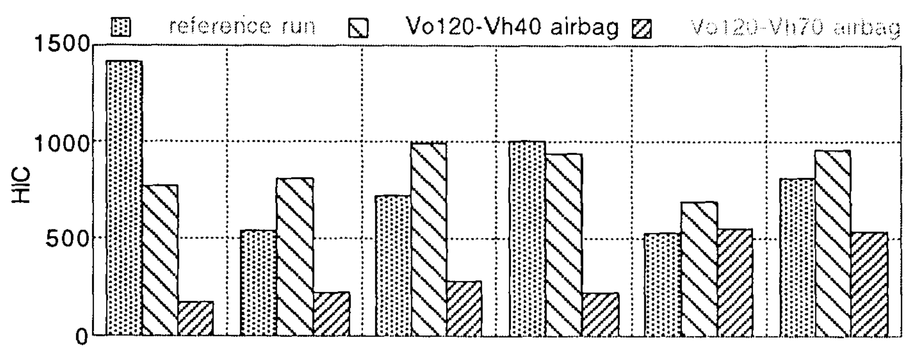
Group	1			2			3			4			5			6		
Vo	150	120		150	120		150	120		150	120		150	120		150	120	
Vh	55	40	70	55	40	70	55	40	70	55	40	70	55	40	70	55	40	70
Dummy	5th			50th			95th			5th			50th			95th		
Belt	Belted									Unbelted								

Figure 16. The calculated quotients for 12 runs with selected variables, at impact speed of 56 km/h (full frontal rigid barrier), the airbag over-pressure of 20 kPa. Vo = the volume of the airbag (liter), Vh = the size of venthole (mm).

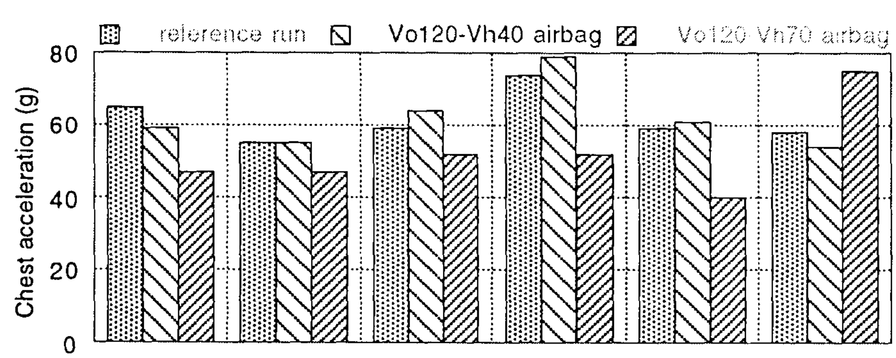


Group	1			2			3			4			5			6		
Vo	150	120		150	120		150	120		150	120		150	120		150	120	
Vh	55	40	70	55	40	70	55	40	70	55	40	70	55	40	70	55	40	70
Dummy	5th			50th			95th			5th			50th			95th		
Belt	Belted									Unbelted								

Figure 17. The calculated quotients for 12 runs with selected variables, at impact speed of 32 km/h (offset deformable barrier), the airbag over-pressure of 20 kPa. Vo = the volume of the airbag (liter), Vh = the size of venthole (mm).



(a)



(b)

Group	1			2			3			4			5			6		
Vo	150	120		150	120		150	120		150	120		150	120		150	120	
Vh	55	40	70	55	40	70	55	40	70	55	40	70	55	40	70	55	40	70
Dummy	5th			50th			95th			5th			50th			95th		
Belt	Belted									Unbelted								

Figure 18. The HIC and chest-g calculated in the parameter study with selected variables, at impact speed of 56 km/h (full frontal rigid barrier) the airbag over-pressure of 20 kPa. Vo = the volume of the airbag (liter), Vh = the size of venthole (mm).

For the 50th percentile male belted dummy (Figure 18, group 2) the peak values are significantly reduced, by 15% for chest-acceleration, and 58% for HIC.

For the 95th percentile male belted dummy (Figure 18, group 3), the peak values are significantly reduced, by 12% for chest-acceleration, and 61% for HIC.

For the 5th percentile female unbelted dummy (Figure 18, group 4) the injury values are significantly reduced, by 30% for chest-acceleration, and 78% for HIC.

For the 50th percentile male unbelted dummy (Figure 18, group 5), the injury values are significantly reduced by 32% for chest-acceleration. There is no reduction for the HIC in this configuration, but the HIC is well under the tolerance level.

For the 95th percentile male unbelted dummy (Figure 18, group 6) the injury values are reduced by 7% for chest-acceleration. In the configuration of run 6b (Vh=40), the HIC is a little higher than for the reference system, but still under the tolerance levels. It therefore could be considered to be included in the optimized system.

Analyses of the results in Figure 16 indicate that in the severe crash type at an impact speed of 56 km/h against a rigid barrier, the 120-liter middle airbag with a large venthole (diameter 70 mm) is a better solution than with small venthole (40 mm), except for the unbelted 95th percentile dummy.

As presented in Figure 17 at the medium impact condition (32 km/h), the same tendency of injury related values as in the severe impact condition can be seen for all sizes of belted and unbelted dummies. The peak values of all injury-related parameters from simulations are much lower than the tolerance levels. The calculated index varied between 0.1257 and 0.3067. For the 5th percentile belted/unbelted dummy and the 95th percentile belted dummy, the middle airbag (120 liters) with a large venthole (70 mm) works better by comparing in group 1, 3, and 4. For the 50th percentile belted dummy the middle airbag with 40 mm venthole works slightly better by comparing in group 2. For the 50th percentile unbelted dummy and the 95th percentile unbelted dummy the big airbag with 55 mm venthole works better by comparing in group 5 and 6. However, the 120-liter middle airbag is fully acceptable since all parameter values are well below their respective tolerance levels.

In the present study, it was found that the performance of the airbag was mainly dominated by the size of the ventilation hole. The best effect of the passenger airbag system for an unbelted occupant in a 48 km/h rigid barrier impact could be obtained either by using a small airbag with high pressure and small venthole or a big airbag with low pressure and a large venthole.

Airbags with different volumes can have the same or similar function by changing design parameters as inflation and ventilation. The study has shown that a conceptual adaptive airbag restraint system could be build by choosing one volume (mid size - 120 liters) with variable size of venthole.

Criteria to assess airbag performance - For evaluation of the occupant protection potential of the airbag restraint system, the most relevant biomechanical criteria were used. The protective capacity of the simulated systems was evaluated by means of the criteria stipulated in the FMVSS 208, together with various neck and chest criteria. In the present study all selected parameters were given equal weight. However, the various criteria have different significance according to the actual occurrence of occupant injury in real-world crash accidents. It is therefore to recommend to use the distribution of the occupant injuries to different body regions in real crash accidents to weight the relative significance of individual responses from crash tests with human substitutes. In this way some criteria may be given higher weight, when evaluating the restraint systems. A weighted index can be defined (as a example) in the following way.

$$I = 0.5\left(\frac{HIC_{15}}{1000}\right) + 0.3\left[\left(\frac{Cg_{3ms}}{60}\right) + \left(\frac{Cd}{50}\right)\right] + 0.15\left(\frac{M_{neck}}{59}\right) + 0.05\left(\frac{F_{femur}}{10000}\right) \qquad (4)$$

SUMMARY AND CONCLUSIONS

This study presents a general approach of using computer modeling of the passenger airbag restraint system in car frontal crashes. The aim is to improve the frontal passenger protection by using an adaptive airbag restraint system. The adaptive passenger airbag can be adjusted to fit different crash conditions, such as severity and type of impact, belt use, and passenger size. This approach can be used in an early design stage which corresponds to the conceptual phase of the development of an adaptive airbag. The improved performance of the adaptive passenger airbag could cover a wider range of crash situations than a conventional airbag system of today, and result in overall lower risk of occupant injury.

The task was thus to optimize the airbag system for different sizes of passengers in different impact conditions. The airbag performance can be varied by varying such parameters as volume, shape, inflation, and ventilation. The possibility to make the airbag system to adapt to various impact situations allows for the introduction of more than one performance into one and the same system, and to choose performance according to the actual crash situation. For practical reasons, the number of performances has to be of reasonable size.

Computer simulations of the passenger airbag system in frontal impacts were performed to find out whether airbags with smaller volume and other characteristics than today's full size passenger airbags can offer adequate protection in a number of different impacts and for different sizes of passengers. The present study was divided into four steps.

In the first step, a FEM airbag model was developed by using the MADYMO 3D program. The model was based on an existing prototype of a passenger airbag with a volume of 150 liters.

Then, the 150-liter airbag model was used to simulate the airbag static deployment test, airbag drop test, and sled crash test with an unbelted Hybrid-II dummy. The validity of the 150-liter airbag model was evaluated by comparing the results from simulations and tests.

In the third step, the performance of the passenger airbag system was investigated in terms of different injury criteria for the front seat passenger, with varying airbag models in various types of impacts and for various size of Hybrid-III dummies. The full size airbag (150 liter) was redesigned to smaller volumes (90 and 120 liter, respectively), by reducing the cross-sectional area but keeping the width the same. Different airbag pressures were obtained by scaling of the properties of the inflator. Three pressures (small - 10 kPa over pressure, medium - 20 kPa, and high - 30 kPa) were used. The triggering-time for the inflator was determined based on the load pulse in the initial phase of the different impacts.

Finally, a parameter study with different impact conditions and different levels of variables was carried out to find suitable parameters for the design of a future adaptive passenger airbag restraint system. The three variables for optimizing the passenger airbag have been the airbag volume, airbag pressure, and size of ventilation hole. The factors for crash situations have been the dummy size, belted/unbelted, impact speed, and rigid/deformable barrier. The calculated parameters from each simulation were the HIC, chest acceleration, chest deflection, neck moment, and axial femur force. The effect of the airbag variables and crash conditions to the head, neck and chest injuries were analyzed and discussed.

From this study the following conclusions can be drawn:

The size of the ventilation hole is a decisive factor for the airbag performance.

The best effect of the passenger airbag system for an unbelted occupant in a severe impact could be obtained either by using a small airbag with high pressure and small venthole or a big airbag with low pressure and large venthole.

A 120-liter middle volume airbag with varying venthole can cover a wider range of passenger protection in both severe and medium crash situations than the present big (150 liters) passenger airbag.

A substantial reduction of occupant loading could be achieved by optimizing the passenger airbag system to a 120-liter middle volume airbag. For instance, the HIC value could be reduced by 58 - 87% for the belted occupants of different sizes.

The present study approach can be used for the development of future adaptive passenger airbag system.

For further studies it is necessary to make:

- evaluation of the most interesting concept of the adaptive airbag system by sled crash tests;
- optimization of the passenger restraint system in terms of both the airbag system and the seat belt system;
- optimization of the passenger airbag for other crash situations (pole, oblique etc.);
- investigation of different airbag shapes.

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APPENDIX

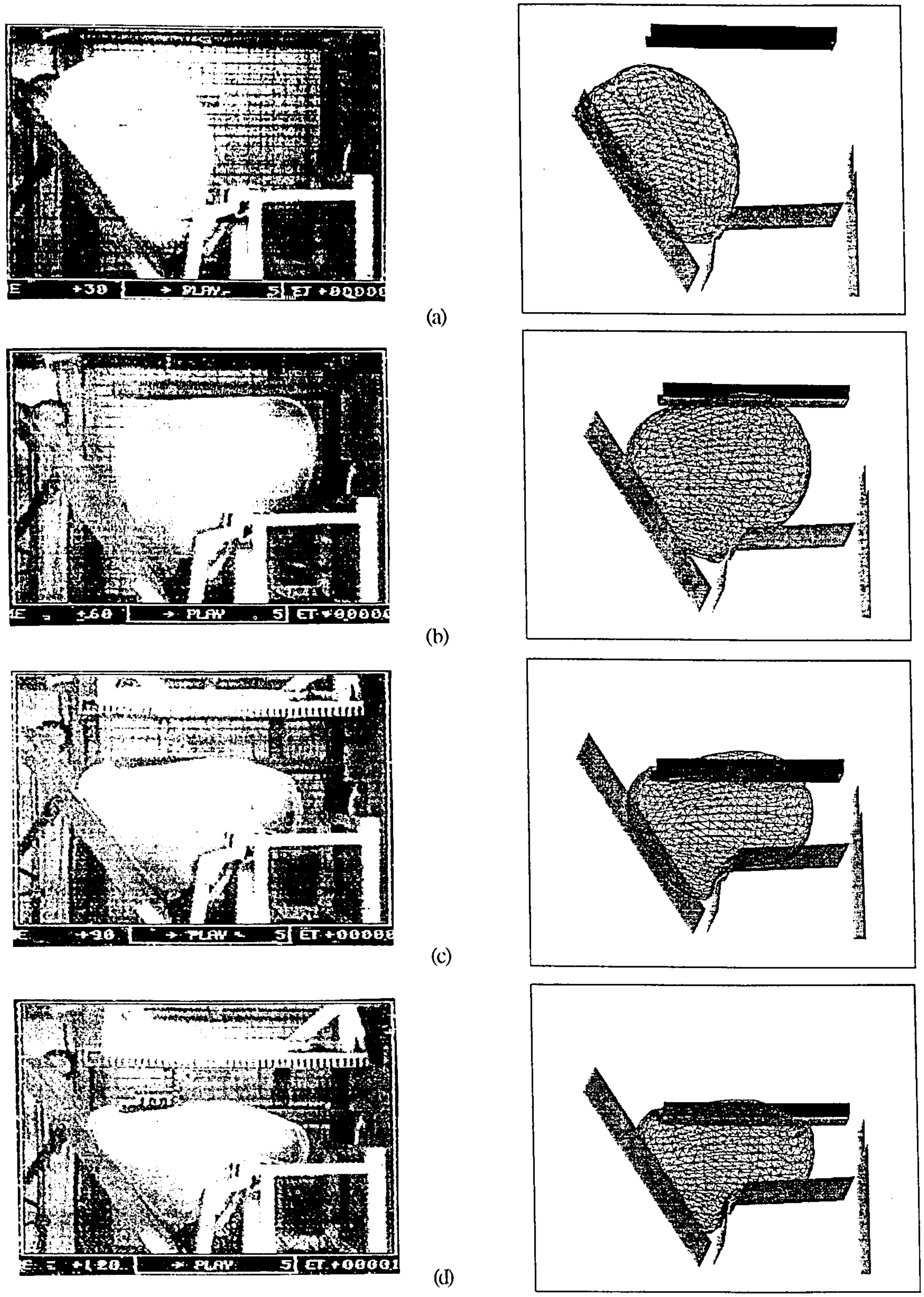

Figure 6. A comparison between an airbag drop test and the corresponding computer simulation for the kinematics of the 150-liter airbag deployment at (a) 30 ms, (b) 60 ms, (c) 90 ms, and (d) 120 ms.

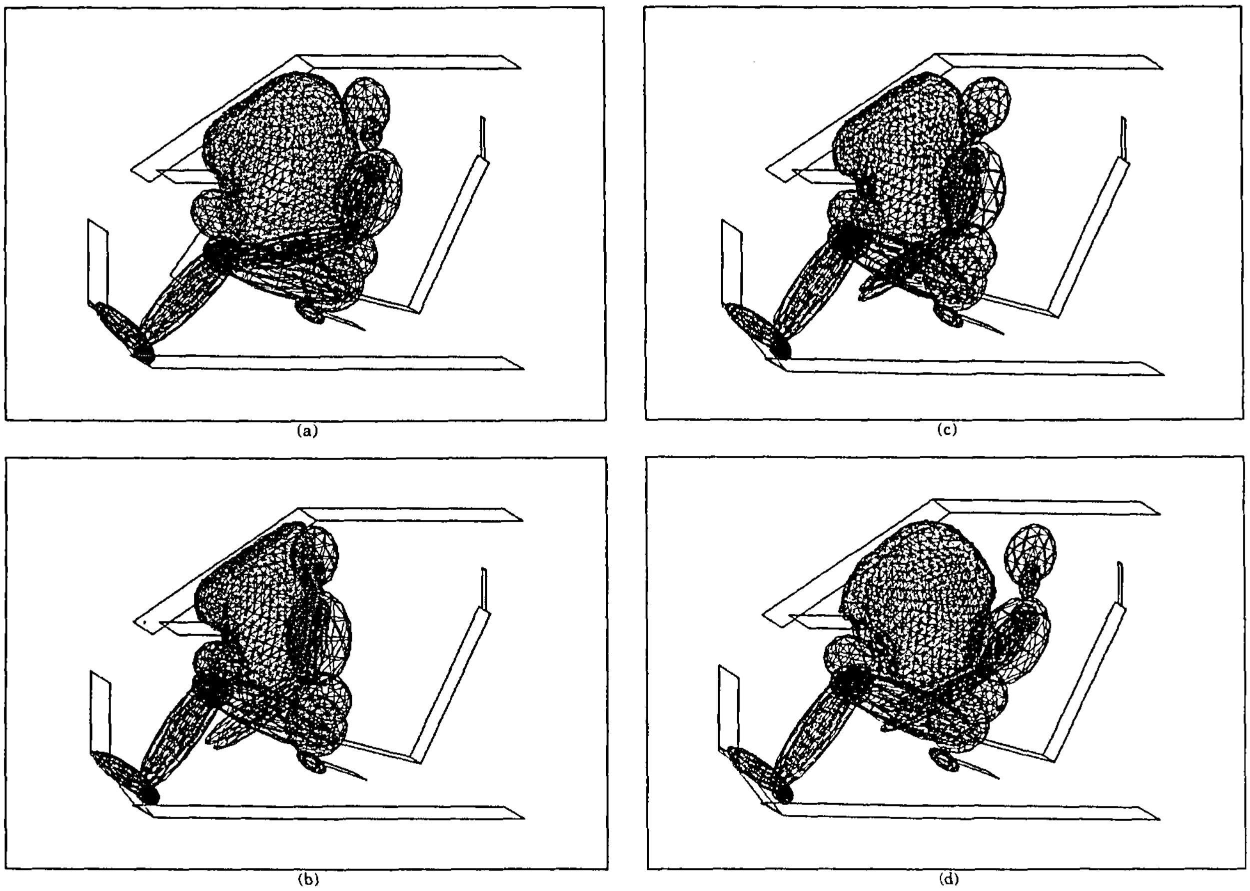

Figure 11. The kinematics from the simulation of the frontal impact with an unbelted Hybrid-II dummy and a passenger airbag at (a) 50 ms, (b) 80 ms, (c) 110 ms, and (d) 140 ms.

DUMMY KINEMATICS IN OFFSET FRONTAL CRASH TESTS

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Paper No. 96-S3-W-12

ABSTRACT

A number of midsize four-door passenger cars equipped with air bags were crashed at 40 mi/h (64 km/h) into a deformable barrier with an overlap of 40 percent of the vehicle width. An instrumented Hybrid III 50th percentile male dummy was positioned in the driver seat with the seat belt fastened. In nearly all these crashes, the air bag and seat belt protected the driver dummy from sustaining loads to the head that would exceed injury thresholds specified by U.S. federal motor vehicle safety standards. However, in several crashes, as the vehicle rebounded and rotated from the barrier, the dummy experienced considerable lateral motion relative to the occupant compartment. This motion suggests that the mostly good upper body injury results may not adequately indicate the risks for occupants of different stature and/or occupants not seated as specified in the Federal Motor Vehicle Safety Standard 208 procedures. High speed film analysis and postcrash vehicle inspection suggest that occupant compartment deformation plays an important role in the performance of the restraint system in many offset frontal crashes. The air bag, seat belt, driver door, and the seat itself are each important components in controlling occupant kinematics as the vehicle rebounds and rotates from an offset frontal crash.

INTRODUCTION

Full frontal, rigid barrier tests and offset frontal, deformable barrier tests are complementary evaluations of occupant protection in frontal crashes. The full frontal test emphasizes the performance of the restraint system while the offset frontal test is more demanding of structural design in preventing occupant compartment intrusion.[1,2,3] In the Insurance Institute for Highway Safety's crashworthiness evaluations of new vehicles, both a vehicle's performance in the U.S. federal New Car Assessment Program test (35 mi/h [56 km/h] full frontal rigid barrier test) as well as its performance in the Institute's offset frontal deformable barrier test (40 mi/h, [64 km/h] 40 percent overlap test) are considered. A vehicle must do well in both tests to assure that it would do well in the range of frontal crashes that occur in the real world.

Although the offset test is considered to be primarily an evaluation of structural designs, the Institute's tests indicate that some aspects of restraint system performance may show up in offset frontal tests but not full width tests. Offset frontal tests can place new, unanticipated demands on restraint systems. For example, increased structural deformation in an offset crash may result in steering column movement or seat movement that compromises the interaction between the air bag and occupant. In addition, the offset frontal crash induces significant vehicle rotation that can expose occupants to potential injury from contacts with side structures that are unlikely to be involved in full frontal crashes. These considerations enlarge the definition of *restraint system* to include the vehicle's seats, doors, and steering column as well as seat belts and air bags.

Restraint system performance and dummy kinematics comprise one of three aspects considered in the Institute's evaluation of vehicle performance in offset frontal crash tests.[4] (The other two are vehicle structure, that is, the extent to which the structure controls and prevents occupant compartment intrusion, and injury measures from an instrumented 50th percentile male Hybrid III dummy.) In this study, the restraint system performance and driver dummy kinematics are reviewed for 16 midsize four-door cars that the Institute has tested.[4] The results of these tests reveal a range of occupant kinematics that illustrates the importance of the various components of the restraint system.

METHOD

The sixteen 1995 and 1996 model year cars were crashed at 40 mi/h (64 km/h) into a deformable barrier with 40 percent overlap. The specifications of the deformable barrier were the same as those recommended by Working Group 11 of the European Experimental Vehicles Committee for a new European frontal crash protection type-approval test, with the minor exception of barrier width.[5] A Hybrid III 50th percentile male dummy was seated in the driver seat with the lap and shoulder belt buckled. The dummy was instrumented with triaxial accelerometers in the head and chest; load cells in the neck, femurs, and tibias; and a displacement transducer in the chest.

All vehicles were equipped with air bags for the driver and right front passenger positions. Three cars were equipped with pyrotechnic seat belt pretensioners, four had dual locking retractors with webbing grabbers, six were dual locking, and three had only vehicle sensitive retractors. In all the cars, except the Chevrolet Lumina, the lap belt buckle was attached to, and moved with, the seat.

Each test was filmed by high speed cameras (500 frames per second) from several angles. The vehicles were marked by targets and inch tape that facilitated subsequent analysis of vehicle and driver dummy motion. The horizontal translation of each vehicle relative to the ground was determined from an overhead view of the crash by recording the longitudinal displacement (forward toward the barrier and rearward away from it) and lateral displacement (left and right of the barrier) of a target marking the precrash center of gravity. The rotation of each vehicle in this plane was determined by recording the angular displacement of the rear of the vehicle (clockwise and counterclockwise) around the center of gravity.

RESULTS

To reduce the likelihood of injurious contacts with hard interior car surfaces during a frontal crash, a driver should remain secured to the seat by the lap and shoulder belt and be protected from injurious steering wheel contact by the deployed air bag.[6] The driver's torso should rotate forward slightly at the hips, until the head and chest contact the air bag. After the initial occupant energy has been absorbed, the driver should move straight back to the center of the seat. This scenario assures that the occupant is decelerated optimally without the threat of injurious contacts with the vehicle interior or exterior. This motion largely describes the action of a belted occupant in a full frontal crash.

In the offset frontal crash test program, substantial variation in the dummy motion was observed among the tested cars. In two of the cars, the 1995 Honda Accord and Subaru Legacy, the kinematic responses were remarkably well-controlled. Several additional cars (1995 Ford Contour, Chevrolet Lumina, Ford Taurus, Toyota Camry, Mazda Millenia, and 1996 Ford Taurus) had similar kinematics, except that the dummies did not move straight back to the seat during rebound from the barrier. Instead, their upper bodies lagged behind slightly as the vehicle translated to the right of the barrier and rotated counterclockwise; as the dummies rebounded rearward from the air bag, the cars continued to rotate and the left B-pillar approached or struck the head. Figures 1-3 show the Mazada Millenia just prior to rebound, at the beginning of rebound and rotation, and at the moment of B-pillar contact.

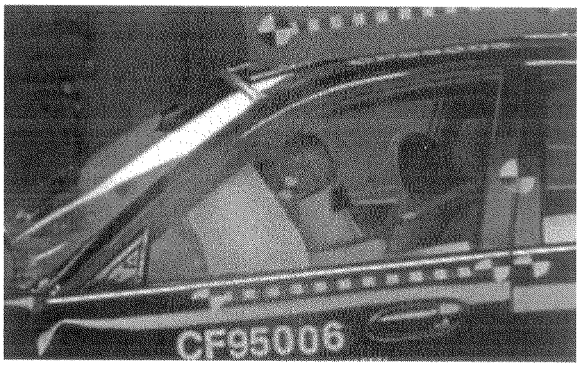

Figure 1. Mazda Millenia near maximum crush

Figure 2. Mazda Millenia during vehicle rotation

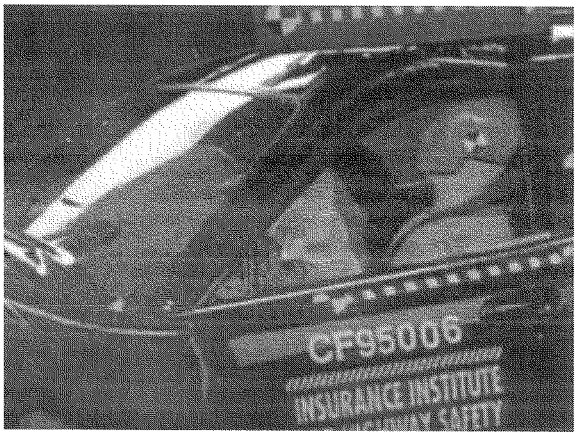

Figure 3. Mazda Millenia B-pillar contacting the dummy head

In other vehicles (1995 Chrysler Cirrus, Chevrolet Cavalier, Nissan Maxima, Saab 900, Volvo 850, and 1996 Toyota Avalon), the body and head lagged even further behind as the vehicle began to rotate counterclockwise; the head turned toward the right of the vehicle as the dummy rolled off the air bag after loading it, and the window sill approached or struck the side or back of the head as the vehicle rotated further off the barrier. Figures 4-6 illustrate this pattern with a series of views of the

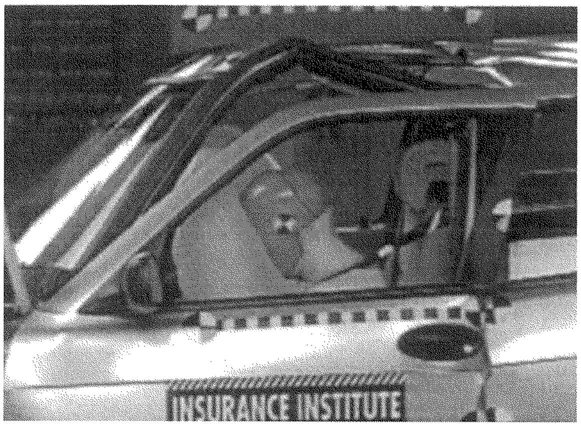

Figure 4. Saab 900 near maximum crush

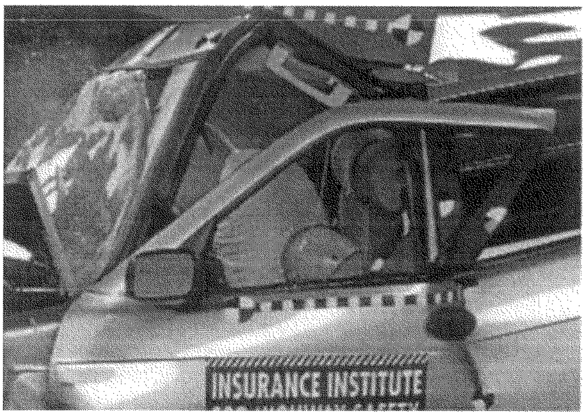

Figure 5. Saab 900 window sill contacting the dummy head

Figure 6. Saab 900 B-pillar approaches head late in vehicle rotation

dummy in the Saab 900 during the moment of maximum crush and the following rebound.

Dummy kinematics in two of the cars, the 1995 Volkswagen Passat and Mitsubishi Galant, had distinctive characteristics not fully captured by this dimension of *rotational lag*, however. In these cars, particularly the Galant, the dummy loaded the air bag somewhat to the left and then slid past the air bag with the head moving toward the A-pillar, which had severely intruded in the Galant.

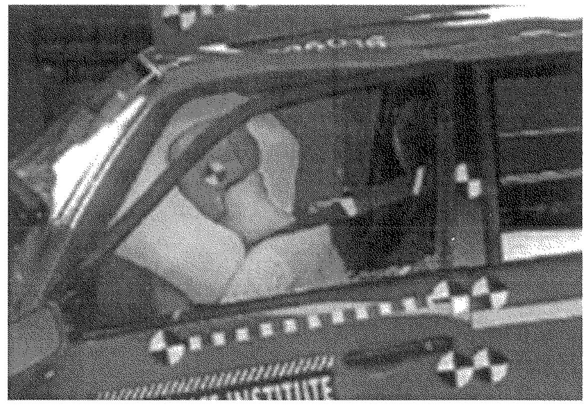

Figure 7. Volkswagen Passat near maximum crush

Figure 8. Mitsubishi Galant near maximum crush

Figure 9. Volkswagen Passat roof rail and exterior door frame contacting the dummy head

Figures 7 and 8 show the dummies in the Passat and Galant near maximum crush. As a result of this unusual behavior early in the crash, the head of the Galant dummy was exposed to the threat of head contact with the intruded A-pillar before returning to the seat, and the head of the Passat dummy was struck by the roof rail and the exterior door frame as the vehicle rotated during rebound (Figure 9).

These observations of kinematic trends illustrate that, when dummy movement is not tightly controlled, the head of the occupant becomes vulnerable to contacts with the vehicle interior, including the window sill, the A-pillar and B-pillar, and the left roof side rail, subsequent to the air bag contact. Interestingly, differences in vehicle motion during the crash appeared unrelated to these differences in dummy kinematics. The lateral displacement of the target marking the precrash center of gravity for the 16 test vehicles is illustrated in Figure 10. Not only did the extent and timing of lateral translation not vary greatly, but the Honda Accord and Subaru Legacy, with the most favorable kinematics, had lateral translation both greater and less than vehicles with less favorable kinematics. The vehicle rotation (counterclockwise) about the center of gravity is illustrated in Figure 11 for the test vehicles. (The results for the Honda Accord and Subaru Legacy are highlighted in Figures 10 and 11.) Rotation varied considerably more

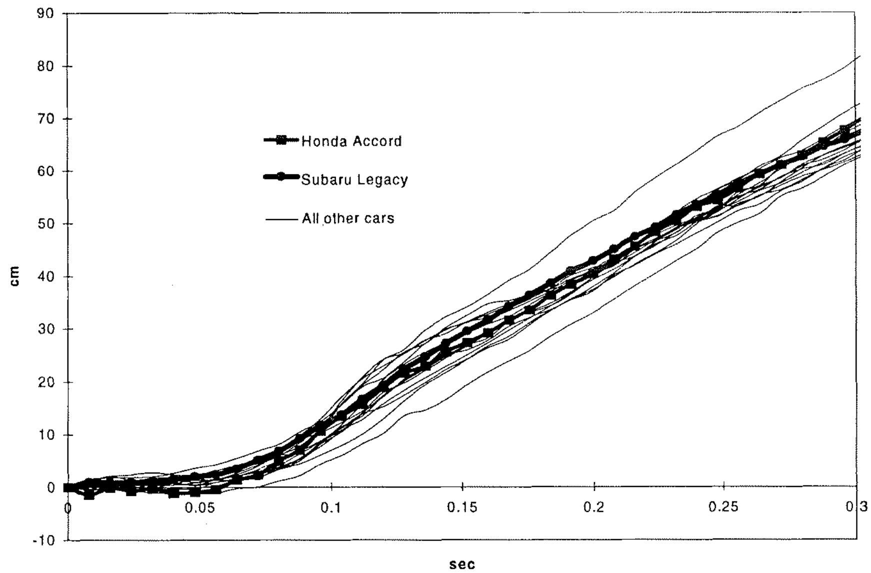

Figure 10. Lateral displacement of vehicles' centers of gravity

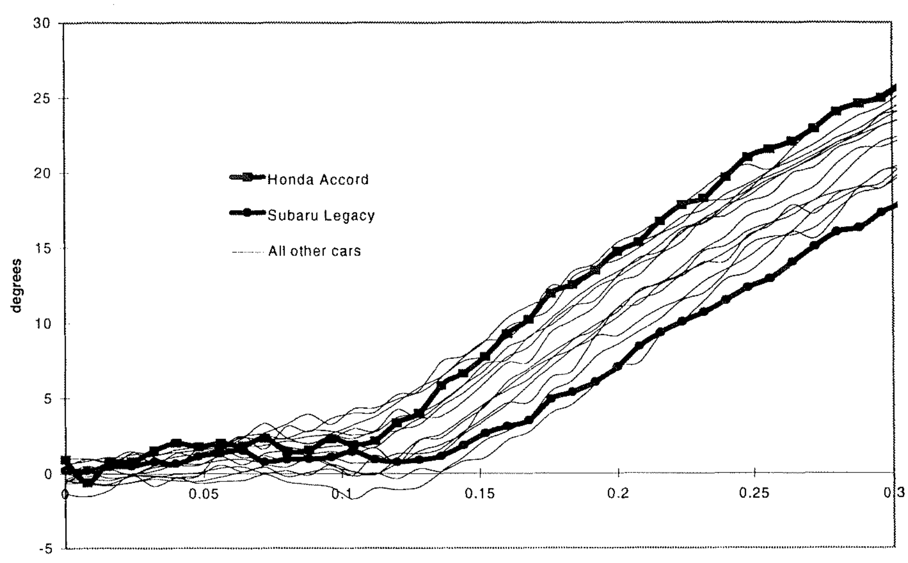

Figure 11. Rotation about the vehicles' centers of gravity

than simple lateral translation, but this variation still failed to account for differences in dummy motions. For example, the Honda Accord experienced one of the largest degrees of rotation while the Subaru Legacy had one of the least; yet, both vehicles had favorable dummy kinematics.

These results were somewhat surprising, because one might expect that less vehicle rotation would facilitate good dummy kinematics. However, there is a negative relationship between rotation and the amount of crush. That is, less rotation was associated with greater crush and, hence, more interior deformation. Not surprisingly, differences in interior deformation appear to influence the occupant kinematic response more significantly than differences in vehicle motion.

The favorable kinematics in the Honda Accord and Subaru Legacy were accompanied by remarkable stability of the seat and the door throughout the crash. As shown in Figures 12 and 13, at the moment of maximum crush, the seat of the Accord remained square with respect to the uncrushed areas of the occupant compartment, and the door was unbowed with little collapse of the door opening between the A-pillar and B-pillar. In addition, when the steering columns in these vehicles moved, they moved toward the left side of the vehicle, keeping the air bag in front of the dummy's head and torso, whose inertia caused it to move toward the driver door as the vehicle rotated counterclockwise off the barrier.

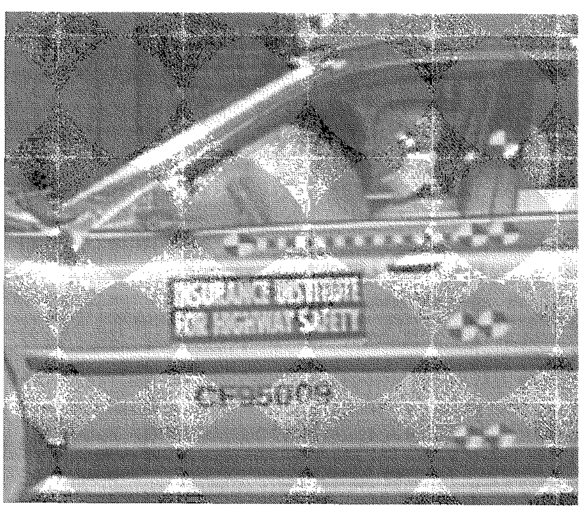

Figure 12. Honda Accord during initial crush

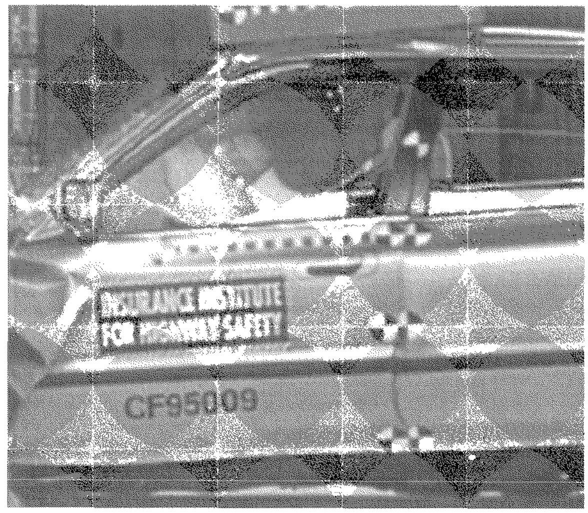

Figure 13. Honda Accord near maximum crush.

These factors, the seat, the door, and the lateral movement of the steering column, are not often included in discussions of restraint effectiveness, which usually focus on the need to absorb occupant kinetic energy. However, when vehicles with less favorable dummy kinematics are examined, each factor appears to have been important in limiting exposure to injurious head contacts.

Seat Stability

The event that appeared to precipitate less than optimal dummy kinematics in many crashes was rotation and/or tipping of the seat. In fact, for many of the vehicles whose only deviation from favorable kinematics was head contact with the B-pillar during rebound, the only common element was moderate tipping of the seat back observed coincident with floor deformation under the driver seat. Figures 14 and 15 show the 1995 Ford Taurus with tipped seat back and floor deformation near the front of the driver's door. A possible explanation for this observation is that, with the exception of the Lumina, the inboard attachment point for the safety belt moves with the seat. If the seat tips forward or sideways during the crash, this permits greater freedom of the pelvis of the dummy, which in turn allows a delay between the beginning of vehicle rebound off the barrier and the time when the lap belt accelerates the dummy to the right with the vehicle. As a result, the left side structure moves toward the dummy before the belt begins pulling the dummy back into the seat. This may be why, in some cases, the head brushes or impacts the B-pillar.

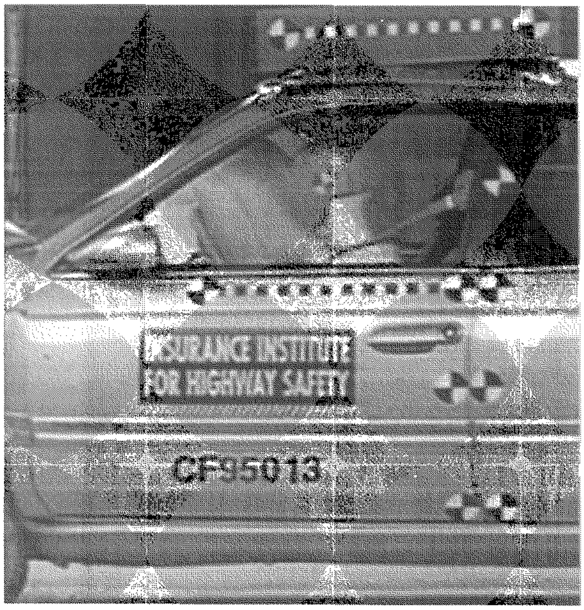

Figure 14. 1995 Ford Taurus before floor deformation and seat tipping

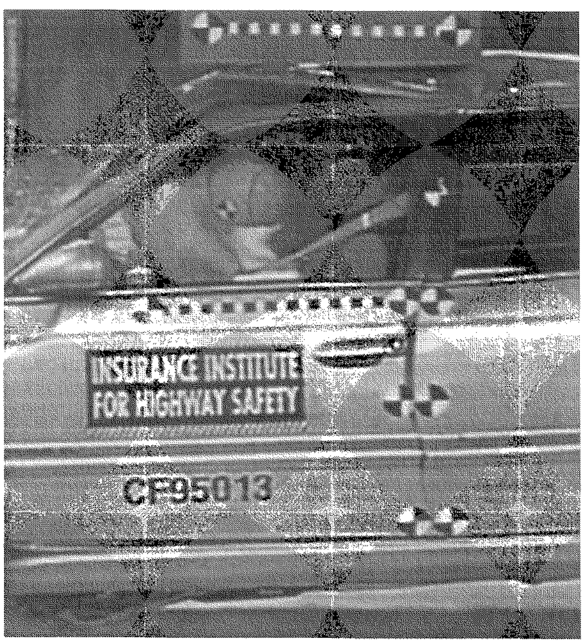

Figure 15. 1995 Ford Taurus with floor deformation and tipped seat

In some cases, the instability of the seat was more pronounced, leading to greater exposure to injurious impacts. In the Volkswagen Passat, the seat tipped sharply forward and toward the driver door as the floor deformed considerably under the driver seat. It is this seat motion that caused the dummy to move left of the air bag toward the A-pillar early in the crash instead of pitching forward to load the center of the air bag (Figures 16 and 17). Similarly, poor control of dummy kinematics in the

Figure 16. Volkswagen Passat before significant seat movement and floor deformation

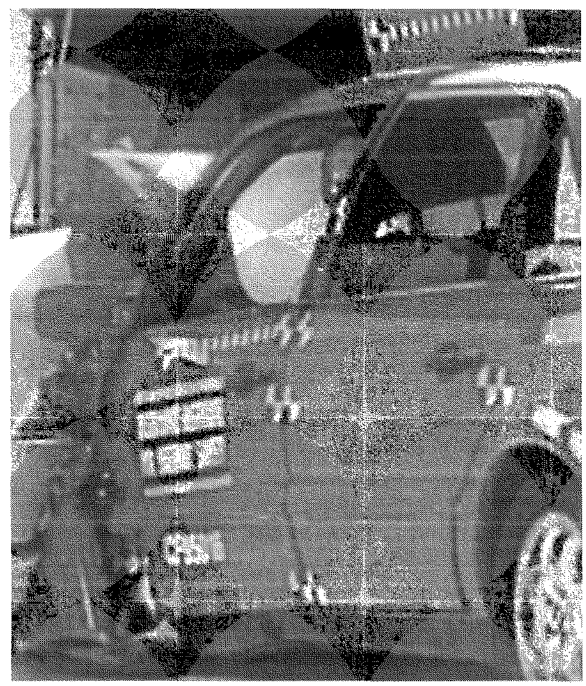

Figure 17. Volkswagon Passat after movement and floor deformation

Maxima was precipitated by failure of the seat to latch in its tracks.

Door Integrity

Although many of the cars had some seat instability, not all of the dummies in those cars experienced severe movement. In part, this was because seat instability was relatively minor for some. However, for others, the critical mitigating factor appeared to be the integrity of the door interior. For example, forward pitch of the driver

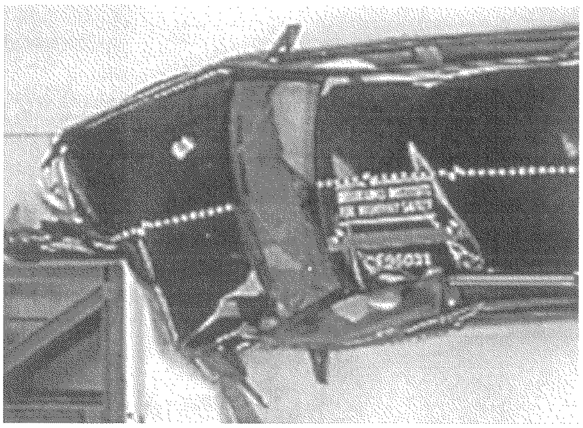

Figure 18. Toyota Avalon door deformation near maximum crush

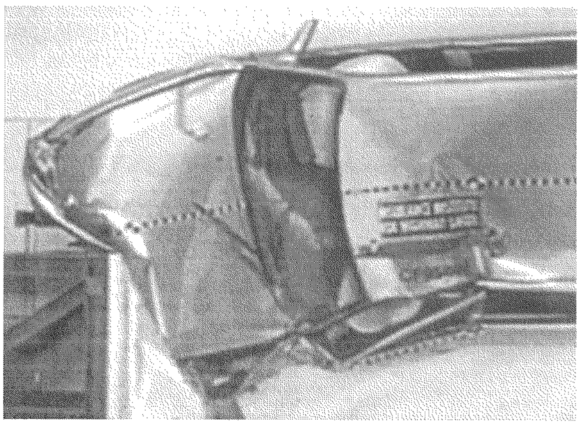

Figure 19. Toyota Camry door deformation near maximum crush

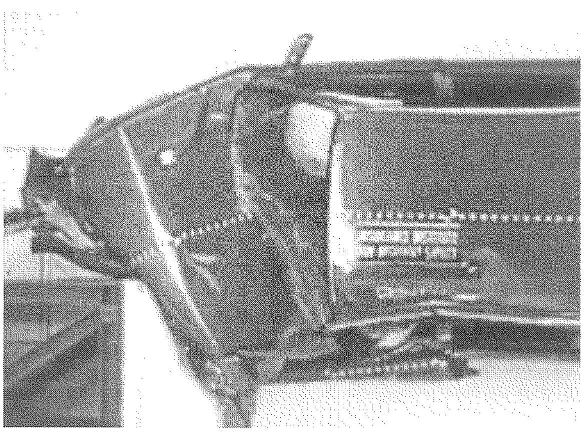

Figure 20. Mitsubishi Galant door deformation near maximum crush

seat in the Volvo 850 was considerable, but the relatively undeformed door loaded the dummy's shoulder, preventing the dummy torso from lagging greatly behind the vehicle rotation. Similarly, the main distinction between the relative motion of the dummies in the Toyota Avalon and Toyota Camry is the relatively better integrity of the Camry's door. Figures 18 and 19 compare the top view of the door deformation in the Camry and Avalon. In the Mitsubishi Galant (Figure 20), the severe bowing of the door left substantial space for the dummy to pass between it and the air bag as the head approached the intruding A-pillar.

Steering Column Lateral Movement

Movement of the steering column is often a negative attribute in occupant crash protection. However, movement that aligns the steering wheel and air bag with occupant motion can be a benefit. That appears to have happened in a number of cases in this series of vehicles. For example, the two vehicles with the most favorable dummy kinematics (Honda Accord and Subaru Legacy) also had the greatest residual displacement of the steering column laterally toward the driver door. In addition, the dummies in some vehicles with somewhat less favorable kinematics (the Toyota Camry and Ford Contour, for example) appeared to benefit from movement of the air bag toward the driver door that kept them from sliding around the left side of the bag. It is also noteworthy that in two of the vehicles with the least controlled dummy movement (Chrysler Cirrus and Mitsubishi Galant), the steering column was actually displaced to the right after moving toward the passenger door during the crash, worsening the alignment of the air bag with the dummy. Figure 21 shows the postcrash lateral displacement of the steering column as measured after the crash for each vehicle.

Differences in these factors, seat tipping, door integrity, and lateral movement of the steering wheel, appeared to contribute to many of the differences in dummy motion, and failures on combinations of these factors produced the poorest kinematic results. It is worth noting, however, that the Chevrolet Cavalier displayed two additional vehicle characteristics that may influence upper body kinematics in a crash. In this crash, the air bag did not deflate at the same rate as other vehicles in this series; in fact, as the dummy loaded the bag, it appeared to provide little or no ride-down (Figure 22).

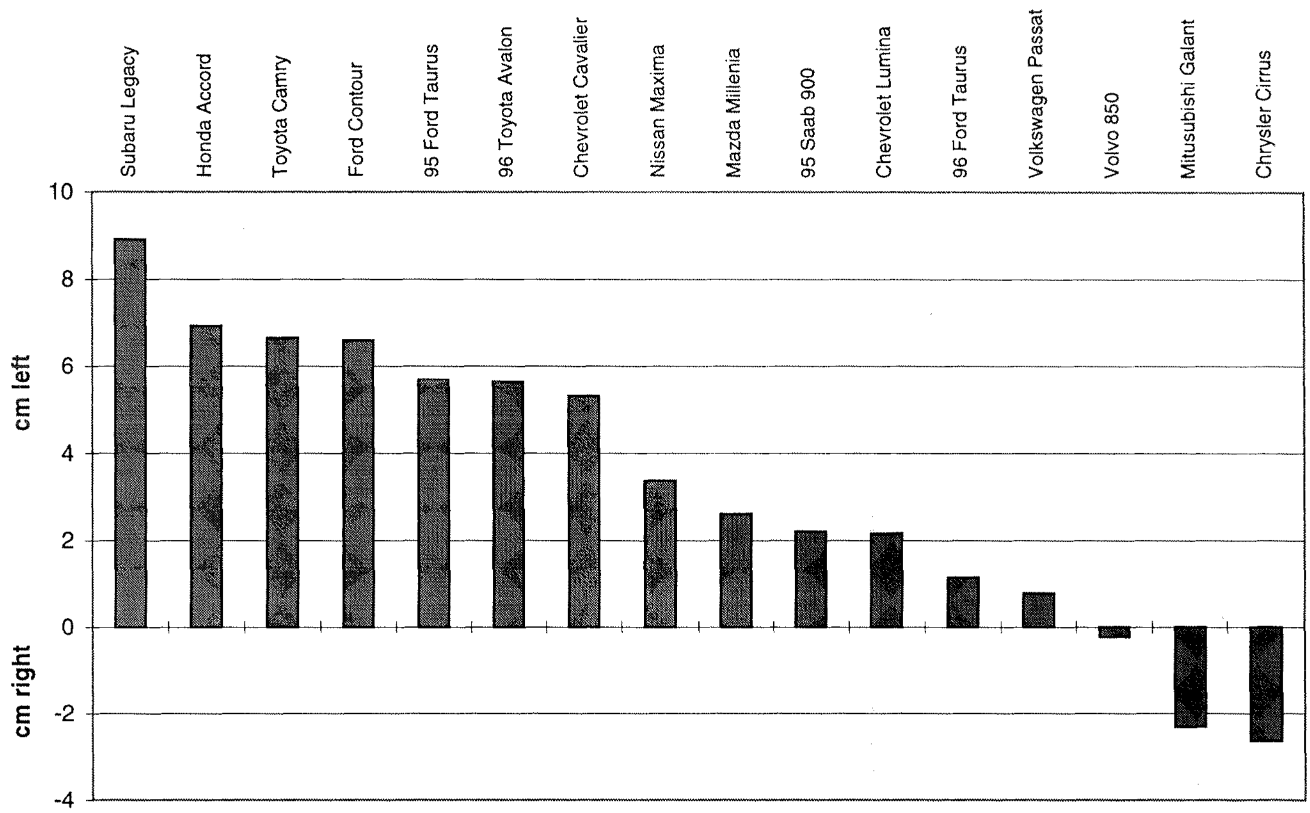

Figure 21. Postcrash Lateral Displacement of Steering Column.

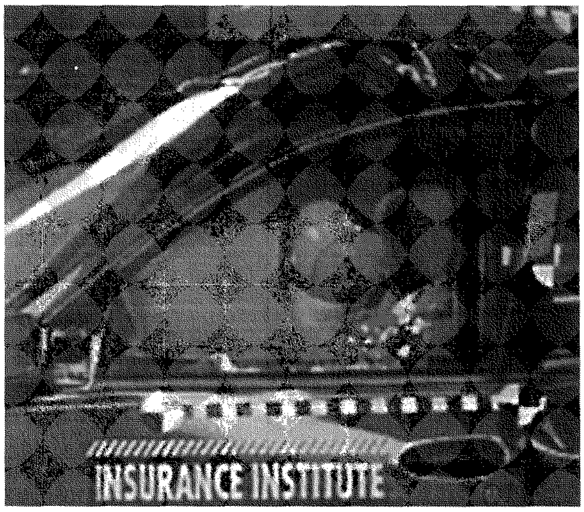

Figure 22. The Chevrolet Cavalier air bag appeared to deflate little as it was driven upward.

This apparently stiff bag was accompanied by upward movement of the steering column, which allowed the chest to continue forward in the vehicle while the head was constrained.

DISCUSSION

Frontal offset crash tests of new cars have revealed large differences in the degree to which the restraint systems control the kinematics of the dummy's upper body during the vehicle rebound and rotation from the barrier. In those cars with well-controlled kinematics, the dummy moved back into the seat uneventfully, while, in those cars with poorly controlled kinematics, the dummy's head lagged behind the rotation of the vehicle. In some cases, the dummy's head approached or struck stiff structure on the side of the car.

Although the source of these kinematics is obviously vehicle rotation during rebound, differences in the rotation or translation did not appear to account for differences in control of dummy kinematics. Instead, the key factor appeared to be the integrity of the occupant compartment and, in particular, how deformation affected the behavior of the interior components that contact or load the dummy: the steering wheel and air bag, the seat, and the driver door. In the vehicles with good upper body kinematics, the steering wheel and air bag tended to align with the upper torso and head throughout the crash; the door maintained a stable surface that the dummy's shoulder loaded as the vehicle began to rotate; and the floor under the seat provided a stable base for the seat and restraint system, firmly tying the dummy to the vehicle motion.

These observations point to a relatively unrecognized problem for protecting occupants in frontal crashes. Even minor deformation of the occupant compartment can be troublesome, if it affects the performance of these systems. Occupant compartment deformation is typically viewed as a source of direct contact injury in a crash, as when a knee is directly loaded by an intruding instrument panel. This series of tests reveals, however, that deformation can also increase the risk of injury indirectly by altering upper body kinematics. When deformation is only enough to tilt the driver seat slightly, it can change the trajectory of a driver's head in a crash so that it strikes an unyielding B-pillar rather than settling back into the driver seat.

For offset crashes as in full frontal crashes, absorbing the occupant's initial kinetic energy without injury is the primary objective in designing a restraint system. Although that is the correct primary objective, the restraint system must also continue to keep the occupant's kinematics under control after the initial energy of the crash has been absorbed and the vehicle has begun to rebound. These 16 crash tests show that the main threat to this control is not the severity of the vehicle motion but the lack of occupant compartment integrity.

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CONCEPT MODELING APPROACH OF VEHICLE STRUCTURE CRASH/CRUSH FEA

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ABSTRACT

This paper presents finite element analysis (FEA) of simplified automotive structures using beam-like elements. The concept is to perform FEA quickly or at early product development stage in order to guide the design process. Two types of finite element models are studied: 1) simplified models which use beam elements to model beam-like components and use other elements for the rest of the vehicle parts such as joints, engines and roofs; 2) thin-walled beam models which purely use thin-walled beam elements. The static stiffness, modal and crash analyses are conducted and the correlation of the results with those from conventional methods are provided.

INTRODUCTION

The concept modeling approach of finite element analysis provides an efficient and practical tool for engineers to assess the vehicle design at an early stage, where the complete design data is not available. It allows engineers to conduct FEA early and quickly to guide the direction of the design and reduce many potential problems down the road. The approach can also be used in other stages of the process to reduce the FEA turn-over time. Its major advantages includes: simplicity, easy modification and feasibility for concurrent design-engineering processes.

Two types of concept models are studied: 1) simplified models which use beam elements for beam-like components and use other elements for the rest of the vehicle; 2) thin-walled beam models which purely use the beam and spring elements. The simplified models can be analyzed by commercial codes for analyses. Since there is no collapsible beam element available in the commercial codes, a thin-walled beam analysis solver was developed. The solver uses the thin-walled beam models for quick nonlinear structural analyses.

The approaches and techniques developed here take advantage of the concept FEA processes: fast speed and simplicity. They are incorporated into a computer program BEAT. The program provides a full range of section generation, analysis and modification capabilities. It automatically calculates beam geometric properties and buckling strengths. In the past a full-vehicle simplified model took months to build. With the computer tool, it takes only days or even hours. It also makes it easy to modify the model simply by modifying the sections, using unique section database. Once the sections are modified, all the properties of the beam elements, that refer to the sections, are updated automatically. The concept FEA approach, presented in this paper, is demonstrated by examples. The accuracies are validated by comparing the results of the concept FEA with those of the detailed model based FEA.

This paper presents the results of crash analysis of an automotive chassis frame. To verify the accuracy the detailed models are analyzed using the nonlinear code Dyna-3D.

The thin-walled beam model FEA process is demonstrated by the crush analysis of an automotive front rail component, using the internally developed beam code. To verify the accuracy, Dyna-3D is used to analyze the detailed model and the results are compared.

THE NEED OF CONCEPT MODEL FEA IN DESIGN PROCESS

In the current automotive structure design process, as shown in Figure 1, designers start with the drawing primarily based on past experiences. The FEA can not be engaged until most of the lines are finished, which often takes more than a year. Then it takes months to establish the detailed mesh of finite element models. The time required to conduct analysis and to modify the mesh often prevents the analysts from feeding the engineering information back to the designers quickly enough, so the FEA mostly functions as a verification tool.

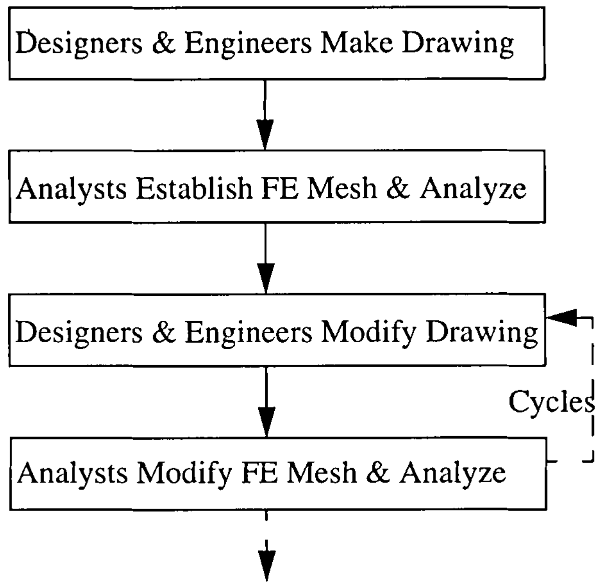

Figure 1. FEA In Current Design Process.

In the new design process proposed in this study, as shown in Figure 2, several simple concept designs can be established in a short time and concept FEA can be conducted on these simple models with reasonable accuracy. With the FEA feedback, the engineers can select the best design with a high level of confidence. The conventional FEA proceeds from there on. This paper, however, is not intended to deal with the design. It focuses rather on the concept FEA process and techniques needed to serve the design process.

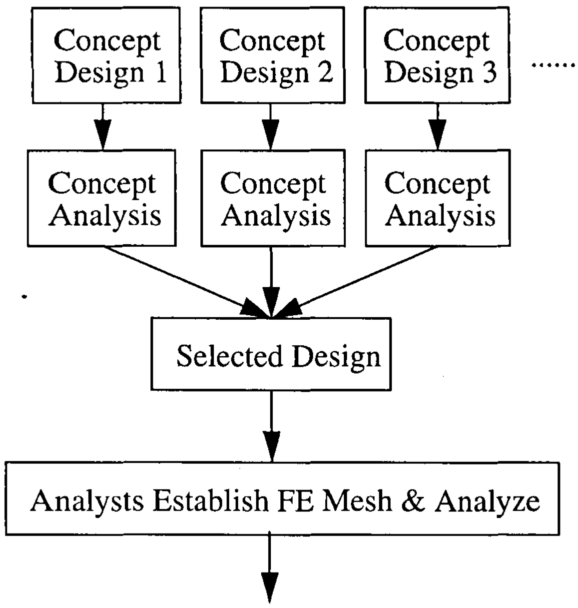

Figure 2. Concept FEA In Design Process.

BUILDING MODELS

To make the concept mode FEA useful, it is important to have the CAE software that utilize the advantages of the approach: simplicity and quickness. The computer program BEAT allows users quickly generated simplified models with many state of the art cutting functions and database features. The program allows the input data from three source:

CAD Line Data - This is useful in the early design stage when there is no FE model available.

Detail Finite Element Mesh - Detailed finite element model can be converted into simplified model easily and quickly. The material and shell properties are kept and transferred into the simplified models. If neither CAD line or FE model available, the FE models with similar geometry can be used. The generated concept model can be easily modified to reflect the new geometry.

Manual Sketching - The graphics interface allows users to draw sections and beams directly on the screen.

SIMPLIFIED MODEL FEA APPROACH

Frontal Crash Analysis of an Automotive Chassis Frame

Figure 9 shows the detailed mesh of an automotive chassis frame. The equivalent mass is lumped at both rails of the frame. The whole frame is moving at 30 m.p.h. toward a rigid barrier in the front to simulate the frontal crash. The nonlinear dynamic code DYNA3D is used to perform the crash analyses of both simplified and detailed models.

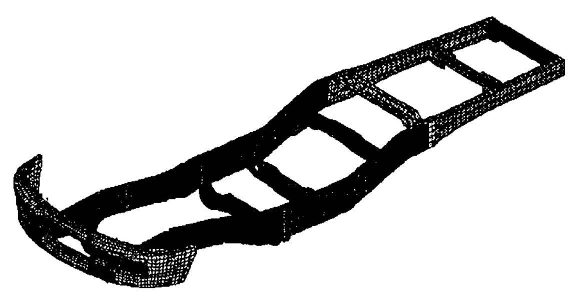

Figure 3. Detailed Model of a Chassis-Frame.

From the detailed mesh, the BEAT program is utilized to obtain the following three simplified models. The first simplified model #1 is simplified at the rear part of the frame and cross members. The detailed meshes modeling the joints remain, as shown in Figure 4. The simplified model #2 is similar to #1 excluding the detailed joints as shown in Figure 5. Those joints are removed and are represented by common nodes. The third simplified model #3 is based on #2 but further simplified to the front end for about 10 inches. The model #3 is shown in Figure 6.

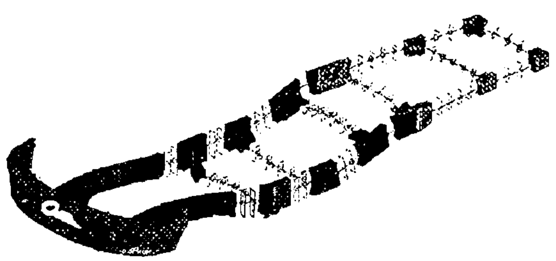

Figure 4. The Simplified Model #1.

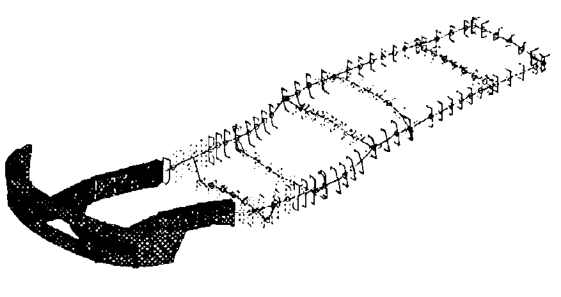

Figure 5. The Simplified Model #2.

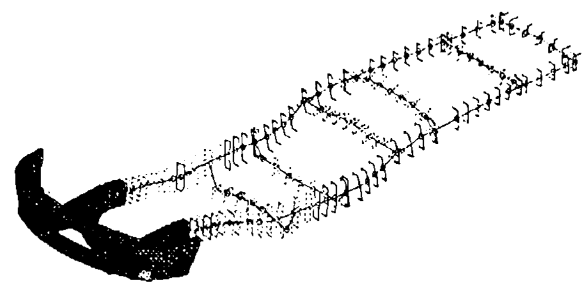

Figure 6. The Simplified Model #3.

The deformed shape of the detailed model is shown in Figure 7. The deformed shapes of three simplified models are similar to the detailed model. The barrier force-time curves of the detailed model and three simplified models are drawn in Figure 8. The curve of the simplified model #1 almost coincides with the one of the detailed model. The curves of the simplified models #2 and #3 show the similar pattern and force peaks, except there is time delay in crashing convolutions after the first peak. The reason is that the joints are represented by common nodes. The stiffness of the simplified models #2 and #3 is higher than that of the detailed models. Despite the time shifting, the overall average crashing forces of the detail model and the simplified models are almost the same.

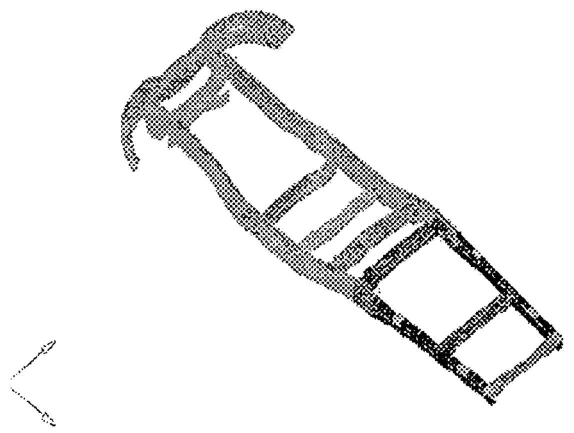

Figure 7. Deformed Shape.

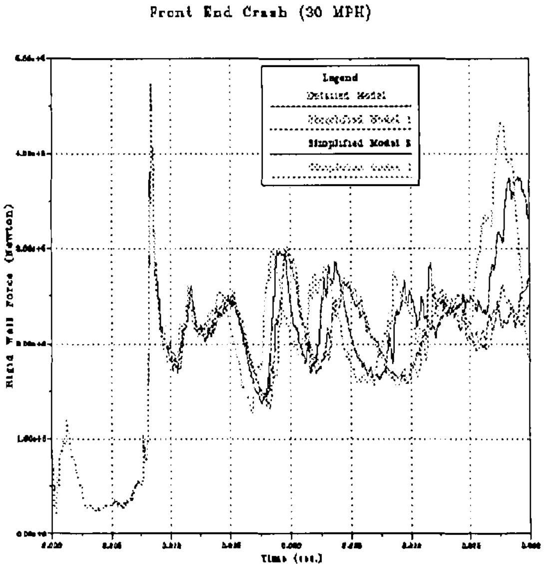

Figure 8. Contact Force -Time.

THIN-WALLED BEAM MODEL FEA APPROACH

Since the beam elements in the commercial crash codes can not adequately predict the large-deformation crush behavior of thin-walled beam components, a nonlinear structural analysis code with thin-walled beam element was developed. The element incorporates the physical shape of the sections at each end. The thin-walled sections are connected to form plates. A thin-walled beam can be considered as consisting of several plates. The buckling phenomenon of these plates can be considered based on the closed-form plate buckling theory. Unlike shell elements, the beam element is assumed to fail only at the ends. Different modeling guidelines from those for detailed elements need to be followed for the thin-walled beam models.

Crush Analysis of a Automotive Front Rail

Figure 9 shows the detailed model of an automotive front rail. By using the BEAT program, the rail is simplified into a thin-walled beam model, consisting of a number of beam elements, as shown in Figure 10. To simulate the frontal crush a large force is applied at the front end. The other is fixed. The thin-walled beam solver is used to analyze the model. To verify the accuracy, a commercial crash code is used to analyze the detailed model with the same boundary conditions and the load. The force-displacement curves from the both analyses are presented in Figure 17. There is about ten percent difference in the peak forces and the average force between the two models.

The times required to modify and analyze the thin-walled beam model are only fractions of the times for the detailed models. Despite the slightly lower accuracy this method can serve as an effective guiding tool for early crashworthiness design of automotive structures and components.

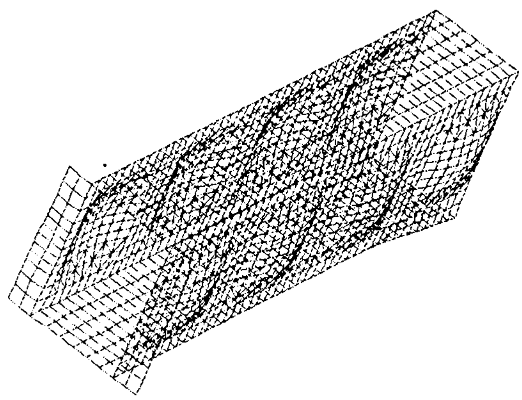

Figure 9. Detailed Front Rail Model.

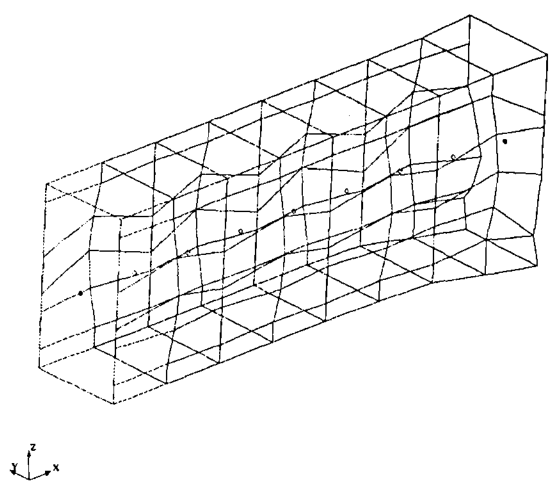

Figure 10.Thin-Walled Beam Front Rail Model.

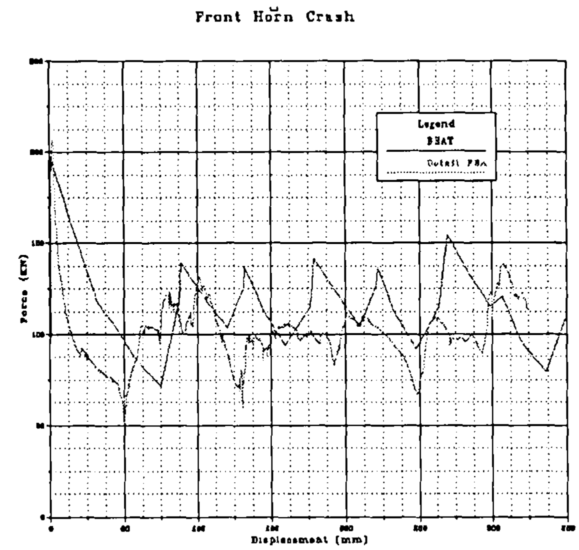

Figure 11. Force - Displacement Curves of Detailed and Thin-Walled Beam Models.

CONCLUSIONS

Simplified finite element and thin-walled beam models are efficient tools for structural analysis of automotive structures and components to provide directional guidance for design. A CAE software package has been developed to quickly generate simplified finite element and thin-walled beam models from the design line data. It can also convert detailed finite element models into simplified or thin-walled beam models. The static stiffness analysis, normal mode analysis and dynamic crash analysis of simplified models presented in this paper show that this approach is practical, efficient and effective. The error is generally within ten percent.

Because the beam elements in the commercial crash codes can not adequately predict the large-deformation crushed behavior of thin-walled beam components, a nonlinear structural analysis code with thin-walled beam element is developed. The element incorporates the physical shape of the sections at each end. The thin-walled sections are connected to form plates. A thin-walled beam can be considered as consisting of several plates. The buckling phenomenon of these plates can be considered based on the closed-form plate buckling theory. The crush analysis example presented in this paper shows that results, with reasonable accuracy, can be obtained with a fraction of the time required by the detailed models. The simplicity and efficiency demonstrate that this approach can be an effective tool for engineers to assess the crashworthiness of automotive structures at early design stages.

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COLLAPSE AND ENERGY ABSORPTION OF THIN-WALLED FRAME WITH POLYGONAL SECTION

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96-S3-W-19

ABSTRACT

This paper describes our study for side member structure which has great influence on crash energy absorption characteristics and introduces the side member structure with rectangular section which has about the same crash energy absorption characteristic with hexagonal section side member.

INTRODUCTION

There have been ever-increasing demands for higher levels of passive safety in automobile occupants. With small-sized vehicles in particular, we are facing the major problem of reducing occupant injury with a limited crush stroke. To cope with the problem, a body structure that effectively absorbs impact due to collision should be designed and developed.

This paper reports our confirmation that a side member structure featuring a rectangular cross section is capable of ensuring a similar level of energy absorption to that of a hexagonal cross section. This was realized through the combination of the advantages of the thin-walled side member characteristics and addition of several corner lines to ensure equal ratios for the width and thickness of each side-wall.

In doing so, we performed static compression experiments and computer simulations. Those results are also described.

BUCKLING OF RECTANGULAR PLATE

When A Compression Force Is Applied

The differential equation of the buckling plate, as shown in Figure 1, is given below;

$$\frac{\partial^4 w}{\partial x^4} = \frac{1}{D} \cdot N_x \cdot \frac{\partial^2 w}{\partial x^2} \tag{1.}$$

Where, 'D' represents the bending stiffness of the plate, which is defined by;

$$D = \frac{Et^3}{12(1-\nu^2)} \tag{2.}$$

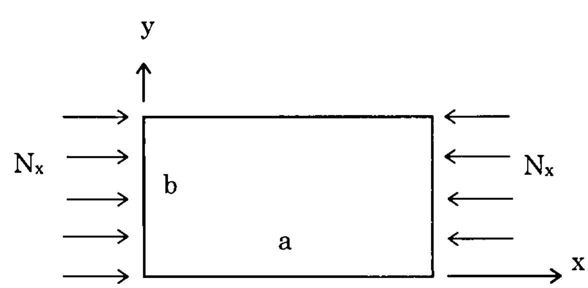
Deflection vertical to the plate is represented by 'w'. Distributed load applying to the edge is represented by 'N_x'.

According to the linear plate theory, the solution of equation (1.) can be determined using the following equation;

$$w = A_{mn} \sin\frac{m\pi x}{a} \sin\frac{n\pi y}{b} \tag{3.}$$

Buckling compression stress of the plate is determined by the equation below, in accordance with equation (3.);

$$\sigma_{cr} = \frac{k\pi^2 \mathrm{E}}{12(1-\nu^2)}\left(\frac{t}{b}\right)^2 \tag{4.}$$



$$x = 0, a : w = \frac{\partial^2 w}{\partial x^2} = 0$$
$$y = 0, b : w = \frac{\partial^2 w}{\partial y^2} = 0$$

Figure 1. Rectangular plate with simply supported edges.

Before adopting equation (4.) to our research, we had to calculate the range of the thickness-to-width ratio (t/b) where equation (4.) is applicable. If buckling stress exceeds the proportional limit, the solution of equation (4.) becomes excessively large.

Assuming that $\sigma_{Y.P}$ = 167 Mpa, E = 206 Gpa, ν = 0.3 and κ = 4 in the case of SPCC material which is widely used for the automobile production, the ratios of b/t and t/b are 67 and 0.015, respectively.

$$\frac{b}{t} = 67 \,, \qquad \frac{t}{b} = 0.015 \qquad (5.)$$

Therefore, the material begins to collapse below the proportional limit, if the thickness-to-width ratio equals 1.5% or below.

Relationship between Postbuckling Behavior of Rectangular Plate and Effective Width

A rectangular plate, as shown in Figure 1, bears a larger load after buckling if it is simply supported at all edges and simple compression force is applied. In such a case, the stress distribution within the plate is not uniform, showing the tendency shown on the left of Figure 2.

Here, we adopted a hypothesis that the stress concentrates on the areas in close proximity to the edge while no stress occurs at the center.

The effective width corresponds to twice of area C, i.e. b_m, on which the hypothetical stress acts.

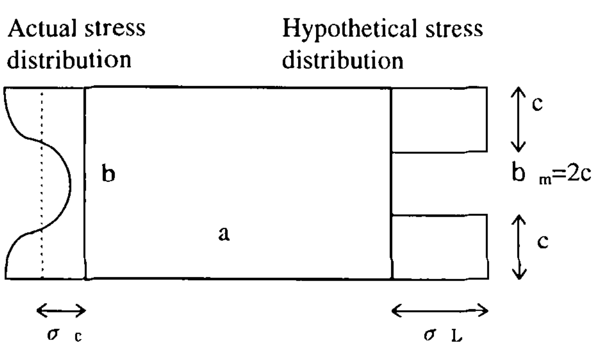


Figure 2. Stress distribution after buckling.

Effective width 2C (b_m) is expressed by the following two equations;

$$\frac{2c}{b} = \sqrt{\frac{\sigma_c}{\sigma_L}} \quad \text{(Kármán's equation)} \qquad (6.)$$

$$\frac{2c}{b} = \sqrt[3]{\frac{\sigma_c}{\sigma_L}} \quad \text{(Maguerre's equation)} \qquad (7.)$$

Where, σ_c is the buckling stress applying to the plate.

According to Eqs. (6) and (7), the relationship between total load P and σ_L is defined by;

$$P = 2c \cdot t \cdot \sigma_L \cdot \sqrt[n]{\frac{\sigma_c}{\sigma_L}} \qquad (8.)$$

By comparing the experimental measurements with the calculation results obtained with equation (8.), we found they conformed to the results obtained with Kármán's equation at immediately after buckling; whereas they conformed to that of Maguerre's equation after the period.

$\frac{\sigma_c}{\sigma_L} > 0.9$ ⇨ conforms to Kármán's equation

$\frac{\sigma_c}{\sigma_L} < 0.7$ ⇨ conforms to Maguerre's equation

equation (8.), however, is not applicable when the stress is over the proportional limit because it pertains to stress below the limit. This is attributable to the fact that the effective width decrease becomes more significant than that assumed by the equation, if buckling stress goes up as thickness increases.

In the practical design process, stress often exceeds the proportional limit. It is required to clarify the behavior of the plate when stress exceed the limit .

Behavior above Proportional Limit

To facilitate the observation of behavior above the proportional limit, it is effective to apply the relationship between the average stress acting on the plate and thickness-to-width ratio. The relationship can be clarified by using Kármán's equation.

Kármán states that the compression stress applying to the effective width equals that acting onto both edges of the plate, and it also corresponds to the buckling stress when the compression stress is 2C. The buckling stress occurring under this condition is determined by the following equation with equation (4.) and the assumption that b = 2c and κ = 4;

$$\sigma_L = \frac{4\pi^2 \mathrm{E}}{12\left(1-\nu^2\right)} \cdot \frac{t^2}{(2C)^2} = \frac{4\pi^2 D}{t(2C)^2} \qquad (9.)$$

The equation is convertible to the following by multiplying $b^2/(2C)^2$ with equation (4.);

$$\sigma_L = \sigma_c \frac{b^2}{(2C)^2} \qquad (10.)$$

Based upon the equation above, Kármán's equation defining the effective width, i.e., equation (6.), is obtained. By assuming that maximum load occurs when buckling stress σ_L reaches yield stress σ_y, the following equation is achievable;

$$\sigma_y = \frac{\pi^2 E t^2}{12(1-\nu^2)C^2} \qquad (11.)$$

Maximum load P_{ult} acting on the plate is;

$$Pult = 2C \cdot t \cdot \sigma_y = \frac{\pi t^2}{\sqrt{3(1-\nu^2)}}\sqrt{E\sigma_y} \qquad (12.)$$

As a result, P_{ult} is proportional to the square of plate thickness t disregarding the value of plate width b.

Average stress σ_{ult} equals the quotient obtained by dividing P_{ult} by the area of the cross section.

$$\sigma_{ult} = \frac{P_{ult}}{t \cdot b} = \frac{\pi\sqrt{E\sigma_y}}{\sqrt{3(1-\nu^2)}} \cdot \frac{t}{b} \qquad (13.)$$

equation (13.) shows the relationship between average stress σ_{ult} and thickness-to-width ratio t/b. The former is in proportion to the latter.

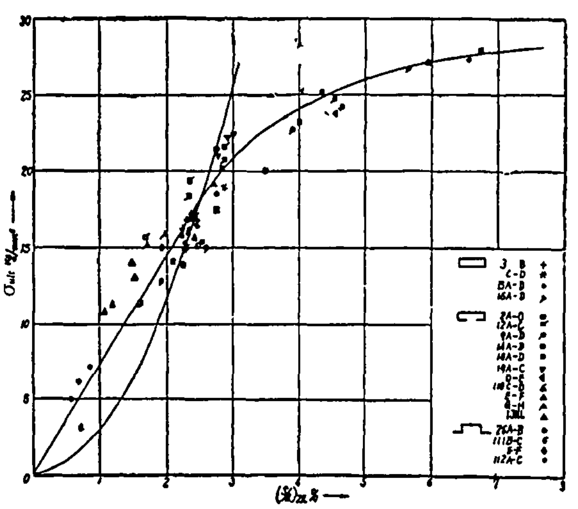


Figure 3. Relationship between average stress and thickness-to-width ratio (duralmin).

As a reference, the relationship in the case of Duralmin is given in Figure 3. In the graph, the area where the relationship is linear almost corresponds to that where buckling occurs under the proportional limit. In this area, equation (12.) is applicable. If the value exceeds the linear area and goes into the non-linear area, the relationship is no longer proportional. This is because plastic buckling occurs due to the increased thickness-to-width ratio.

With this review, the following can be predicted.
① A similar tendency may be seen with the SPCC material. As shown in Figure 4, average stress linearly increases until Point A. After exceeding that point, the increase becomes slight . From a practical point of view, the effect of an increased plate thickness may become less significant if the thickness-to-width ratio is set above Point A when designing structural components that feature better energy absorbing characteristics.

Although the proportional limit of the SPCC material is 1.5%, according to equation (5.), we should not blindly adopt the value without reflecting other parameters such as weight-to-performance efficiency.

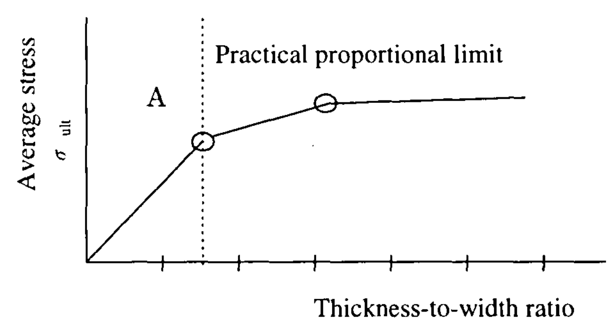


Figure 4. Thickness-to-width ratio limit.

② Since the component has a rectangular cross section, the side walls are under higher stress according to Kármán's effective width. The peak force becomes larger with an increased number of corner lines. We should be mindful, however, that corner lines should be added to the area where the addition is most effective while considering the thickness-to-width ratio.

THICKNESS-TO-WIDTH RATIO APPLICATION TO RECTANGULAR SECTION MATERIAL

Automobile side members are required to effectively absorb energy derived from a collision to ensure sufficient occupant safety. Although a hexagonal cross section materials are deemed applicable because of their high energy-absorption characteristics, but they are not practical in an automobile structure. As a result, rectangular cross section materials have traditionally been adopted.

It is assumable here that the peak force of the rectangular-sectional member corresponds to the sum of the maximum load of each rectangular plate component, and is defined by equation (12.).

$$P_{cr} = \sum_{1=l}^{n} P_{ult\cdot i} \qquad (14.)$$

Where,

P_{cr}: Static peak force of member

$P_{ult\cdot i}$: Maximum load of i-th component

Based upon the equation given above, a rectangular-sectional member may ensure a similar level of energy-absorption performance to that of a hexagonal member provided that the number of corner lines and thickness-to-width ratio are carefully determined.

We further conducted static compression tests to prove our hypothesis. Extensive computer simulations were also performed to predict the practical limit of the thickness-to-width ratio.

STATIC COMPRESSION TEST

Test Piece Configuration

The configuration and major dimensions of the test piece are shown in Figure 5. The length and thickness of the test pieces were fixed at 300mm and 1.2mm, respectively. At the upper and lower ends of the side wall surface, smaller tapers of 1.5° were incorporated in order to promote compression collapse. At the flange, the test piece was spot-welded; whereas a plate that anchored the piece to the test equipment was arc welded at the rear.

The tests were conducted at a speed of 2mm/s. The test piece was collapsed up to 150mm. We determined that the crush force represents the quotient obtained by dividing the energy-absorption quantity measured until the amount of collapse equaled 150mm by stroke.

Type A features a cross section configuration of the popular frame. Type B is a newly designed cross section that was developed to ensure a similar level of energy-absorption to that of the hexagonal cross section member and is based on Type A.

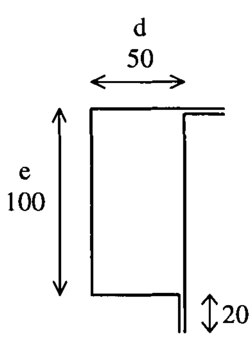
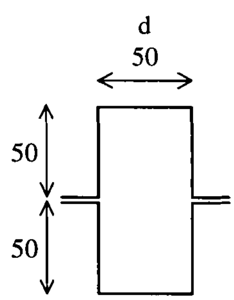
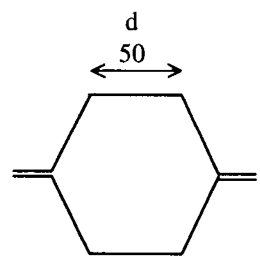
TYPE	A	B	C
Section mm			
Thickness (mm)	1.2	1.2	1.2
Cross section area (mm^2)	456	456	456
Thickness-to-width ratio	1.2 % (t/e)	2.4 % (t/d)	2.4 % (t/d)

Figure 5. Test piece configuration.

The prominent features of Type B are as follows. An additional corner line was incorporated into the center of each side wall to achieve a total of six corner lines. Further, the thickness-to-width ratio of each plate defined by the six lines were designed to be the same. As a result, the collapsing wave length of each plate became equal, showing fairly stable compression collapse characteristics.

Type C was for the evaluation and review of Type B. It shared thickness, cross section area and thickness-to-width ratio with Type B.

Test Results

The loads measured in the static compression test are shown in Table 1. Figure 6 shows the force vs axial stroke of the three test pieces.

Types B and C corresponded to each other in terms of the peak force and crush force. This result proved that a rectangular cross section member was capable of ensuring an energy-absorption level similar to that of a hexagonal member provided that the thickness-to-width ratio is optimized.

Figure 7 shows the crushing mode of the Type B test piece. As a result of the uniform thickness-to-width ratio throughout the rectangular plate, the cross-sectional deflection showed a symmetrical and stable compression collapse.

When compared with the peak force calculated by using Kármán's equation, the results from Type A conformed to the calculated values. With Types B and C, the calculated values were approximately 10% higher than the actual measurements because both of the two pieces had thickness-to-width ratios exceeding the proportional limits.

Table 1.
Static Compression Test Result

Type	Peak force Pcr [kN]	Crush Force $Pm_{(150)}$ [kN]	Average stress σ_{ult} (Mpa)	Crushing mode	Peak force calculated using Kármán's Equation[kN]
A	54.1	24.5	118.6	Non-Sym	52.1
B	68.3	32.4	149.8	Sym	76.0
C	68.7	32.6	150.7	Sym	76.0

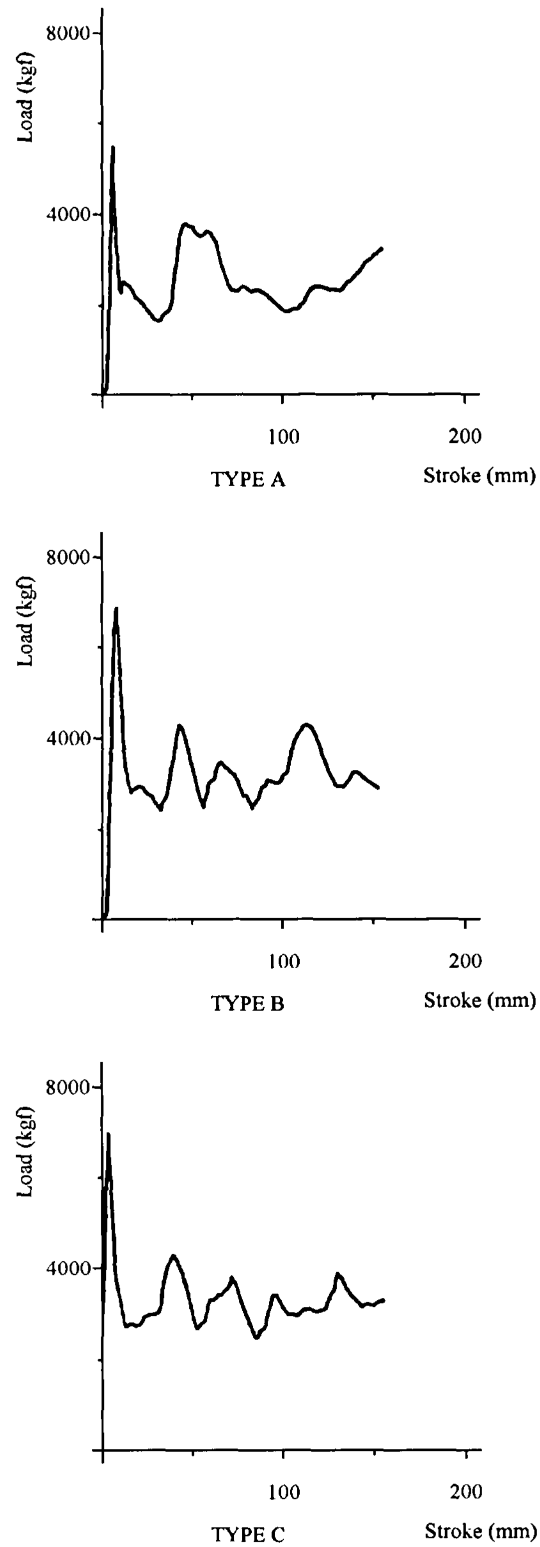


Figure 6. Static compression test result.

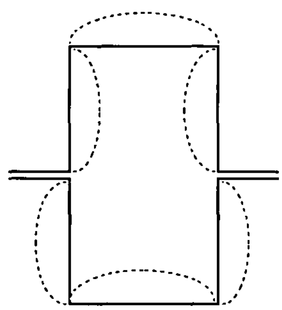


Figure 7. Type B test piece crushing mode.

Concerning the load efficiencies of Types B and A, an increase in the peak force of 26% over the original. The crush force were upgraded by 32%, more than the rate of the peak force. This is because the dominant factor of the loads are different from each other. More specifically, the maximum load depends on the corner-line stiffness of each component whereas the crush force is dominated by side-wall stiffness.

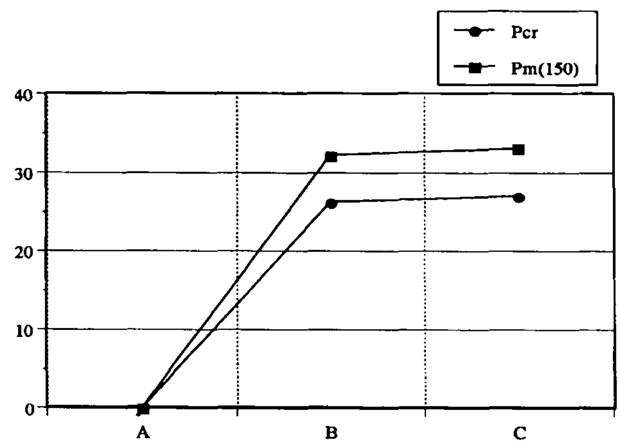


Figure 8. Load increase ratio.

COMPUTER SIMULATION

We obtained the practical limit of the thickness-to-width ratio for Type B through the use of the finite element method.

The simulation results are shown in Figure 9. The calculation results were approximated by a tertiary polynomial curve. The dotted line represents the proportional limit determined by equation (5.). It can be said that the straight line formed between the proportional limit of the proximity curve and origin represents the average stress obtained by Kármán's equation.

As shown in the figure, the increase in the average stress becomes slight when it exceeds the proportional limit of 1.5% and is almost stabilized at around 5%. In other words, after the threshold of 5%, the static maximum load is only proportional to the area even though the thickness is increased.

As a result, the practical limit of the thickness-to-width ratio is 5% from the viewpoint of energy-absorption efficiency per unit weight.

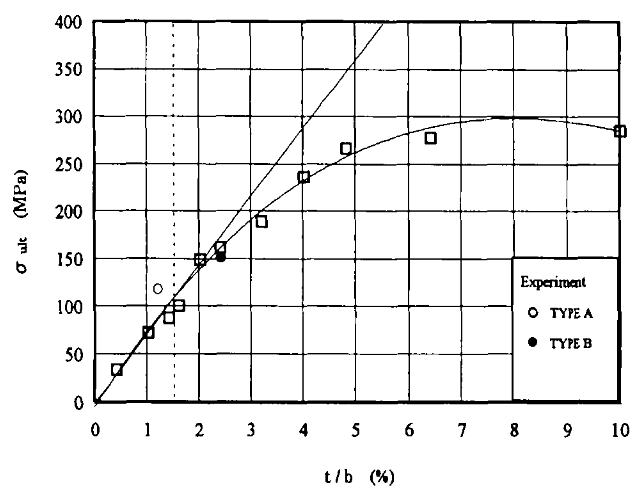


Figure 9. Relationship between average stress and thickness-to-width ratio (SPCC).

CONCLUSION

The investigation of rectangular plate behavior over the proportional limit has disclosed the following:

1) The relationship between average stress and the thickness-to-width ratio is an important design factor. The relationship is expressed by the following equation provided that the proportional limit is not exceeded. After the limit, it becomes non-linear.

$$\sigma_{ult} = \frac{P_{ult}}{t \cdot b} = \frac{\pi\sqrt{E\sigma_y}}{\sqrt{3(1-v^2)}} \cdot \frac{t}{b}$$

2) A rectangular cross section member is capable of ensuring an energy-absorption performance similar to that of a hexagonal cross section member when the

number of corner lines and thickness-to-width ratio are carefully determined.

3) The practical limit of the thickness-to-width ratio can be confirmed by computer simulation.

4) We have come up with the hypothesis that the crush force depends more on the side-wall stiffness than the corner-line stiffness. In future research, one of the areas of our focus is to clarify the hypothesis by using the theory of a thin plate.

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Body Structure of Mitsubishi's Advanced Safety Vehicle

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Paper Number 96-S3-W-20

ABSTRACT

Mitsubishi's Advanced Safety Vehicle (ASV) has been developed in conjunction with the ASV project proposed by Japan's Ministry of Transport. Among the various safety technologies incorporated in the Mitsubishi ASV, this paper focuses specifically on the features of the lighter and safer body structure and the associated development process. Mitsubishi ASV has been developed around two fundamental concepts ; improvement of the crashworthiness of the front end and the integrity of the passenger compartment. The aim was to achieve both a lighter vehicle weight and improved safety performance.

To improve the energy-absorbing capacity of the front end, hybrid side members composed of steel and carbon fiber reinforced plastic (CFRP) were adopted along with a CFRP chassis center member and a CFRP bumper reinforcement beam. The CFRP composition and materials combination were selected on the basis of fundamental component tests. The energy-absorbing performance of the front end structure basis was then confirmed by FEM analysis and in sled crash tests.

An optimization analysis was used to determine the layout of under-floor members to improve the structural integrity of the passenger compartment while minimizing increase in body weight. In addition, the passenger compartment was effectively reinforced by filling the rear portion of the side members with urethane foam selected on the basis of fundamental component tests. The characteristics of the resulting passenger compartment structure were then confirmed through FEM analyses and static crash tests.

Finally, based on the test results for the front end and passenger compartment members, the full body structure was designed and manufactured and its performance was confirmed in dynamic crash tests.

INTRODUCTION

Many different types of studies have been conducted in recent years to improve vehicle safety performance. In Japan, the Ministry of Transport proposed the ASV project with the objective of reducing traffic accidents. This project started in 1991, and each of the participating vehicle manufactures has developed its own advanced safety vehicles. This ASV project has stimulated further research and development work on the safety aspects of the vehicle structure and equipment. The aim of this project is to develop both active safety technologies and passive ones. The Mitsubishi ASV was created as part of this project for the purpose of researching and developing various active safety measures such as a collision avoidance system and a drowsiness warning as well as crash safety technologies for keeping occupant injury to a minimum if an accident should occur.

In order to minimize occupant injury in an accident, the body structure must efficiently absorb crash energy and occupants must be safely protected from secondary collisions in the passenger compartment. One crash safety measure adopted in Mitsubishi ASV is the extensive use of materials besides steel to ensure efficient absorption of crash energy. Another measure taken was to optimize the structure around the passenger compartment to suppress deformation.

This paper describes the development process and features of the body structure of the ASV which incorporate various crash safety measures.

Basic Concept

With respect to the body structure, reducing occupant injury in a collision requires the sufficient crashworthiness and occupant protection in the passenger compartment. These requirements must be met efficiently in order to achieve both the target safety performance and a minimal weight increase. To accomplish these objectives in the ASV, the body structure was developed in two modules, as shown in Figure 1. The energy-absorbing module (denoted as A in the figure) serves to absorbs the crash energy while the integrity module (denoted as B) maintains occupant survival space and receives the reaction force that is generated when the front end absorbs the crash energy. The development targets set for each body structure were to increase the energy-absorbing capacity of the front end by 30% and to reduce passenger compartment deformation by 30%. Trying to accomplish these targets through the use of steel reinforcement alone would have resulted in considerable weight increases. Therefore, it was decided to utilize CFRP and urethane besides steel in order to minimize increase in body weight.

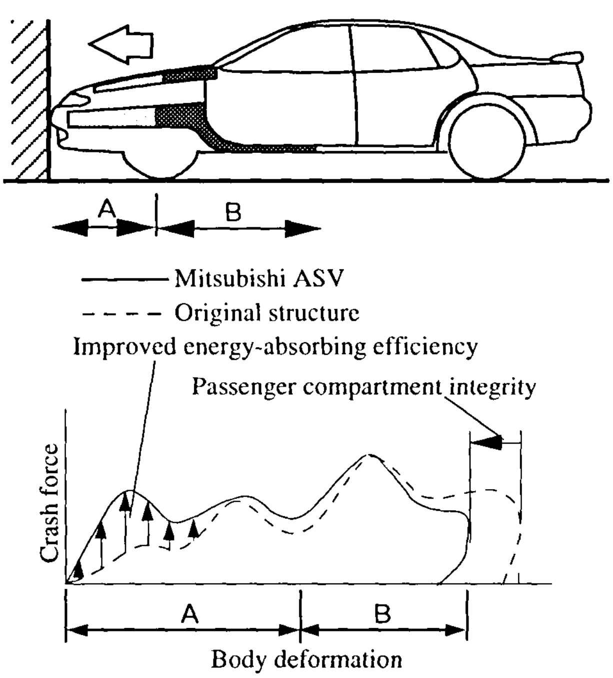


Figure 1.Basic Concept of Mitsubishi ASV

Development Procedure

In order to carry out the development work efficiently, the structures of the energy-absorbing module and the integrity module were examined separately (Fig.2). Fundamental component tests were conducted to examine the materials to be used at each location, and the characteristics of the structures made of the selected materials were confirmed by FEM calculations. Based on the FEM results, prototype components were fabricated and their compliance characteristics were verified. After completing the structural studies of the energy-absorbing and reaction modules, the two modules were combined to create a prototype vehicle, and crash tests were conducted to confirm the crash performance of the body structure.

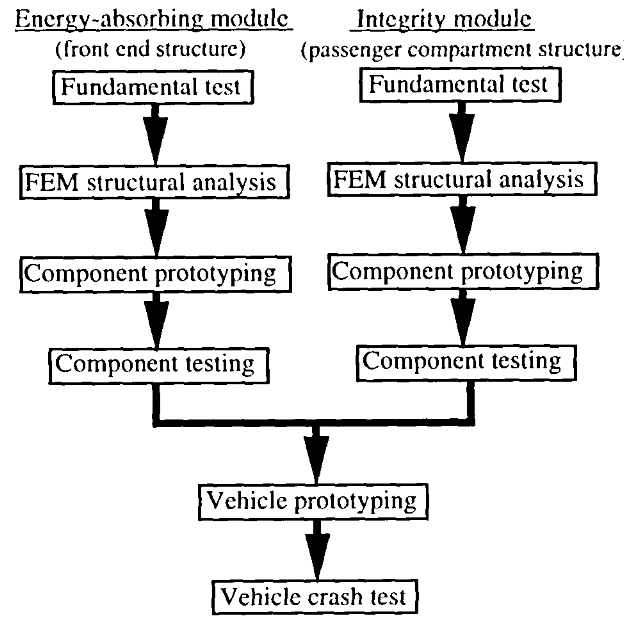


Figure 2. Outline of Development Process

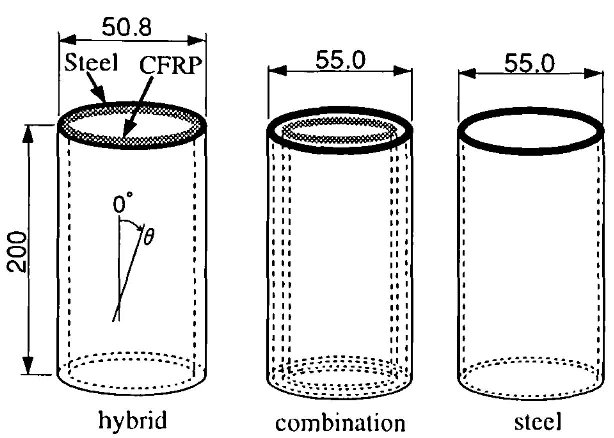
Design of Energy-absorbing Module

In ordinary passenger cars today, the greater proportion (approximately 70%) of the crash energy produced in a frontal collision is absorbed by the structural members of the body. Front side members in particular account for approximately 80% of all the energy absorbed by body structural members [1]. Accordingly, increasing the energy-absorbing efficiency of the front side members is a crucial factor in improving the capacity of the body structure to absorb crash energy. In the ASV , CFRP was bonded to the inside of the steel front side member to form a hybrid structure with the aim of improving energy-absorbing efficiency. In addition, the chassis center member and the upper frame members were also designed to serve as supplemental energy-absorbing members.

Fundamental Test of Energy-absorbing Module

Various fundamental tests were performed in connection with the adoption of CFRP in the energy-absorbing modules. As the first step, axial compressive strength tests were conducted on cylindrical test pieces made of different types of CFRP materials in order to determine the type of fiber and matrix material that should be used[2)3)]. Based on the test results for axial compressive strength and energy absorbing capacity, it was decided to adopt a combination of carbon fiber and epoxy resin as the CFRP materials.

Axial compressive strength tests were conducted on cylindrical test pieces to examine the effect of combining CFRP and steel. The types of test pieces used in the tests are shown in the Figure 3. The hybrid specimen on the left side had CFRP bonded to the inside of the steel cylinder. The combination specimen in the center combined the two materials without any bonding. The specimen on the right was made of only steel. Tests were conducted on four types of fiber distribution angles. The test results are shown in Figure 4 and 5. The hybrid specimen showed higher specific compressive strength than the other two types,exceeding the strength of the steel specimen by 1.7-1.3 times and that of the non-bonded combination specimen by 1.1-1.5 times. It also displayed higher specific mass energy absorption that was 1.5-1.7 times greater than that of the steel specimen and 1.1-1.4 times greater than that of the non-bonded combination specimen.



Thickness of CFRP : 1.6mm
Thickness of Steel : 2.5mm

Figure 3.Cylindrical Test Specimens

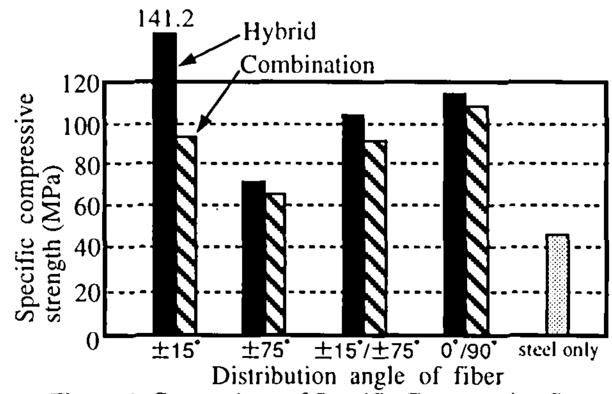


Figure 4. Comparison of Specific Compressive Strength

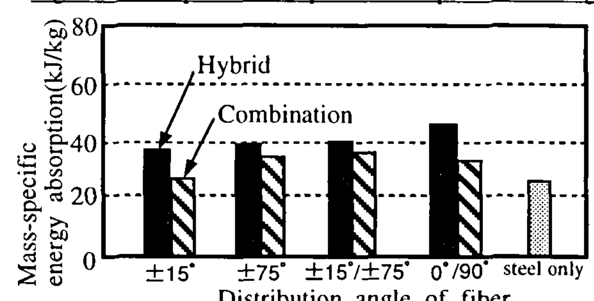


Figure 5. Comparison of Energy-absorbing Capacity

To determine the distribution angle of the carbon fibers used in the CFRP, axial compressive strength tests were conducted on straight frames having a box-shaped cross section, simulating the actual the side members. The test piece shape is shown in Figure 6. A box-shaped steel frame was spot-welded and CFRP was bonded to the inside. Figures 7 and 8 show the test results. Similar to the results seen for the cylindrical test pieces, the best combination of specific compressive strength and mass specific energy absorption was obtained with a fiber distribution angle of 0° /90° or ±15°

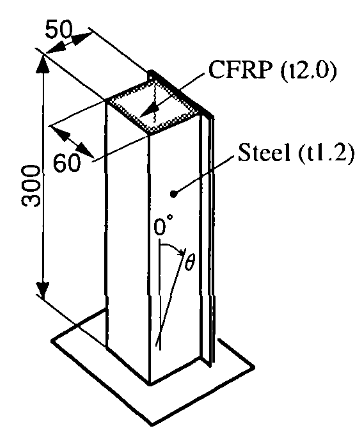


Figure 6.Straight Frame with Box-shaped Section

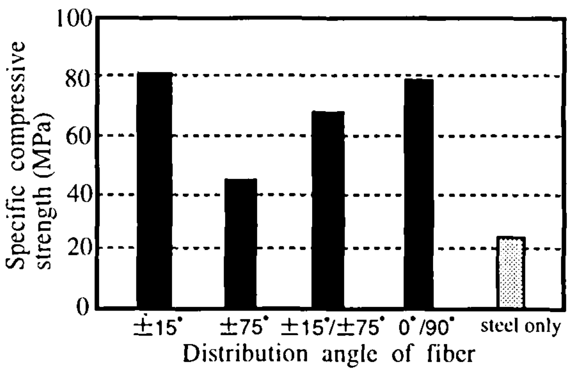


Figure 7. Comparison of Energy-absorbing Capacity

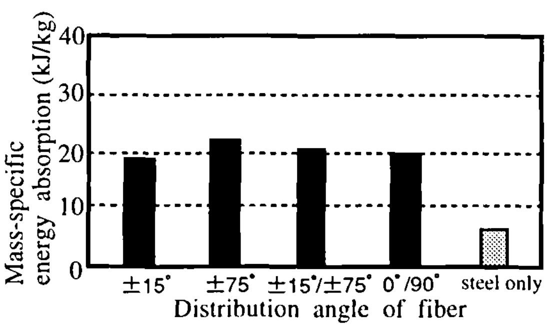
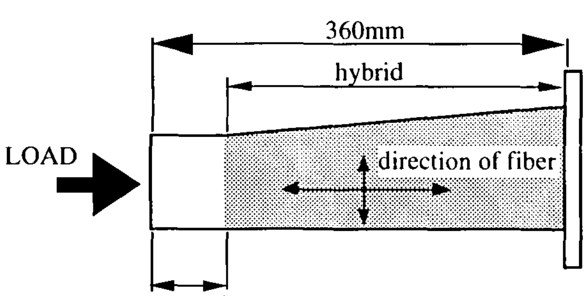


Figure 8. Comparison of Energy-absorbing Capacity

Quasi-static axial compressive strength tests were conducted on the end of the front side member to investigate the effect of the thickness of the CFRP layer. Figure 9 shows the configuration of the test piece, simulating a side member with CFRP bonded on the inside. CFRP was not applied to the front 60mm of the side member as that is where the cross member is normally attached by welding. The relationship between the mean force capacity and the thickness of the CFRP layer is shown in Figure 10. The mean force capacity increases linearly in proportion to the thickness of the CFRP layer.



Thickness of Steel Inner Panel : 1.4mm
Thickness of Steel Outer Panel : 1.2mm
Figure 9. Configuration of Test Specimen

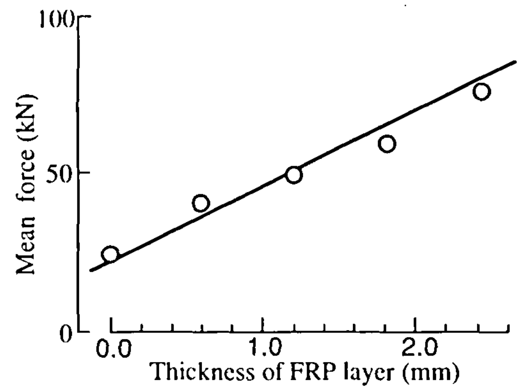


Figure 10. CFRP Thickness vs. Mean Load Capacity

FEM Analyses of Energy-absorbing Module

Effective use of FEM analysis is a key factor in body development. FEM analyses were employed extensively in developing the body structure of the ASV. For instance, FEM analysis was applied to the energy-absorbing module to examine the strength balance between the part where CFRP was bonded and the subsequent portion of the structure. One aspect that presented a problem here was how to model the part where CFRP was bonded.

It is generally quite difficult to analyze a CFRP structure using an FEM model because the structure buckles with accompanying rupture or crack propagation. To avoid that difficulty, the procedure used in the FEM analyses was simplified to replace CFRP with steel having the same buckling force as the plastic material. The relationship between the buckling force of CFRP and steel was determined by using the axial compressive strength test results for the box-shaped straight frame and the side member.

Figure 11 shows the FEM model of the energy-absorbing module. Containing approximately 6,700 elements, this model represents one-half of the front end from the wheel housing forward. The front portion of the side member for approximately 300mm corresponded to the hybrid member. The thickness of that portion was used as a parameter in conducting an analysis of a crash into a rigid barrier.

Figure 12 shows the maximum deceleration and maximum deformation of the body as a function of the thickness of the CFRP layer. Up to a thickness of 2.0mm, the maximum deceleration of the body increased and the amount of deformation decreased. However, above a thickness of 2.0mm, the maximum deceleration remained nearly unchanged while the maximum amount of deformation increased. This is due to the fact that the deformation mode of the side member changed from a folding mode to a bending mode (Fig.13).

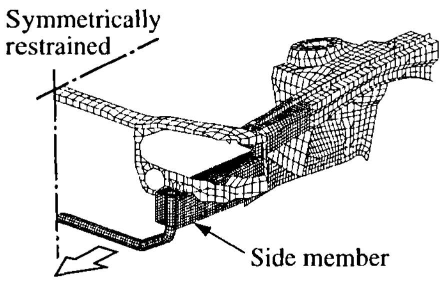


Figure 11. Model of Energy-absorbing Module

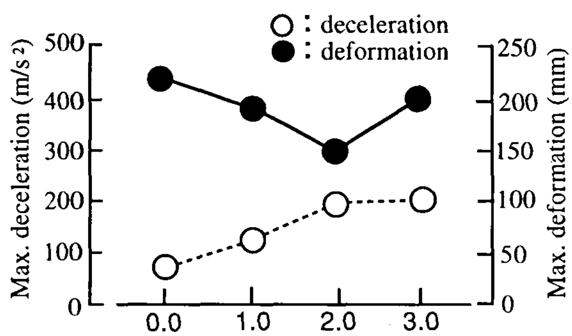


Figure 12. CFRP Thickness vs. Maximum Deceleration and Deformation

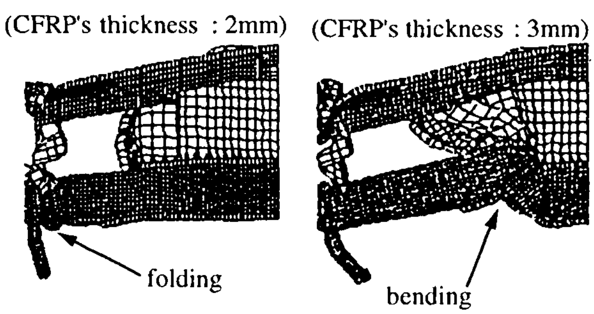


Figure 13. Deformation Mode

Component Test of Energy-absorbing Module

Prototypes of the front end were fabricated, attached to a sled and crashed into a rigid barrier in order to confirm the compliance of the energy-absorbing module[3]. A schematic of the crash test conditions is given in Figure 14.

Figure 15 compares the deceleration vs. deformation characteristics of the sled for a hybrid member and a steel one which have the same structure in order to make a simple comparison of the effect with and without CFRP. A 2mm thick layer of CFRP was bonded to the front portion of the side members for approximately 300mm. The distribution angle of the fibers was 0° /90° . The results indicate that the application of CFRP increased the mean deceleration by approximately 50%, confirming that the hybrid member was effective in raising the energy-absorbing capacity of the front end structure. This test also verified that the hybrid member displayed a folding deformation mode.

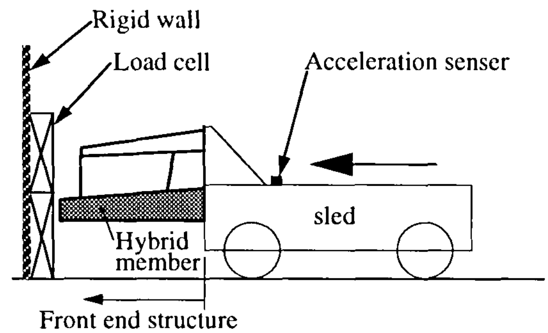


Figure 14. Crash Test Setup for Energy-absorbing Module

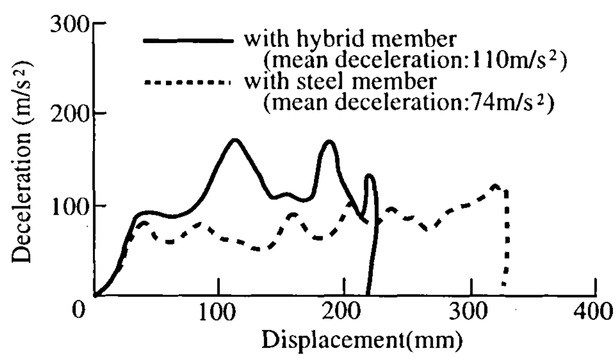


Figure 15. Sled Deceleration vs. Deformation

The effect of the CFRP chassis center member, which was designed to play a supplemental energy-absorbing role, for increasing body strength was also examined. This member was made entirely of CFRP instead of using a hybrid construction, because the complex structure of this member makes it difficult to bond CFRP on the inside and the largest possible weight reduction was desired while bonding CFRP on the outside of the member would have little effect. Quasi-static compressive strength tests were conducted to confirm the effect of using CFRP. An outline of the test procedure is shown in Figure 16. The rear part of the member was restrained and a quasi-static load was applied to the front of the

Figure 17 compares the test results obtained for CFRP and steel specimens. The plate thickness distribution of the CFRP specimen used in this comparison was found by FEM analyses, assuming that the external shapes of the two specimens were the same. The results indicate that the application of CFRP increased the maximum force capacity by approximately 50% and also reduced the member weight by 60%.

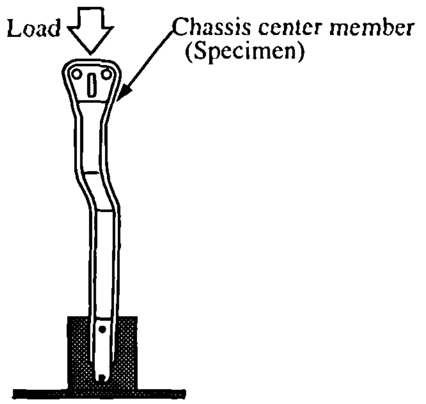


Figure 16. Quasi-static Compressive Strength Test Setup

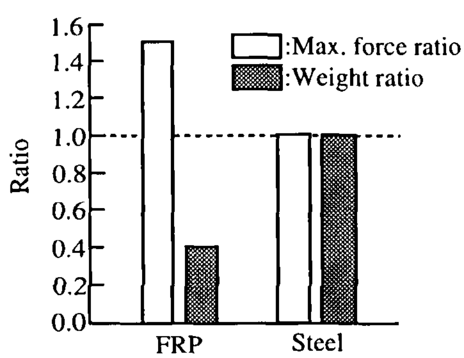


Figure 17. Max. Force and Weight Ratio

Design of Integrity Module

The integrity module must be capable of assuring occupant survival space by fully withstanding the reaction force generated while the energy-absorbing module buckles and absorbs crash energy. To strengthen the integrity module of the ASV, the layout and cross-sectional shape of the floor side member were modified from the rear of the front side member down to the floor. In addition, the floor side member and passenger compartment frame members were filled with urethane foam in order to reinforce their bending stiffness efficiently.

Fundamental Tests of Integrity Module

Most of the frame members of the integrity module generally incorporate bends. Effectively reinforcing these bends is an important factor in augmenting the strength of the integrity module. An investigation was made of a method for strengthening the bends by filling them with urethane foam so as to restrain their cross section.

Figure 18 shows the bent frame specimen that was used to confirm the effect of filling the bends with urethane foam. The specimen, consisting of two bent frames having a box-shaped cross section, was filled with urethane foam and subjected to a quasi-static compressive strength test. Figure 19 shows the measured maximum force capacity as a function of the density of the urethane foam. The results confirmed that filling the frame with urethane foam is an effective way of reinforcing the bends.

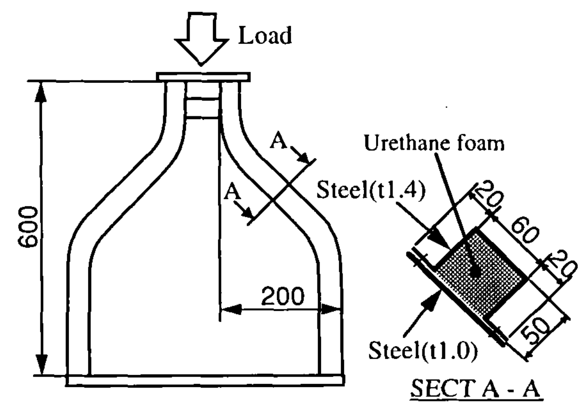


Figure 18. Schematic of Bent Frame Filled with Urethane Foam

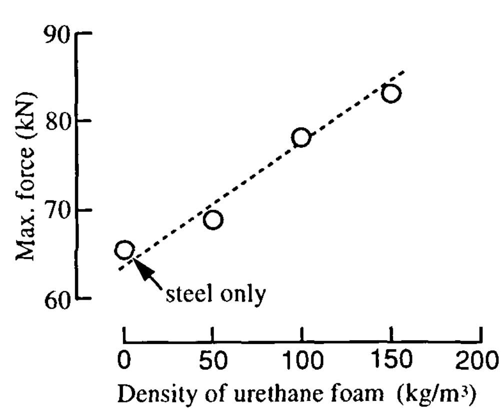


Figure 19. Maximum force Capacity vs. Urethane Foam Desity

FEM Analyses of Integrity Module

FEM analysis was used to examine the frame members of the integrity module, focusing in particular on the floor side members. A sizing optimization method (fully stressed design) was used in order to carry out the structural analysis efficiently [4)5)]. A linear analysis was made of the frame members for the sake of simplifying the calculations and also in view of the fact that the regions to be calculated are ones that experience a small degree of deformation.

The calculations were performed in two stages. Initially, the frame members of interest were modeled using beam elements having a circular cross section (Fig.20). In the calculations, the diameter of the beam elements was employed as a design variable [3)]. The results indicated that the front portion of the floor side members should be increased in size and that the lateral offset of the floor side members should be reduced. Based on the calculation results, the shape of the floor side members was then designed (Fig.21). The determined structure was modeled using shell elements and optimizing calculations were performed using the plate thickness as the design variable (Fig.22) [3)]. On the basis of the results obtained, the plate thickness of the floor side members was then determined.

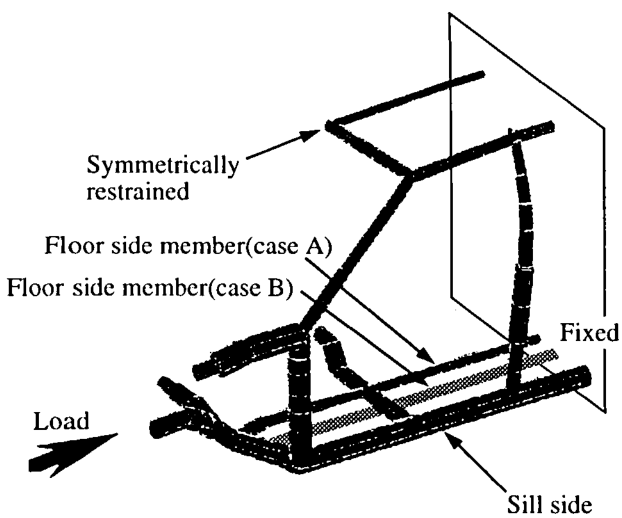


Figure 20. Beam Element Model of Passenger Compartment

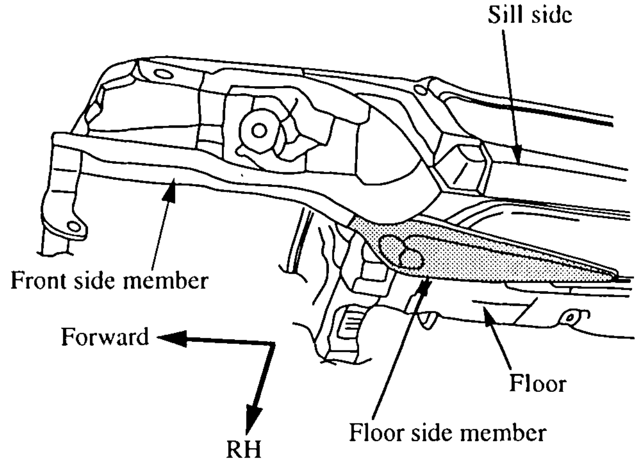


Figure 21. Configuration of Floor Side Member

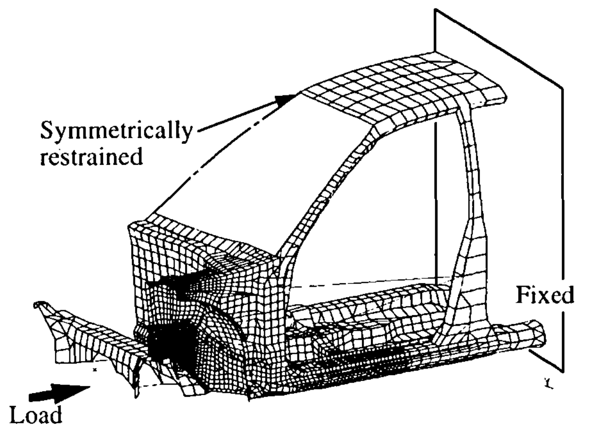


Figure 22. Shell Element Model of Passenger Compartment

Component Test Results for Integrity Module

Passenger compartment prototypes were built and subjected to quasi-static compressive strength tests in order to confirm the strength of the integrity module designed on the basis of the fundamental tests and FEM analyses. Three types of reaction structure were tested. One represented the original structure; in the second, the layout and cross section of the floor side members were modified; and in the third, the modified floor side members were filled with urethane foam ($150 kg/m^3$). Figure 23 shows a schematic of the test conditions, and Figure 24 shows the relationship between the applied force and the resulting deformation of the three body structures.

Figure 24 indicate that modifying the cross section and the layout of the floor side members increases the maximum force capacity by approximately 50% and improves the energy-absorbing capacity by 30%. Filling the floor side members with urethane foam increased the energy-absorbing capacity further by approximately 30%,it is noted that the maximum force capacity didn 't increase because the initial deformation occurred at a part of side member without urethane foam filed .

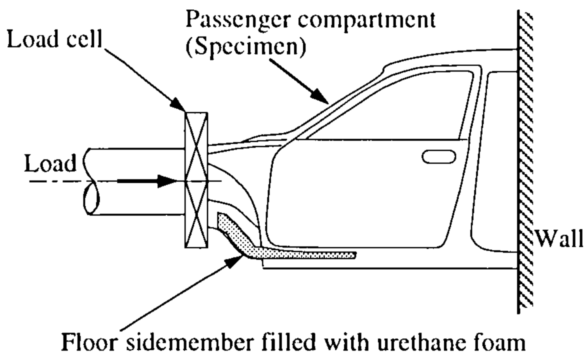


Figure 23.Outline of Quasi-static Compressive Strength Test

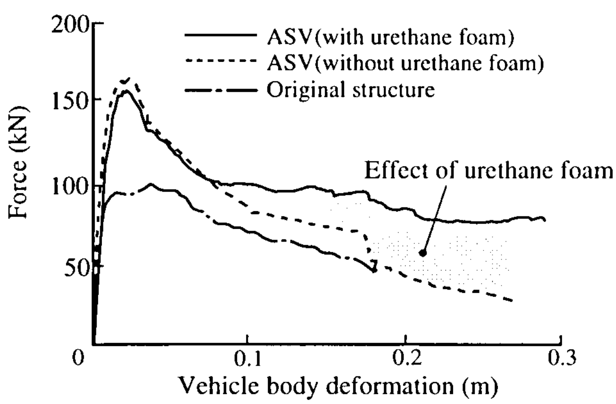


Figure 24. Maximum Load Capacity vs. Vehicle Body Deformation

Design of Overall Body Structure

As the final step, the overall body structure, incorporating the energy-absorbing area and the reaction area which were derived through separate studies, was combined in order to confirm its compliance characteristics. In the process of designing the body structure, its characteristics and deformation modes were confirmed by FEM analysis (Fig.25), and the detailed design of the structure was then determined so as to strike a balance between the energy-absorbing and integrity modules.

The final body structure of the ASV is shown in Figure 26. In the energy-absorbing structure, a CFRP layer 2mm in thickness was bonded to the inside of the side members. The upper frame members and the bumper reinforcement beam, which function as supplemental energy-absorbing members, were both made of CFRP. In the integrity module, the floor side members were designed with a larger cross section and were filled with urethane foam(150kg/m[3]). The bends of the side frame members were also filled with urethane foam and the inner door panel was reinforced.

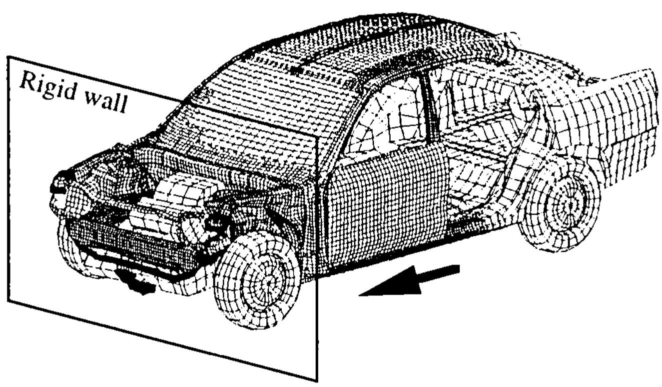


Figure 25. Displacement Modes Found By FEM Analysis

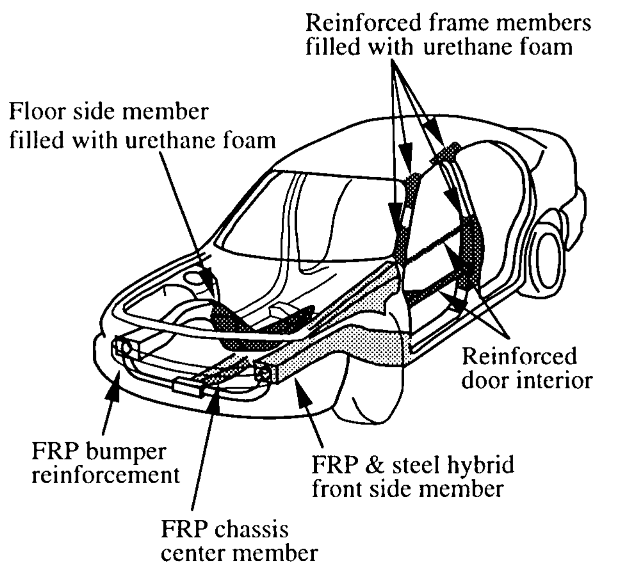


Figure 26. Body Structure of ASV

Vehicle Crash Tests

The ASV prototype was subjected to crash tests to confirm its characteristics. The results indicated that its energy-absorbing capacity was approximately 30% greater than that of the original vehicle structure and the deformation of its overall length was reduced by approximately 10%.

The amount of permanent deformation respected various components of the passenger compartment structure was measured with a 3-d measuring instrument (Fig.27). It was confirmed that the fire wall and front pillars of the ASV showed approximately 30% less deformation than their counterparts on the original vehicle.

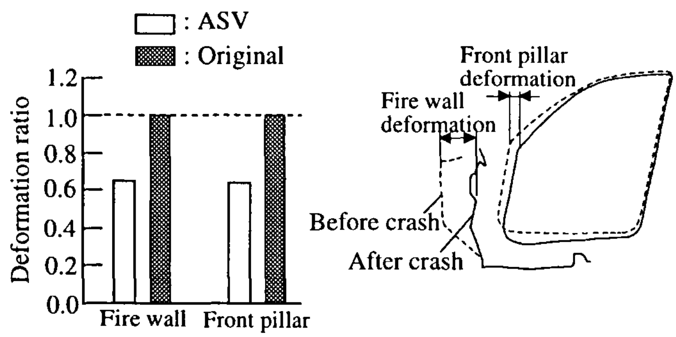


Figure 27. Comparison of Body Part Deformation

Conclusion

This paper has described the FEM analyses and fundamental component tests that were conducted in the course of developing the body structure of Mitsubishi ASV. The following conclusions can be drawn from the results obtained so far.

(1) The hybrid member that was created by bonding CFRP to a steel side member produces a high buckling force capacity and a stable folding deformation mode. These characteristics make it an effective member for absorbing crash energy efficiently.

(2) Effective reinforcement of the integrity module requires optimization of the crash force load capacity based on studies of the layout and cross-sectional shape of the various frame members. In this research, an optimization method based on FEM analyses were used to optimize the layout and cross-sectional shape of the floor side members.

(3) Filling frame members with urethane foam to restrain their cross section was shown to be an effective method of reinforcing the bends of such members in the integrity module.

(4) The determined body structure of the ASV provides improved crashworthiness with reasonably minimal weight increase.

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FRONTAL COLLISION MITIGATION USING INTELLIGENT EXTENDING BUMPER

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Paper Number 96-S3-W-21

ABSTRACT

This paper investigates feasibility and performance of collision mitigation using an intelligent extending bumper system to absorb impact energy in vehicle frontal collisions. It is proposed to use a 1-1.5 m extending bumper supported by two hydraulic cylinders fixed to the longitudinal columns of the vehicle. A specially adapted fast response hydraulic flow control valve with a learning digital computer control is proposed. Simulation investigations revealed a maximum level of 9g deceleration could be achieved in a 30 mph collision

INTRODUCTION

Car occupants account for almost half of the fatally and seriously injured road casualties and two thirds of these are injured in frontal collisions. Car occupants are considered the most prominent group in traffic accidents involving injuries or fatalities. Car to car or single car accidents make 71% of all car accidents with fatal injuries [3]. For car occupants, frontal impacts are still the major cause of severe or fatal injury.

Occupant protection was confined in the early stages to the structural integrity of the occupant compartment in frontal crashes. The sixties have witnessed progress in Crashworthiness aiming at preserving the occupant's survival space by employing the concept of energy management through front structure. Developments in this field within the last two decades have achieved significant progress in optimising vehicle frontal structure to protect occupant's survival space and, at the same time reducing impact acceleration on the occupant.

The problem of occupant's safety in vehicle frontal collision has been traditionally addressed by optimising energy absorbing front structures, occupants' restraints legislation, and introduction of front seat air bag. The worldwide uptrend in consumers' interest and legislators' involvement in vehicle safety have injected new momentum in expanding the limits of today's technology. Surveys indicate that consumers consider vehicle safety an important factor in their buying decision. Offset frontal impact protection and dynamic side impact protection are currently Europe's top two regulatory priorities. Next generation of safe vehicles features Collision Avoidance system using accident sensing radar, and "tailored" frontal and side airbags.

Worldwide research programmes have been pursued in the last decade to develop "Intelligent Vehicle safety Systems". IVHS (US Intelligent Vehicle Highway System), PROMETHEUS (EU Programme for European Traffic with Highest Efficiency and Unprecedented Safety), and VICS (Japan Vehicle Information and Communication System). The US IVHS strategic plan is to reduce the fatality and injury rate by 8% by the year 2011. Collision accidents would, therefore, inevitably occupy high priority in traffic accidents no matter how advanced collision avoidance technology is. This research aims at complimenting Collision Avoidance programmes by attempting to protect occupants when collision does occur. This programme is considered to be an aid to crashworthiness by mitigating impact and reduce energy absorbed by the collapsing car body incase of severe impact.

INTELLIGENT BUMPER CONCEPT

The passive safety criteria is thus to

1. Reduce deceleration level throughout the period of collision. Soft plastic deformation of the car frontal structure.
2. Increase energy absorbing capability of the car front. Hard plastic deformation of the car frontal structure and longer deformation length.

The limit of passive safety clearly lies in the length of the frontal structure and the maximum structure force throughout plastic deformation. The maximum structure force is usually limited by the maximum deceleration level the human torso can sustain. Ideally it is required to make the structure softer and double the length of the car frontal collapsing structure. This is what the intelligent bumper system is trying to do.

The concept of the Intelligent Bumper stems from limitation of the energy absorbing capabilities of the frontal structure. It is based on the idea of simulating artificial deformation by providing longer crush zone. A longer crush zone can decrease both intrusion and compartment deceleration level, thus improving safety at wider impact speed range [8]. Two hydraulic cylinders is proposed to support the bumper as shown in Fig. 1.

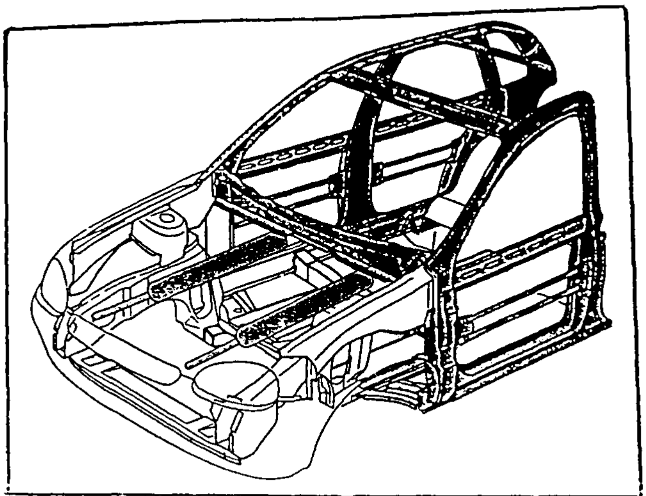

Figure 1. Conceptual fitting of two hydraulic cylinders and the bumper to a small vehicle

This paper attempts to apply active crash control to secondary safety in vehicle frontal collisions. The operational concept of the intelligent bumper system is based on prediction of collision by means of a radar and possibly other acceleration sensors. Once the control systems detect a collision is immanent, the two hydraulic cylinders holding the bumper at the front are controlled accordingly to timely extend the bumper to its full length so that it reaches its full stroke ideally immediately prior to collision first contact.

The function of the intelligent bumper is to generate a controlled collision force at a level high enough to decelerate the vehicle to a stop within constraints of the bumper travel distance while keeping it below plastic deformation thresholds of the body shell. This will ensure a safe deceleration level to protect the human torso as well as the car body.

OPERATION

The operation of the bumper in a crash scenario consists of two phases. The first phase is to extend the bumper under no load prior to collision. The second phase of operation is the collision phase when the bumper is pushed back under the collision load to its retracted position. Extending the bumper must be very fast to avoid having to extend it earlier, and thus risking the possibility of false alarm incase of no collision taking place.

The second phase of operation starts on contact between the intelligent bumper and the other vehicle, or object. It is assumed that the valve directing the flow into the cylinder prior to collision would be closed by the time collision contact takes place. A flow control valve would be open using energy of the impact pushing the cylinder back into retracted position, and forcing the flow back to drain through the control valve.

Cylinder pressure, and thus collision force, could be controlled at a specified level during cylinder retraction. This level of collision force would determine the rate of deceleration of the car and hence the rate of collision energy dissipation. The maximum collision force that can be tolerated is limited by the strength of the longitudinal support of the vehicle where the two hydraulic cylinders are fixed. The actual collision force that the control system would maintain is determined by the level of deceleration pulse required to protect the human torso, constrained by limits on the maximum practical stroke length of the cylinders. The latter was found to be most significant in determining the collision force level.

SYSTEM LIMITATION

A practical theoretical limit of 1.5m length means system capability of completely absorbing impact energy up to about 35mph. A one meter length is capable of completely absorbing impact energy up to 30 mph at a 9g deceleration level. The one meter extended length is more likely proposal for a practical implementation of the concept.

The proposed system is only effective within the following constraints:

1. Operative in frontal collisions only
2. Effective in partial overlap of 40 -100%
3. Effective in oblique frontal collision of angle < 30^{o}
4. Partially effective in collision speed range 30-50 mph
5. Pedestrian safety considerations
6. Weight and space limitation.

European data [3] suggest that frontal impacts account for between 40% and 66% of impacts causing severe or fatal injuries. As shown in Fig. 3, partial overlap and oblique frontal collisions account to over 70% of head-on accidents with occupant fatality and over 35% of head-on accidents with occupant injury. Thus partial overlap and oblique frontal collisions continue to be major contributor to fatal and severe injuries.

German statistical data [3] shows that over 80% of head-on collisions with serious injury or fatality occur with impact speed range 20 - 40 mph. It is therefore claimed that improving safety within 40mph speed range would have significant impact on reducing fatality or serious injury rate in frontal collisions.

Partial and oblique impacts impose serious problem to the integrity of the bumper by exerting asymmetrical loading on the longitudinal supports to which the hydraulic cylinder is attached. A special mechanism design is recommended to distribute and transmit the load evenly at the bumper side as well as at the car structure loading.

It is proposed to use two cylinders freely hinged at the bumper front end to allow for asymmetrical retraction of the two cylinders during collision. Two flow control valves are proposed to allow cross-cylinders flow in case of asymmetrical loading on the cylinders.

Problems of pedestrian safety can be addressed by softening the front end of the bumper with foam padding material and possibly employing a radar sensor distinguishing pedestrians from objects.

Total weight of a prototype system consisting of a bumper, accumulator(s), flow control valve(s), two cylinders and piping is estimated to range from 150 kg to 200 kg which is approximately 10-15% of the total weight of the car. A lower figure is expected for a production version of the equipment.

Space availability to fit a prototype equipment to an ordinary middle size car is an arduous task to achieve without interfering with the car controls. Two 80 mm bore cylinders must be firmly fixed to the longitudinal columns of the car. Details of such arrangement must be tailored to the particular car under consideration. Some modification to the car may be necessary to fit the prototype equipment.

Hydraulic Power

Power requirement is mainly to extend the bumper under no load in the pre-collision mode. Throughout the collision mode power is dissipated in the hydraulic system amounting to the full impact energy. A minimum of 1 m/s bumper extend speed is required to avoid false alarm would imply over 500 l/m flow into the cylinders during the pre-collision extend mode assuming two cylinders of 80mm diameter. Pressure required during extend mode is only to offset any backup pressure due to losses in the pipes and cylinder. The practical choice to accommodate such a high flow is an accumulator system where hydraulic fluid is stored under pressure and released to provide high flow under high pressure drop.

The hydraulic flow in the collision mode would vary with time starting from maximum determined by the collision speed and ending with zero flow at the end of collision. For a 40mph collision the maximum flow forced through each cylinder is over 5000 l/m. Piping problems must be addressed to accommodate this level of high flow. The collision energy absorbed during the collision mode is dissipated in the hydraulic system in the form of heat. An estimate of temperature rise due to even dissipation absorbed by all the swept hydraulic fluid would produce about 12.5 °C temperature rise. A 25% localised dissipation would produce about 50 °C localised temperature rise.

COLLISION TRAJECTORIES

Operation of the system depends mainly on collision prediction. Collision certainty depends is predicted by monitoring basically relative velocity and range. Relative acceleration and absolute acceleration are also necessary information to predict the probability of collision more accurately. Once a combination of relative velocity and range is reached, collision is said to be certain and immanent, and activation of the bumper must be immediate if not already taken place. The decision to activate the bumper needs to process all available information throughout the collision path of the vehicle. A model of collision of a controlled vehicle against another monitored vehicle assumed to be moving with constant speed V_m is presented here:

Starting with initial positive relative speed V_i (controlled vehicle faster than monitored vehicle), and initial range R_i. Assuming constant controlled car deceleration rate a_c. Maximum braking deceleration is a_{max}.

Let V and R be the instantaneous relative speed and range respectively.

$$V = V_c - V_m$$
$$dV/dt = dV_c/dt = a_c$$
$$dR/dt = -V = -(a_c.t + V_i) \qquad (1)$$
$$R = -1/2.a_c.t^2 - V_i.t + R_i$$
Substitute t from (1)
$$R = -1/2.a_c.\{(V-V_i)/a_c\}^2 - V_i.(V-V_i)/a_c + R_i$$
$$R= -1/2.(V+V_i).(V-V_i)/a_c + R_i$$

Assuming minimum impact speed involving bumper activation is V_o (assumed to be 3 mph) when R =1.0 m (assumed bumper stroke).

$$R_i= 1/2.(V_o+V_i).(V_o-V_i)/a_c+1.0 \qquad (2)$$

Equation (2) describes the path of speed V_i versus range R_i for a given deceleration a_c. The critical path of V_i versus R_i is determined by a_{max} giving minimum range R_i. To minimise R_i, for a given V_i and V_o in equation (2), deceleration $|a_c|$ must be maximum. Any speed-range combination requiring more than a_{max} to decelerate to zero range is considered to be definite crash zone as shown in Figure 2. Assuming V_o = 3.0 mph, minimum impact speed, Figure 2 shows two definite collision paths for high and low deceleration levels (depending on road condition).

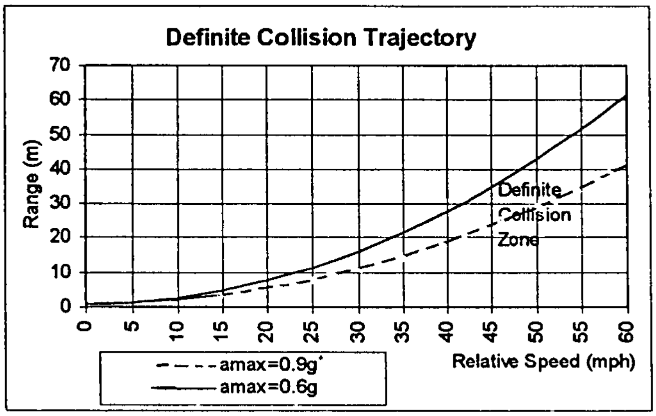

Figure 2. Definite Collision Path of Range versus speed for two levels of deceleration.

Collision is certain once driving path crosses the definite collision curve of Figure 2 into definite collision zone. Similar curve may be obtained for collision of two moving vehicles in which case measurements of absolute speed and acceleration is necessary for accurate prediction of collision.

Collision may be predicted before entering the definite collision zone. If the driving path is one with acceleration rather than deceleration then probable collision needs to be predicted. This is an area of common interest with Collision Avoidance Programme. Prediction zone of a combination of relative speed and range is mainly based on acceleration of the controlled vehicle or relative acceleration in case of two accelerating vehicles. The bounds of collision prediction zone or critical zone as called here is constrained by the time to impact assuming continued acceleration throughout the collision path.

Figure 3. shows the bounds of critical trajectory on the Relative velocity-Range map where collision prediction is critical and possible activation of the bumper may be required. This trajectory is based on assumption of early activation of the bumper so as to be fully extended upon the moment of impact in case of accelerating collision path when time to impact becomes shorter. Detailed analysis of the derivation of the Critical Collision Trajectory is given in Appendix 1.

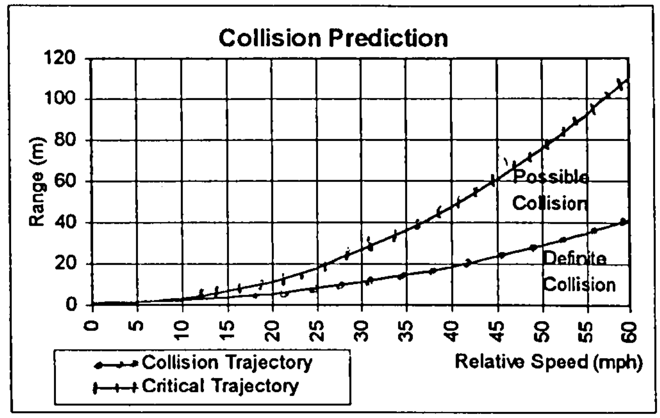

Figure 3. Bounds of collision prediction zone

COLLISION PREDICTION

Collision can only take place when controlled vehicle is faster than monitored vehicle. Collision prediction is therefore relevant when the relative velocity is positive, ie decreasing range. There are three possible collision paths to cross the critical collision curve in Figure 3.

1. Collision path with acceleration
2. Collision path with constant relative speed
3. collision path with deceleration

Figure 4. shows typical collision path starting on the left hand side of the Range-Relative speed map where relative speed is negative, ie range increasing due to controlled vehicle travelling slower than monitored vehicle. The controlled vehicle would accelerate to make relative velocity positive, thus crossing to the right hand side of the Range-Relative speed map. On entering the critical zone two likely collision courses are considered.

1. Continue acceleration of the control vehicle, producing higher collision speed on impact (R= 0).
2. Take braking action and decelerate control vehicle, producing lower collision speed on impact.

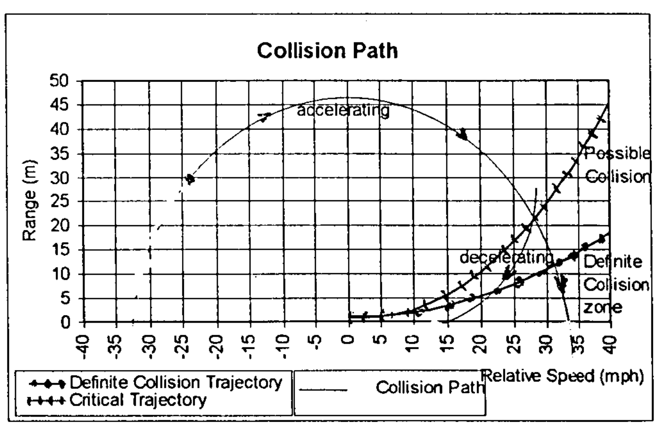

Figure 4. Possible collision path.

Note that Range always increasing in the left hand side and decreasing in right hand side.

The significance of acceleration or deceleration in the trajectory of a collision path is the time it takes to reach crash. This is a crucial parameter in pre-collision control which is required to extend the bumper to its full stroke prior to impact, and to avoid false alarms of extending the bumper without any collision actually takes place.

TIME TO IMPACT

Pre-collision control of the timing of extending the bumper is primarily related to predicting likelihood of impact as well as estimating time to impact in order to activate and extend the bumper to its full stroke before the moment of crash. To achieve this the time taken for the bumper to extend to its full stroke must be less than the time to impact. The time to impact t_i is estimated as follows

$$dR/dt = -V$$
$$t_i = -\int dR/V$$

Where V is the trajectory of the path on the Range-Relative speed map as shown in Figure 4.

Any trajectory to the right of the map should take smaller time to impact compared with one to the left of it. If the bumper is activated upon entering the 'Definite Collision zone' and the driver does not take braking action then the vehicle would reach impact before the bumper is fully extended.

The time to impact, in a decelerating case, is constrained by maximum braking deceleration whilst the latter depends on possible corrective action of the driver or indeed from the controller in case of a Collision Avoidance System. The time to impact has been computed versus relative speed and shown in Figure 5. A constant acceleration from the critical trajectory or constant deceleration from the definite collision trajectory has been assumed.

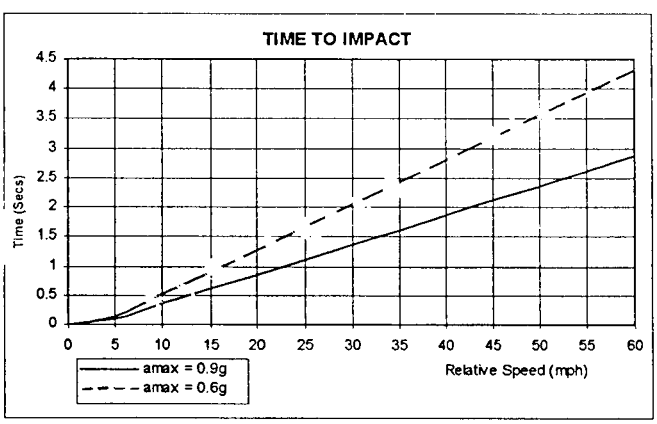

Figure 5. Time to impact based upon maximum deceleration on the definite collision trajectory, and maximum acceleration on the critical trajectory.

Figure 5. shows clearly that the extend time of the bumper must be less than 0.1 second for the bumper to be fully extended under extreme conditions of low speed and low level of maximum deceleration. This is not a practical suggestion to implement. Partial extension of the bumper at low impact speeds must be allowed to trdeoff for slower extension rate of the bumper. A one second extend time of the bumper would allow full extension of the bumper at impact speeds higher than 25 mph under 0.9g deceleration, or 15 mph under 0.6g deceleration.

PRE-COLLISION CONTROL

Three criteria considered in formulating pre-collision control strategy are:

1. The likelihood of a collision taking place. Whether or not the critical trajectory has been crossed and the level of relative acceleration.
2. the predicted time to impact based on projected relative acceleration and the set time of extending the bumper to its full stroke.

Whenever the time to impact as given in Figure 5, is less than the set time of extending the bumper (assumed to be 1 second here), earlier extension of the bumper is necessary to in sure a maximum of one second time to impact. The range-relative speed relationships must be drawn on the pre-collision control map. Since the real time to impact before crossing the definite collision trajectory would depend on the relative acceleration status, a maximum and minimum time to impact exists for each point on the range-relative speed map. Figure 6. shows the trajectories of both maximum and minimum time to impact based on maximum relative deceleration or acceleration.

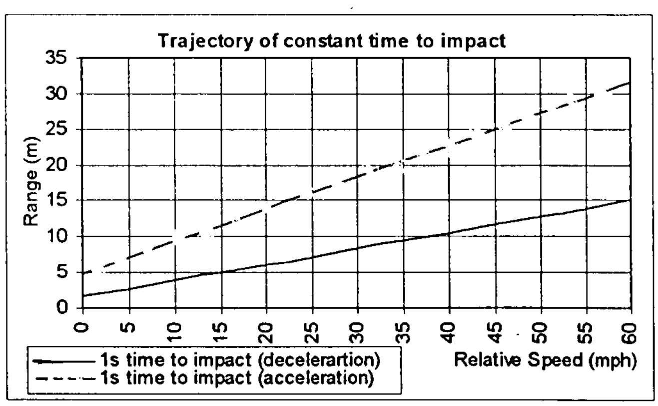

Figure 6. Trajectory of Range versus Relative speed of 1 second time to impact; decelerating path represent maximum time to impact; accelerating path represent minimum time to impact

The two criteria of devising a pre-collision control strategy mentioned earlier are best demonstrated in figure 3 (criterion 1), and figure 6. (criterion 2.). To consider both criteria at the same time it is best to combine both figure 3. and figure 6. as in figure 7. Figure 7. is drawn with a range of relative speed up to 50 as the system is considered to be in effective for impact speeds more than 50 mph.

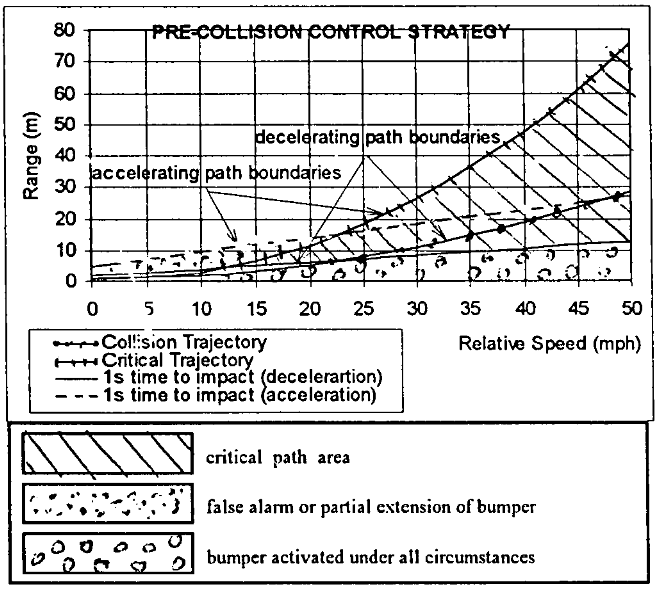

Figure 7. Three areas of pre-collision control strategies

Three control areas can clearly be identified from figure 8.

1. Critical zone where collision is likely based on three possibilities:
   (i) Non-critical zone where maximum time to impact is less than one second. Bumper is only activated in this area in case of extreme accelerating path.
   (ii) Critical zone where impact is likely based on non-maximum deceleration. Bumper activation in this area depends on level of acceleration or deceleration.
   (iii) Area of definite collision zone where minimum time to impact is greater than one second. Bumper activation may be delayed in this zone depending on possible escape path avoiding collision.

2. Critical zone where minimum time to impact is less than one second. Possible false alarm in case of earlier activation of the bumper, or partial extension of the bumper in case of activation upon crossing the definite collision trajectory.

3. Definite collision zone where minimum time to impact is less than one second. Bumper must be activated in this area under all conditions.

A learning adaptive control system like Neural Networks or Fuzzy logic may be necessary to develop the final control system.

COLLISION CONTROL

The main hydraulic control requirement is some form of a flow control valve required to maintain constant pressure during the collision mode. Directional flow valve is only needed in pre-collision extend mode. The simplest form of a flow control is a constant orifice flow control valve. Performance curves with this kind of valve show wide pressure variation and hence longer bumper travel required. The alternative next choice is to use a pressure relief control valve to regulate orifice area so as to maintain constant pressure in response to flow variations. The required response for such a valve must be within 40 ms in order to keep pressure variation and the expected pressure peak at the beginning of impact within acceptable limits. Such electro-hydraulic control valve is not yet in hand within our present technology.

A specially designed high performance flow control valve is proposed where the orifice size is mechanically and instantly changed using the energy of collision. The variation in orifice size should start from fully open at the beginning of impact to fully closed at the end of impact. the rate of change of the orifice size could then be optimised to maintain constant pressure throughout collision. The performance curves shown show that a linearly varying orifice area at a fixed rate would suffice. This area of hydraulic control constitutes yet another technological challenge to design a controllable valve that responds practically in milliseconds while allowing huge level of flow through it.

SIMULATION

Extensive simulation studies have been carried out on various collision scenarios and various hydraulic control configurations. A range of orifice size control laws have been investigated. An optimum law defined to produce constant cylinder pressure throughout collision is the ultimate goal of a refined system. A typical maximum orifice area of 32 cm^2 linearly varying with time has been taken as a sample of results in this paper. The linear control law is given below:

$$A = 32 - 150 * t$$

Typical simulation curves of vehicle velocity, acceleration and bumper travel for the above range of collision speed are shown in Figs. 8, 9, 10, 11. No response time has been assumed. Mechanically actuated valve is assume making use of impact power to activate the valve.

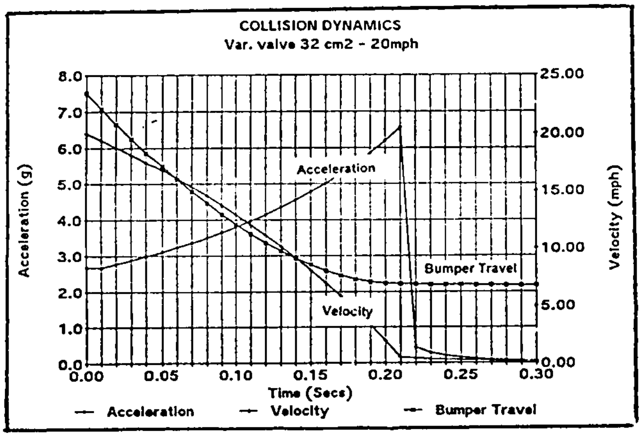

Figure 8. Collision Dynamics-20mph, 1.5m bumper

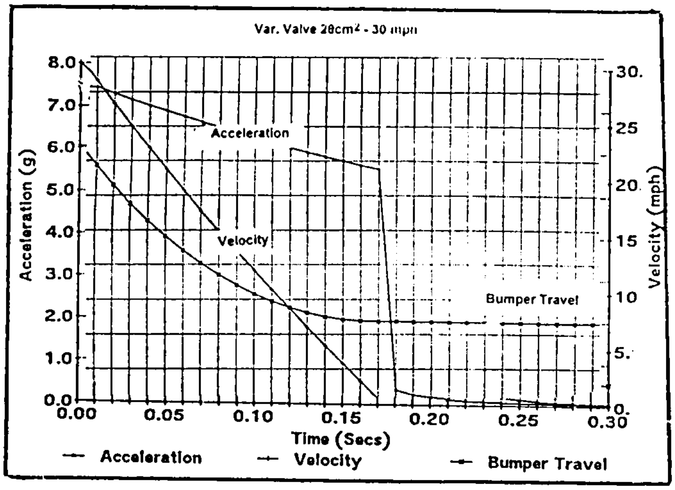

Figure 10. Collision Dynamics-30mph, 1.5m bumper. Alternative flow control

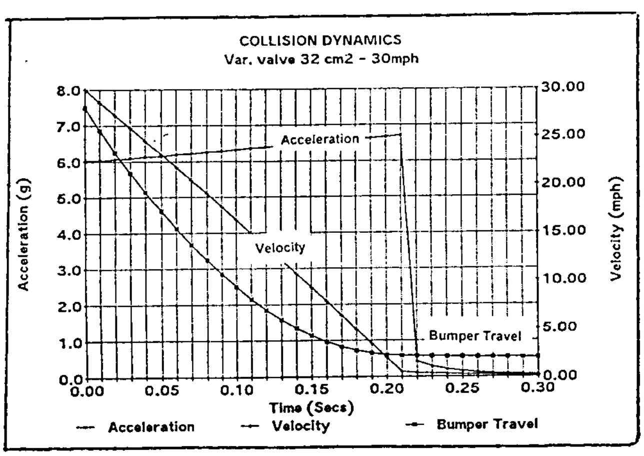

Figure 9. Collision Dynamics-30mph, 1.5m bumper

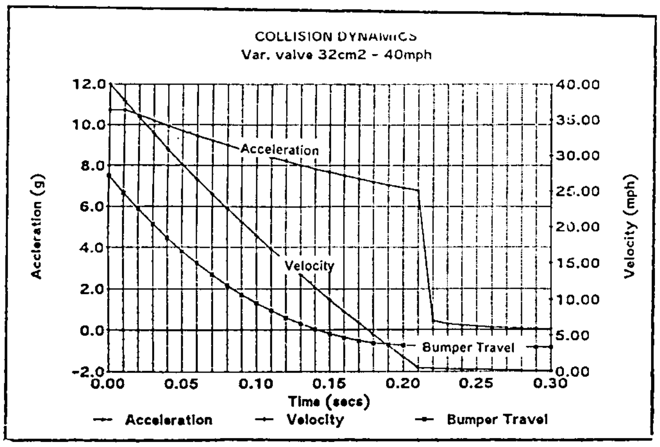

Figure 11. Collision Dynamics-40mph, 1.5m bumper

CONCLUSION

An intelligent bumper system that extends for 1-1.5 meter prior to collision is shown to be capable of absorbing full impact energy at 30mph without any injury to the occupant. The expected g level deceleration for a 20-40 mph collision speed range is given below

For a 20mph collision, deceleration range: 3g - 6.5g.
For a 30mph collision, deceleration range: 6g - 7g.
For a 40mph collision, deceleration range: 11g - 7g.

The main viability problem areas in implementing the system on a prototype car are considered to be:

1. The space requirements and weight of the equipment
2. Provision of the hydraulic power
3. Facilitating high flow-fast response control valves

APPENDIX 1

Let V and R be the instantaneous relative speed and range respectively. Define the critical trajectory to be the path where time to impact assuming maximum acceleration of 0.8g = time to impact on the definite collision trajectory from the same relative speed and assuming maximum deceleration of 0.9g.

$V = V_c - V_m$

$dV/dt = dV_c/dt = a_c$

$dR/dt = -V = -(a_c.t + V_i) \qquad (1)$

$R = -1/2.a_c.t^2 - V_i.t + R_i \qquad (2)$

From (1) time to impact

$t_i = (V_0 - V_i)/0.8g \qquad (3)$

Where V_0 is relative speed at impact with accelerating path

By definition $t_i = (V_o - V_i)/(-0.9g)$

$V_0 = V_i + 0.8/0.9*(V_i - V_o) \qquad (4)$

Substitute t from (3) into (2), $a_c = -0.8g$, R=1

$1=(V_0-V_i)/(0.8g)(-1/2(V_0 - V_i)-V_i)+R_i$

$1=(V_0-V_i)/(0.8g)(-1/2(V_0 + V_i))+R_i$

$R_i=(1/2)(V_0-V_i)/(0.8g)(V_0 + V_i)+1$

Where V_0 is given by (4)

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PARTICLE METHOD FOR AIRBAG DEPLOYMENT SIMULATION

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Paper Number 96-S3-W-22

ABSTRACT

The aim of this paper is to present the use of particle method to evaluate the stress modes of a deploying Airbag, starting from its folded condition.

The innovation is in the gas modelization which is assumed to be made off a large number of particles. As that number is bounded by memory and CPU performance, it is clear that the particles are not identified with the molecules. They are instead abstract building blocks, whose dynamics are chosen to reflect the macroscopic behaviour of the gas.

The natural time scales of the molecular system are of the order of 10^{-8} seconds, the typical time interval between collisions. For obvious reasons, the time step of the simulation should integrate the effects of many molecular events.

The approach consists of replacing the molecular content of the system by a different « microscopic » description, whose coarse-grained space-time behaviour approximates the macroscopic dynamics.

The main idea is to consider the gas as an hybrid system containing a fluid in the classical sense (obeying fluid dynamic equations) and particles in interaction with it. Both elements correspond to the same physical system, but they represent different aspects of it. The fluid represents the part which is in thermal equilibrium, while the particles handle its turbulent aspect.

As the experience with this method is innexistant, it is needed to fine-tune the model in order to build up a predictive capability.

Therefore, it is necessary to proceed by step, here the simulation will be compared to experiment results for a flat unfolded Eurobag simply lining on a flat surface.

DESCRIPTION OF THE PARTICLE THEORY

Introduction

Both the gas mixture and the bag are considered to be composed of a large number of particles. The simulation software can handle an a-priori unlimited number of particles. That number is however bounded by considerations related to the available memory and CPU performance.

It is therefore clear that the particles are not to be identified with the molecules of the gas or the bag. They are instead abstract building blocks, whose dynamics are chosen to reflect the macroscopic behaviour of the system.

The natural time scales of the molecular system are of the order of 10^{-8} seconds, the typical time interval between collisions. For obvious reasons, the time step of the simulation must be significantly longer. Each of these steps should therefore integrate the effects of many molecular events.

Our approach consist of replacing the molecular content of the system by different « microscopic » description, whose coarse-grained space-time behaviour approximates the macroscopic dynamics.

The gas mixture in the bag

Assume that the bag contains a molar fraction r(i) of gas type i:

$$r(CO2)=0.23,\ r(CO)=0.42,\ r(H2)=0.20,\ r(N2)=0.15$$

One mole of gas contains $6.022\ 10^{23}$ (Avogadro's number) of molecules. The mass of one mole is M grams, where M is the molecular weight:

$$M(CO2)=44,\ M(CO)=28,\ M(H2)=2,\ M(N2)=28$$

If N_{moles} is the total number of molecules (in moles), the number of molecules of gas is $r(i)*N_{moles}$. Given a total weight of 8 grams, and the above values of r's, we find: $N_{moles}=0.302$.

Phase space

Define the phase space element $dt=d^3x*d^3v*d\tau_{int}$, where x are the position, v the velocities, and $d\tau_{int}$ correspond to the internal degrees of freedom (such as rotations, vibrations or orbital excitations).The spatial distribution of the molecules is given by:

$$N(x,t) = \int f(x,\Gamma,t)d\Gamma,\ \text{where } d\Gamma = d^3v\ d\tau_{int} \qquad (1.)$$

Ideal gas in the thermal equilibrium

Assume a homogenous gas composed entirely of molecules of type i in thermal equilibrium. The equation of state is PV= NkT, where k is the Boltzmann constant, P the pressure, V the volume, N the number of particles, T the temperature. The velocity distribution of the particles follows the Maxwell-Boltzmann expression:

$$f_0 = n\ \exp(-m/2kT(v-v_0)^2) \qquad (2.)$$

n is the particle density N/V, m the mass of the molecule, v_0 the translational velocity of the gas. The most probable velocity (relative to the overall translational velocity) is $v = v_{mp} = \sqrt{(3kT/m)}$.

The particle system approach

We wish to represent the gas mixture by a system containing as many particles as can be reasonably handled by a computer, say a few millions, while keeping the macroscopic quantities approximately unchanged. There is a good theoretical reason to believe that this can be done. The opposite would imply that by purely macroscopic measurements one would be able to derive the real number of molecules in the system, and therefore infer the atomic scale of mass, which is contrary to physical intuition.

Assume that we replace N molecule by N_p particles, and define $z = N/N_p$. By fixing the total mass, which is a macroscopic quantity, we find mp = zm. Apply this transformation to ideal gas. If we fix P, V, and T, we find that in the particle system world, the Boltzmann constant would be much higher, kp =zk. This is not surprising, since k in the real world is a tiny number when expressed macroscopic units.

Another more fundamental, justification for the above relation, and which defines its limitations and extensions, results from the definition of the entropy: $S = k\log(N_s)$, N_s being the number of microscopic states compatible with the macroscopic quantities. For large N, $\log(N_s)$ is proportional to N, and we confirm the relation kp =zk. Corrections to the simple proportionality rule lead to a systematic expansion of the particle system parameters as a function of z^{-1} . to leading order, we find kp/mp = k/m, and Maxwell-Boltzmann distribution is therefore identical for the gas and the particle system representation.

Particle collisions

The mean free path is the average distance a molecule travels between two collisions, and τ the average time between two collisions. They are given by:

$$\lambda = \frac{1}{4}\sqrt{(\pi/2 * 1/n\,\sigma)}\ , t = \lambda/v \qquad (3.)$$

s is the cross section for inter-molecular collisions, which is of the order of the surface area of the molecule. Typical values for these parameters for light molecules in room temperature are:

$$v = 10^3\ \mathrm{m/s},\ \lambda = 10^{-5}\mathrm{m},\ \tau = 10^{-8}\mathrm{s}$$

As we have seen, a particle system representation would have the same average velocities. We should now deduce the values for the collision parameters for that system, based on macroscopic quantities related to particle collisions. Those quantities are the thermal conductivity and viscosity. Define the energy flux as $q = \int \varepsilon v f\, dG$, where ε is the energy.

Assume that a small volume of gas is moving as a whole with velocity V. We place ourselves in a co-moving frame, O'. The velocities of the individual molecules in the original frame are related by:

$$\varepsilon = \varepsilon' + mV^* v' + 1/2\ mV^2 \qquad (4.)$$

Therefore $q = V(1/2\rho V^2 + W)$, where W is the heat function per unit volume. The thermal conductivity κ is implicitly defined by $q = -\kappa\nabla T$.

The transport equation implies $\kappa \propto kv/\sigma$, which should be equal to the particle system value $\kappa_p \propto k_p v_p/\sigma_p$. We find $\sigma_p = z\sigma$.

Similarly, the kinetic theory determines a value for the viscosity $\eta \propto mv/\sigma$. From the macroscopic constraint $\eta_p = \eta$, we rederive $\sigma_p = z\sigma$.

The mean free path $\lambda p \propto 1/\eta_p\ \sigma_p = 1/n\ \sigma$ and therefore $\lambda_p = \lambda$. Since the average velocities are the same for molecules and the corresponding particles, we conclude $\tau_p = \tau$.

The above results imply that it is possible to represent the gas as a particle system with a relatively small number of particles, but that the natural length and time scales of the problem would be unchanged. The time interval for simulation is 30 ms. With a time step of 10^{-8} seconds, we would have about a million steps, which we cannot achieve in a reasonable computing time. We are therefore forced to sum over individual collisions and develop a time integrated formalism.

Our approach is to consider the gas as a hybrid system containing a fluid in the classical sense (obeying fluid dynamic equations of motion) and particles in interaction with it. Both elements correspond to the physical system (I.E. the gas), but they represent different aspect of it. The fluid represents the part which is in thermal equilibrium, while the particles are not necessarily in such a state. This approach has the advantage of dealing with the turbulent and near-equilibrium aspect of the problem. As our research evolves, we will find-tune our model so that it describe as closely as possible the gas in the bag.

The hybrid approach

Consider each one of the gas in the mixture to be made up of two components:

1. a gas in a thermal equilibrium (the « **background gas** » name here **BG**). Since we will not treat the BG particles individually, we may as well take them to be the constituent molecules.

2. a collection of heavy particles (the « **particle gas** » name here **PG**) interacting often with the BG gas and rarely among them selves. Each one of the heavy particles represents a large number of molecules. The interactions are insufficient for achieving thermal equilibrium for heavy particles. The definition of the

two components is local, so that transitions between the states are allowed.

Assume for simplicity that we are dealing with a homogenous gas (I.E. not a mixture). BG has a tranlational velocity V_{BG} , which may be a function of space and time. It is the average velocity of the molecules in BG. v_{BG} is the velocity of an individual molecule. Let us place ourselves in the rest frame of a heavy particle moving with velocity v_p. The velocity of the molecule in this frame is $v = v_{BG} - v_p$. The velocity distribution of the BG molecules in this frame is given by $f_0(v+v_p-V_{BG})$. Assuming the v_p-V_{BG} is much smaller than v.

$$f_0(v+v_p-V_{BG}) \approx f_0(v)[1-mv*(v_p-V_{BG})/kT] \qquad (5.)$$

Let us consider a collision event between the light and the heavy particle. After the collision, the light particle is deviated by an angle α. The azimuth-averaged momentum transferred to the heavy particle is $mv(1-\cos\alpha)$. To obtain the total momentum transferred to the heavy particle per time unit we multiply by the light particle flux and by collision cross section, the integrate. We find:

$$\delta p_p/\delta t=m\int f_0(v+v_p-V_{BG})v\sigma_t d^3p \qquad (6.)$$

$\sigma_t(p) =\int(1-\cos\alpha)ds/d\Omega * d\Omega$ is the transport cross section. Substituting the expression for $f_0(v+v_p-V_{BG})$ and averaging over the direction of v, we obtain:

$$\delta p_p/\delta t = -n_{BG}\, m^2 /3kT *(v_p-V_{BG})\langle\sigma_t v^3\rangle \qquad (7.)$$

If we assume that BG is an ideal gas, we find:

$$f_{drag} = \delta p_p/\delta t = -P\, m^2 /3kT *(v_p-V_{BG})\langle\sigma_t v^3\rangle \qquad (8.)$$

Where P is the pressure. The change of momentum we have computed acts as a drag force on the heavy particles. As a test, let us compute the effect of representing BG by a particle system, with particles having a heavy mass with respect to the molecules, but light compared to PG constituents.

Since $(n_{BG})_{P'} =z'^{-1}n_{BG}$, $(m^2)_{P'} = z'^2m^2$, and $k_{P'} =z'k$, conclude that the drag force is unchanged as long as st does not depend on z'. This is indeed the case, since we consider for example a hard sphere collision model, the dominant contribution to st comes from the size of the heavy particles. In fact

$$\sigma_t = \tfrac{1}{4}\pi\, (d_{BG} + d_{PG})^2 \approx \tfrac{1}{4}\pi\, d_{BG}^2 \approx z\, \sigma_{BG} \qquad (9.)$$

Define the drag coefficient K implicitly by $f_{drag} = -K(v_p-V_{BG})$. With typical values for an ideal gas, and $N_P \approx 10^5$, we get $K\approx 1$ (Joule * sec / meter²).

Collision in a mixture of gas

Assume that a molecule of a given type is propagating in a mixture of gases. The total probability for a collision in a short time interval is the sum of probabilities for collisions with each of the constituents.

$$\lambda_i = \tfrac{1}{4} \sqrt{(\pi/2)}/\textstyle\sum_j n_j \sigma i_j \qquad (10.)$$

In the hybrid gas model, fdrag is the sum of the forces exerted by the individual constituents. If we assume ideal gases with hard sphere collisions, all sharing the same vBG, we find for the drag coefficient acting on a given type of gas:

$$K_i = zd_i^2/3 \sqrt{(2/\pi)} *(kT)^{1/2} \textstyle\sum_j n_j\, m_j^{1/2} \qquad (11.)$$

To obtain the deceleration coefficient, we divide the deceleration force by the mass of the heavy particle, zmj
The result is a macroscopic parameter, expressed entirely in term of microscopic quantities. We will try to estimate corrections to this quantity in later phases study. The following values for the diameters of the molecules in the bag are known to reproduce the experimental values for viscosity using the hard sphere model:

$d(CO_2)$=4A, $d(CO)$=3.6A, $d(H_2)$=2.9A, $d(N_2)$=3.7A

The unit A = Angstrom = 10^{-10} meters

The background gas as an ideal gas

The simplest fluid model imaginable is an ideal gas at rest. This model may naturally be improved later, but it is already highly non- trivial in the hybrid context. We therefore assume $v_{BG} =0$, and that the temperatures, pressures and densities are homogenous in the bag . These quantities will be determined by ideal gas equation of state, as well as the energy balance.

The BG molecules are assumed, due to large Maxwell-Boltzmann velocities, to fill instantaneously the available (unfolded) volume in the bag. In the hybrid gas model, we are not obliged to represent the BG by particles. It is, however, rather convenient to do so for « accounting » purposes (counting the number of particles, the total mass, energy, ect.). We take the ratio z to be the same for BG and PG.

The total internal energy of gas constituent in BG is therefore:

$$E_{BG}(i) = 3/2\, z\, N_{BG}(i)\, kT \qquad (12.)$$

From the emission rate function of the gas generator one may in principle (we will return to this later) infer a partition:

$$R(i) = R_{BG}(i) + R_{PG}(i),\ R(i)= \Delta N(i)/\Delta t \qquad (13.)$$

In the course of the bag inflation, the particles of PG, once their velocities become small enough (say p% of the most probable Maxwell-Boltzmann velocity) may be identified with BG. This will not modify the total number of particles. The energy of a PG particle is the sum of its kinetic energy and internal energy of the gas it represents:

$$E_{PG}(i) = ½\ zm(i)\sum vP^2 + 3/2\ zN_{PG}(i)\ kT \qquad (14.)$$

If p% <<100% the second term on (14.) is much smaller than the first one, a necessary condition for the particle representation to be coherent. The total internal energy for the gas i is:

$$E(i) = 1/2\ zm(i) \sum v_P^2 + 3/2\ zN(i)kT \qquad (15.)$$

The dynamic of the surface

The force on a given surface element is the sum of four contributions: $F = F_{BG} + F_{PG} + F_{SC} + F_{EL}$, where F_{SC} is the force exerted on the surface element due to collision with other surface elements, F_{EL} is the elastic force due to local deformation of the surface. The first three contributions are computed as follows:

$F_{BG} = P_{BG}\delta S = N_{BG}kT/V\delta S = n_{BG}kT\delta S$, δS is directed normal to the surface.

$F_{PG} = P_{PG}\delta S + \delta P/\delta t = n_{BG}kT\delta S + \delta P/\delta t$, $\delta P$is the total momentum transferred to the surface by the particle collisions, n_{PG} is computed locally in the vicinity of the surface.

F_{SC} is computed from the momentum transferred during collision of the two surfaces, assuming damping coefficients for the relative normal and tangential relative velocities.

We plan to apply the theory of finite element shells as the theoretical framework for elastic properties of bag. In our first simulation our goal is mainly to compare the ideal gas to the hybrid approach, and the elasticity model is therefore simplified.

If we define $l_{ij} = x_i - x_j$ to be the difference in the positions of two adjacent mesh vertices, the force acting on the vertex i is taken to be $FEL = -c\sum_j (l_{ij} - l_{ij}^0)$, l_{ij}^0 is the distance between the vertices in equilibrium, c is assume to be a constant independent of the mesh element: $c = Et/(1-v^2)$ E Young modulus, n Poisson's ratio and t bag thickness.

Analysis of the simulation of unfolded, infinite plane geometry

As a first step we have done two simulations:

Simulation 1: The gas ideal throughout the expansion, the pressure and the temperature are uniform in the bag at each time step, but may vary as a function of time (I.E. classical modelization used for Airbag)

Simulation 2: A hybrid system, with 95% of the incoming gas in form of particles. As the particles slow down due to the action of the drag force, they become indistinguishable from the background gas.

__Conclusion-__

1. The effect of the simplification of the elastic forces is apparent in the seams, where the meshing is rather irregular.

2. The particle approach simulation is very different from the ideal gas one both numerically and visually. The final pressure and temperature are lower, since:more energy is transferred to the bag, mainly during the first stages of the inflation. The results are, however, sensitive to the particle parameters, in particular their velocities as they come out of the gas generator.

3. The results are sensitive to the numerical integration method. Further theoretical study is needed in order to determine the best approach.

4. The measurement of temperature, and to a lesser extent of pressure, would be very useful for constraining the parameters model. The best information is, however, visual (analysis of films).

SIMULATION WORK

Description

The aim of this study was to evaluate the potential of the particle model for simulation of the airbag deployment.

For obvious simplification reasons, only « unfolded bag » case was treated in this study.

Parameters used in the model

__The gas parameters__ - The model of the gas is based on an Hybrid approach: an ideal gas component called background gas, and a jet component called particle gas.

- *The background gas:* It is considered to be an ideal gas. At the beginning of the simulation, we insert enough gas to fill it up to 1 atmosphere, so the difference of pressure inside and outside the bag is 0. We consider the initial temperature of the gas is 300°K. The average velocity of the background gas is 0 during the whole simulation. The background gas is not considered as made up of particle.

• *The particle gas:* The jet of gas from the generator are treated as particles. Each particle represents 6.37E17 molecules of gas. The temperature of the jet is constant: 1300°K. The particles are slowed down by drag which depends on the amount of gas inside the bag. When the particle velocity is lower than a given threshold (1/10 of the original velocity of the given particle) it is changed into background gas. When a particle bounces off the bag surface, its relative velocity after the shock is half of the one before the shock.

The generator is modeled by a collection of sources. For each particle of gas, four sources are given (figuring the 4 nozzles) . The emission of the particles is at 45° above the xy plane. The opening angle of each source is 30°. 95% of the gas coming out is treated as particle gas and 5% as background gas. The mass flow is taken to be constant during all the simulation. Finally, the velocity distribution is uniform for all sources.

<u>The bag parameters</u> -The geometry of the bag was given directly by AUTOLIV as a FEM mesh, figuring an Eurobag. The surface density of mass is 0.02 gr./cm2. We considered every triangles of the mesh as 3 springs with the same elasticity coefficient. This elasticity parameter is directly derived from the Young modules, the Poisson coefficient and the thickness of the material. The initial position of the bag is supposed to be an equilibrium state. The bag has no vent. There is no heat transfer between the gas and the bag so far.

The bag is above an infinite plane. The boundary condition for the nodes touching this plane is a bouncing condition: the z velocity component is inverted.

Analysis

In Order to validated our choices, we compared the dynamics of the bag processed on the computer with the real test motion pictures.

• The model of gas: Regarding the second simulation (flat bag with real gas), it seems that the motion of the bag due to the jet of gas is not fast enough. At the very beginning of the simulation, we can see distinctly the central growing hump caused by the powerful jet effect of gas. It is due to the local pressure consequent to the high density of particles. After a while, as the particles slow down they change themselves into background gas. So the local pressure decreases. After a few iterations the bag is only influenced by the global pressure of the background gas. That is why the hump stop growing. In the real test, the hump never stops growing. This difference between simulation and test is due to the initial velocity of the gas jet coming out of the generator. Arbitrarily, set to 40 cm/ms for the H2, it do not allow to reproduce the jet effect; but when the speed of the income particles is close to sound speed then the torch (or mushroom) phenomena is obvious.

Concerning the heat transfer inside the bag, the current model is very simple. There are only two different temperature in the system: the temperature of the jet (1300°K) and the variable temperature of the background gas. There is no heat transfer to the bag nor to the outside environment. It would be interesting to have some more information concerning the effect of such phenomena.

• The bag model: In the last simulation the model of the bag work properly. Never the less, the elasticity model is simplified. It works well for flat and regular meshed airbag. But for folded and irregular areas, we found some oscillations. We solved this problem numerically, but the best solution is to implement a real finite element module which will give better results. We also have to take into account the real friction between two parts of bag to be able to modelled the unfolding process of a fold bag.

FIRST RESULTS:

Here one will find the first results plots of the simulation of a flat airbag lining on an infinite plane:

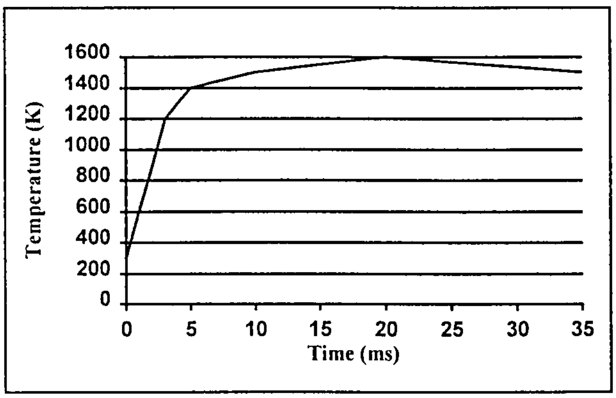

Figure 1. Temperature in the « flat airbag » when inflated versus time (K/ms) Simulation results.

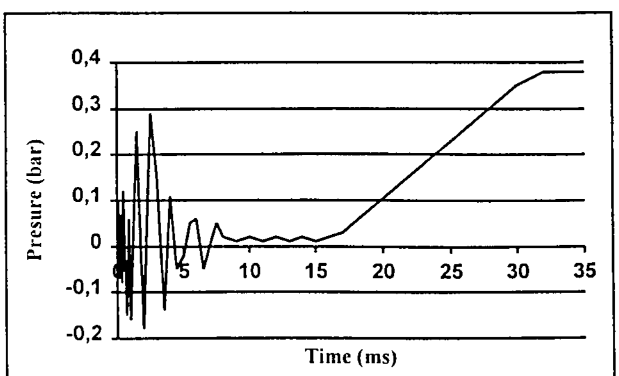

Figure 2. Relative pressure of the « flat airbag »

when inflated versus time (bar/ms) Simulation results.

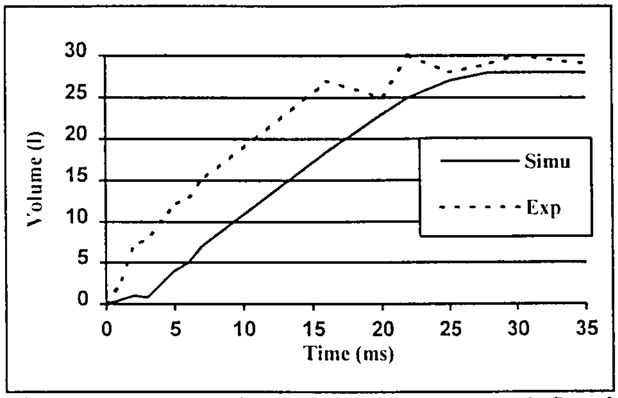

Figure 3. Volume of the « flat airbag » when inflated versus time (l/ms) Simulation results and experimental data.

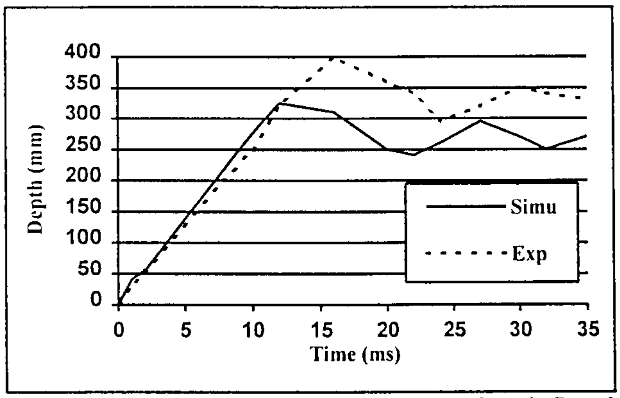

Figure 4. Depth of the « flat airbag » when inflated versus time (mm/ms) Simulation results and experimental data.

DISCUSSION

Temperature:

The temperature (Figure 1.) of the gas in the bag as a experimental results is some thing we do not know very precisely, but thinking the gas are perfect we could say:

$$pV/T = cst \quad (16.)$$

At ti=0ms we got 8liters of gas at Ti=295°K pressure=1atm

At tf=30ms we got 30liters of gas at Tf with a pressure of 1.125atm

Because of (16.) **the final temperature should be around 1245°K**

The simulation (Figure 1.) over estimated the temperature of 20% which is not to bad for a first step, in fact at the end of the simulation 95% of the gas are perfect and so with the pressure of 1.39atm and the volume of 28liters find the temperature is the one of a perfect gas. No surprise !

Pressure:

It is commonly admit that in an Eurobag inflated with a 8g Euroflator and no vent the pressure at 30ms should be around 1.125atm . The simulation (Figure 2.) over estimated it of 23%, this is acceptable for a first simulation result but should be improved.

Volume:

On the curve (Figure 3.) the volume plotted as experimental results was obtain from film analysis. We have supposed that the airbag shape was of rotational symmetry.

Over the differences between the two curves, one seems to be very important, in simulation the volume during the first 15ms in growing linearly which is obviously not the case in the experiment. This is surprising, especially as the depth of the airbag (see next paragraph and (Figure 4.)) is now very well correlated during the first 10ms. This could be link to the simulated shape of the airbag during the inflation which is not round enough. Two main things have to be study to solve this problem: the modelization of the fabric behaviour and the shape of the sources.

Bag depth:

On the curve (Figure 4.) the depth plotted as experimental results was obtain from film analysis.

We could observe that the simulation is very well correlated during the first 10 ms, but the where as the experiment still growing the simulation start to oscillate. That's also what one could see from the animation, at the very beginning of the simulation the effect of the gas sources are visible but the more gas are in the bag the slowest they are because of the interaction with the background gas. One can also see that in the simulation the effect of the jet is limited to the centre of the bag, whereas in reality the all face of the airbag is drag by the jet to produce the torch (or mushroom) shape. This could be linked to the fabric behaviour modelization (very simple elastic model compute for the moment), and moreover to the interaction of particles with it. At the beginning the fabric is not bend (because of the low pressure in the bag) and so the particles can not rebound on it.

FUTURE WORK

Conclusion

Indeed, a new theoretical model of gas was elaborated on the basis of « Hybrid » approach: background gas and particle gas. This model uses well-known microscopic parameters and common state variables from mechanics and chemistry. This model improves significantly the current models used in the industry.

A model of bag deformation has been implemented with surface interactions. It was possible to deal with complex numerical and geometrical problems related to such surface deformations. To integrate, with very few modifications, the bag meshes always used by AUTOLIV. The elasticity model was directly derived from common parameters employed in mechanics of structures.

With this model of gas, it was possible to demonstrate the non uniform motion of the surface of the bag due to the gas. Some parameters must be further adjusted in order to obtain more realistic dynamics.

Next phases:

The simulations predict a bag behaviour quite close to the actual one. The phenomenon of central bulge is shown though not large enough and of too low amplitude. To solve this problem, it would be necessary to study fabric behaviour and interaction of particles with it especially before the bending of the bag.

An improved finite element model must be implemented to obtain a better dynamic behaviour of the bag. These results will be necessary for a quantitative study on flat and folded bags.

The heat transfer must be studied and the influence of this phenomenon in the inflation process as well. Again, the flat bag configuration will be used first.

The study on folded bag will start only when a good level of correlation with the actual deployment will be achieved.

The final aim of this project is to be able to couple this methodology to a modelization of door opening, to allow the simulation of the first 30 ms of deployment in order to optimise folding of airbag and door definition.

THE FRONTAL IMPACT PERFORMANCE OF CHILD RESTRAINT SYSTEMS (CRS) CONFORMING TO THE ISOFIX CONCEPT

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Paper No. 96-S3-W-23

ABSTRACT

In Europe, universal child restraints are attached to the vehicle using Adult seat belts,-This has been found to result in interaction problems and in a high proportion of misuse. It is hoped that at some point in the near future a universal child restraint attachment concept will be adopted that will greatly reduce or eliminate some of the failings found in existing restraint designs and the misuses that these failings generate. The proposed child restraint fixation system, 'ISOFIX', is based upon the provision of suitable attachment points in appropriate seating positions onto which suitably equipped child seats may be simply and safely latched.

This paper details some initial dynamic testing of early prototype ISOFIX seating systems and, where appropriate, compares and contrasts these results with relevant data from similar tests with typical current UK child restraint systems.

INTRODUCTION

The ISOFIX concept of child restraint installation has evolved to a point where it is appropriate for it to be developed into an ISO standard. ISOFIX is based around a system of four 'rigid' attachment points mounted in a vehicle in appropriate seating positions. Onto these attachment points any suitably equipped ISOFIX CRS can be attached.

The anticipated advantages of this concept are:-

- Elimination or large reduction in incorrect fitting of the CRS to the vehicle.
- Reduction in child occupant excursions, decelerations and injury risk.
- Elimination of the vehicle seat cushion as a factor in both installation and dynamic response of the child seat.
- Elimination of the vehicle seat belt and anchorage positions as a factor in both installation and dynamic response of the child seat.
- Elimination of the need for adults to adjust the CRS restraining straps (seat belts and tethers)
- Improved stability of the CRS in the vehicle during general driving.
- Permit easier transfer of CRS between vehicles.
- A common stable fixation system for both forward and rearward facing CRS.
- To enable standards organisations to develop improved CRS standards.
- To enable CRS manufacturers to develop safer restraints by knowing more accurately how they will be used and fitted in vehicles.

Some organisations have recognised the potential for a partial use of the ISOFIX anchorages when used in conjunction with existing tether attachment points. One such partial use of ISOFIX is called CANFIX[1]. CANFIX utilises the two rear ISOFIX attachments and a top tether strap.

In conjunction with the Transport Research Laboratory (TRL), Middlesex University Road Safety Engineering Laboratory (MURSEL) has conducted a series of studies into the dynamic response and loads developed by CRS conforming to the ISOFIX and CANFIX concepts.

Twenty nine ECE R44 02 based sled impact tests were conducted at MURSEL. The study focused on two major areas of interest associated with ISOFIX and CANFIX. These are, the dynamic performance of typical ISOFIX seating systems upon the child occupant and the dynamic loading imparted to the ISOFIX anchorages and vehicle structure in which the seating system is installed.

The ISOFIX CRS used for the evaluation were supplied by two UK manufacturers. It is felt that they realistically reflect products that could be seen in the market place when ISOFIX becomes a reality. The CANFIX CRS similarly represented a realistic product, which, as with the ISOFIX examples, would be refined for production, but essentially reflects a likely future partial ISOFIX CRS.

[1] CANFIX has subsequently been called CAUSFIX.

PARAMETERS EVALUATED DURING TESTING

The parameters upon which these evaluations are based are those criteria specified in ECE R44 02[1] for the dynamic approval of current production CRS. The parameters are resultant chest acceleration ($\leq$55g); vertical chest acceleration ($\leq$30g); and head excursion related to the ECE R44 02 test seat Cr point ($\leq$550mm). *(The Cr point is a specific point on the ECE R44 02 bench test seat at the intersection of a plane on the top of the seat cushion and the forward surface of the seat back. The reference point on a vehicle seat is the 'H' point, but this bears no dimensional relationship to the Cr point.)*

DYNAMIC TEST FACILITY

Testing was conducted at the dynamic test facility of Middlesex University's Road Safety Engineering Laboratory. The dynamic impact test rig is based upon a design used for certification purposes by the British Standards Institution (BSI). The facility consists of an elastic cord propelled, rail mounted sled which is decelerated, in this instance, by the appropriate ECE R44 02 polyurethane tubes and olives. The facility has been described in detail elsewhere[2]. A TNO P3 child manikin was used fitted with head and chest tri-axial accelerometers. Sled deceleration was recorded and, where appropriate, belt forces and forces imparted by the restraint system upon its fixings. These were recorded using a PC based high speed data acquisition system. Kinematic motion of both manikin and sled throughout the event were recorded using either high speed film or high speed video equipment. Standard data processing techniques were employed during the analysis.

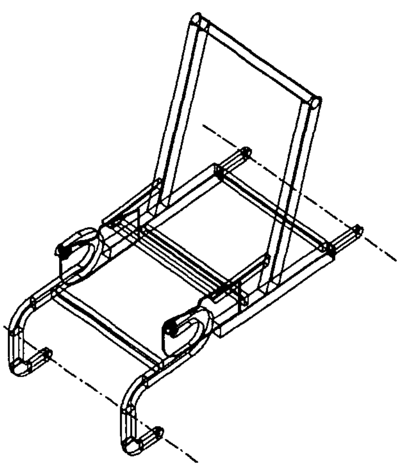

Figure 1 Four point ISOFIX CRS frame.

DESCRIPTION OF CRS EVALUATED

A typical ISOFIX CRS consisted of a steel frame incorporating two fixed attachment latches at the rear, and a hinged or telescopic front leg assembly, incorporating two further latches. Figure 1 shows a typical four point ISOFIX frame assembly.

Attached to the frame is a conventional child seat shell incorporating upper wings to afford a degree of lateral head protection. The harness system on both the ISOFIX CRS evaluated consisted of a five point belt system featuring single pull adjustment. Figure 2 shows an example of a complete seat assembly. This figure shows the 'rigid' ISOFIX CRS described later in the paper.

Figure 2 Four point ISOFIX CRS.

The CANFIX CRS was similar in design to the ISOFIX CRS but this particular example was of slightly greater mass and overall dimensions, with the following exceptions: The front leg assembly incorporating the two attachment latches of the four point ISOFIX design were omitted being replaced by a top tether affixed to the upper section of the seat frame in the area to the rear of the child's head. When installed, the CANFIX CRS is vertically and for/aft supported by the rear ISOFIX anchorages. The top tether pulls the CRS back against the vehicle seat back unlike the four point ISOFIX design which is completely independent of the vehicle seat position and strength characteristics. The harness assembly of the CANFIX CRS was of the three point variety, employing a 'T' shield in the junction between the shoulder straps and crotch/buckle attachment. It should be noted that this CRS incorporated an automatic harness retractor which assisted in

Figure 3 CANFIX 2 point + top tether

eliminating slack in the harness assembly. Figure 3 shows the CANFIX item evaluated in this test programme.

CRS. TEST SERIES

The tests detailed in this paper are divided into the following CRS groups :-

i) Hinged ISOFIX. This group includes data obtained from testing of seats from both CRS manufacturers.
ii) Rigid ISOFIX. Data for this group was obtained by locking the hinged front leg assembly of one of the above seats, hence producing a rigid ISOFIX base structure.
iii) CANFIX - Two rear points plus top tether.

The tests are, where appropriate, compared and contrasted with similar tests conducted with existing production UK CRS retrained on the 'vehicle' seat by the adult seat belt. These CRS were obtained from a variety of suppliers[3].

PRESENTATION OF HEAD DATA

The performance of a CRS is determined with respect to chest acceleration and head displacement. Within the paper head displacement is expressed in terms of two parameters:-

- Head Travel is the measurement of the movement of the head target (at the centre of gravity) at its maximum forward position of impact with respect to its initial position.
- Head Excursion is the measurement as defined in ECE R44 02 and assumes a calculated position of the Cr point, based upon the draft ISOFIX standard, forward of the rear pin centres

DYNAMIC RESPONSE OF P3 MANIKIN

The CRS were subjected to test pulses created by the stopping device specified in ECE R44 02.

Hinged four point ISOFIX

Initial testing was conducted with early ISOFIX seats, reflecting the original UNIFIX[2] suggested maximum mounting dimensions of 500mm horizontal and 160mm vertical between pins. Subsequent modifications agreed by the ISO/TC22/SC12/WG1 extending these dimensions to 520mm and 180mm respectively. All of the initial tests were conducted with the CRS installation angle set to the nominal 17°.

Figure 4 details chest acceleration measurements from fifteen tests conducted at a variety of impact velocities up to the full ECE R44 02 impact specification. Peak resultant chest deceleration for all the tests performed with this seat type lies within the range 20.4g to 43.8g and at the R44 pulse the range was 37.1g to 43.8g, and peak chest 'z' accelerations for all the tests ranged between 9.6g and 27.6g with the R44 range being 23.7g and 27.6g.

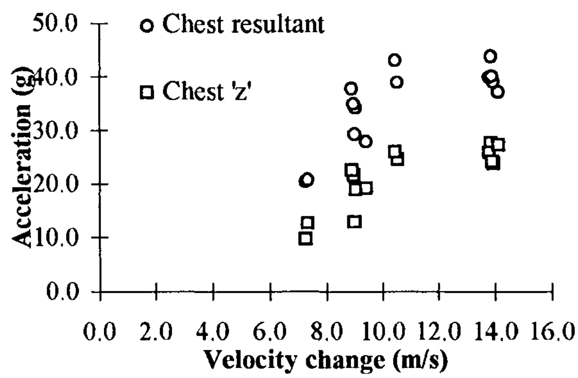

Figure 4 Chest acceleration in a hinged ISOFIX CRS.

Figure 5 details the head travel for these tests. The intercept on the ordinate axis is a function of the initial slack in the child harness. ECE R44 02 allows head forward movement of about 150mm before the child harness comes under load. The range of head travel in these tests was between 226mm and 440mm and at the R44 pulse 334mm and 440mm.

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[2] UNIFIX was a UK based working group that evaluated the original Swedish proposals as they would have applied to the UK market. The work of this group was adopted by ISO as the basis for ISOFIX.

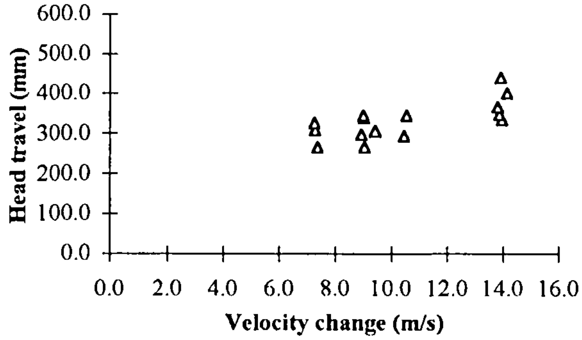


Figure 5 Head travel in a hinged ISOFIX CRS.

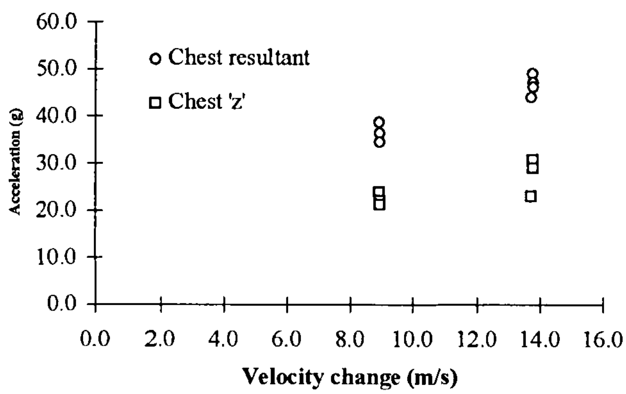


Figure 6 Chest response of rigid ISOFIX CRS.

Rigid four point ISOFIX

Figure 6 details the dynamic performance of seven tests of a rigid version of the ISOFIX CRS at impact velocities of 9m/s and 13.9m/s. The ISOFIX seat was set to leg lengths of 500x160 mm x 17° installation angle.

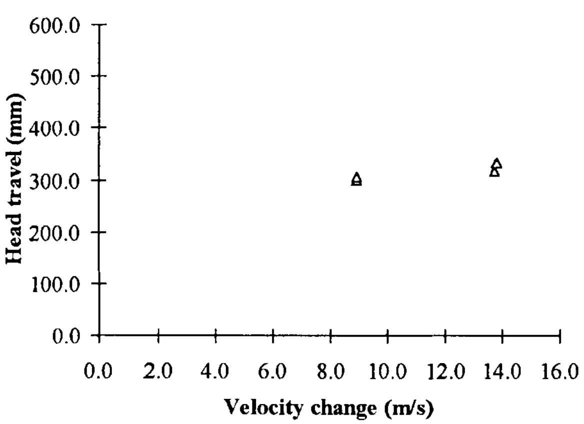


Figure 7 Head travel of rigid ISOFIX CRS.

The peak resultant chest deceleration for all these tests lie in the range 34.5g to 49.0g and at the R44 pulse 43.9g to 49.0g, whilst the peak chest 'z' accelerations ranged between 21.3g and 30.8g and at R44 pulse 23.0g and 30.8g. Figure 7 shows the corresponding head travel, for these particular samples of ISOFIX CRS where these were recorded during these tests. The intercept with the ordinate axis reflects slack in the child harness. Shortest head travel was measured at 300mm and at the R44 pulse between 317mm and 333mm but it should be noted that only four tests are included in this analysis.

CANFIX two point with top tether

Only one single test was conducted on the CANFIX CRS to the ECE R44 02 pulse. Peak chest resultant deceleration was measured at 44.5g, peak chest 'z' acceleration was 7.1g and head travel was 460mm. Head excursion ahead of the Cr point was 450mm.

SUMMARY OF DYNAMIC RESPONSE

The arithmetic mean of the ISOFIX results are summarised in Figures 8 and 9 with respect to tests conducted to the full ECE R44 02 pulse. They show that the more flexible hinged ISOFIX structure gives a superior resultant deceleration outcome for the occupant, at the expense of increased head travel. The CANFIX CRS interacts with the vehicle through the seat cushion and top tether and this gives greater head travel. CANFIX resultant chest acceleration is similar to the rigid ISOFIX CRS. Based upon this single test result the chest 'z' component of deceleration is greatly reduced, possibly due to the elasticity in the tether strap.

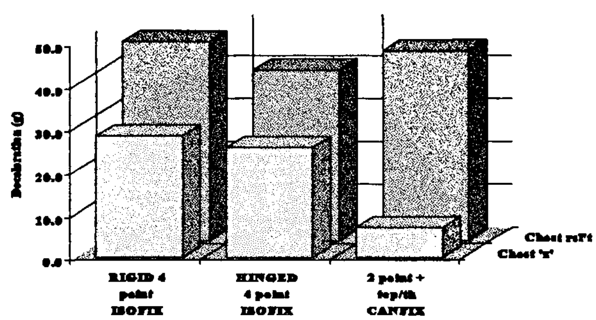


Figure 8 Summary of chest accelerations

LOADS IMPARTED BY ISOFIX/CANFIX CRS ON MOUNTINGS

The loads transmitted to the fixing pins and tethers by the various CRS were evaluated for all the tests using a

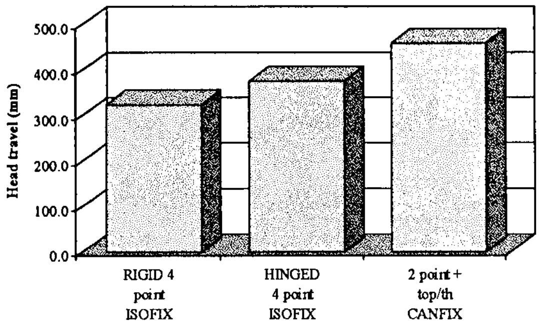
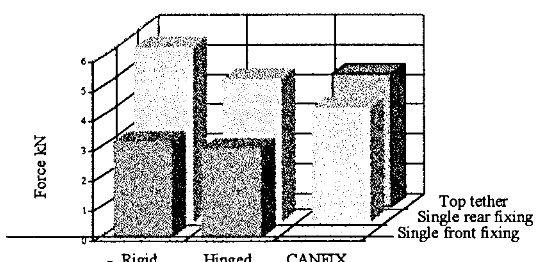


Figure 9 Summary of head travel.

CRS type	Rear attachment force kN	Angle of force to the vertical (Degrees)	Front attachment or top tether force kN	Angle of force to the vertical (Degrees)
Hinged 4 Point ISOFIX	5.9	40	3.2	159
Rigid 4 Point ISOFIX	4.8	53	3.0	166
CANFIX 2 Point + Top Tether	3.8	73	4.4	n/a

Table 1 Anchorage forces for the three CRS types

combination of specially developed load cells and Denton[3] belt force gauges. The data are presented in Table 1 for the different CRS types. Figure 10 displays these forces graphically.



[3] Denton's is the name of a US transducer manufacturer.

Figure 10 Summary of peak forces measured at the mounting pins and top tethers.

PERFORMANCE OF ISOFIX/CANFIX IN COMPARISON TO EXISTING CRS DESIGNS

To compare ISOFIX/CANFIX with existing UK CRS retained by adult seat belts Figures 11 (chest resultant deceleration) and Figure 12 (head excursion) have been combined. The following codes have been used to identify the different CRS and fixation systems in the figures.

- A — Booster seat employing the adult vehicle belt to restrain the occupant.
- B-D — Conventional framed CRS from a variety of leading manufacturers.
- E-M — Moulded CRS.
- HIF B D — Hinged ISOFIX manufacturer (B) deformed upon impact.
- HIF B S — Hinged ISOFIX manufacturer (B) stiffened to prevent deformation upon impact.
- HIF J — Hinged ISOFIX manufacturer (J).
- RIF B — Rigid ISOFIX manufacturer (B).
- CF — CANFIX.

Four point ISOFIX.

The figures show that the four point ISOFIX gives a valuable reduction in head travel, equivalent to the levels achieved by restraining the manikin with a booster seat and adult lap/diagonal belt, but without the high levels of chest deceleration associated with adult belt systems. The chest resultant deceleration levels are lower than those associated with the majority of currently available (ECE R44 02) CRS, but it should be noted that the chest 'z' decelerations were slightly greater.

The advantages offered by the four point ISOFIX in terms of total head travel beyond the Cr point are not as great as one might have expected. This can be attributed to the occupant placement upon the ISOFIX chassis in these particular samples of CRS to the ISOFIX concept, the head of the manikin being further forward than in conventional designs. With most, if not all existing belt retained CRS, the occupant's head (i.e. the upper back of the CRS) is pulled into close proximity to the seat squab during installation, the result being that the centre of mass of the head is positioned behind the Cr point of the seat (up to 80.0mm on the ECE R44 02 test seat). The ISOFIX systems reviewed tend to have much greater clearance to the seat squab, possibly to accommodate installation

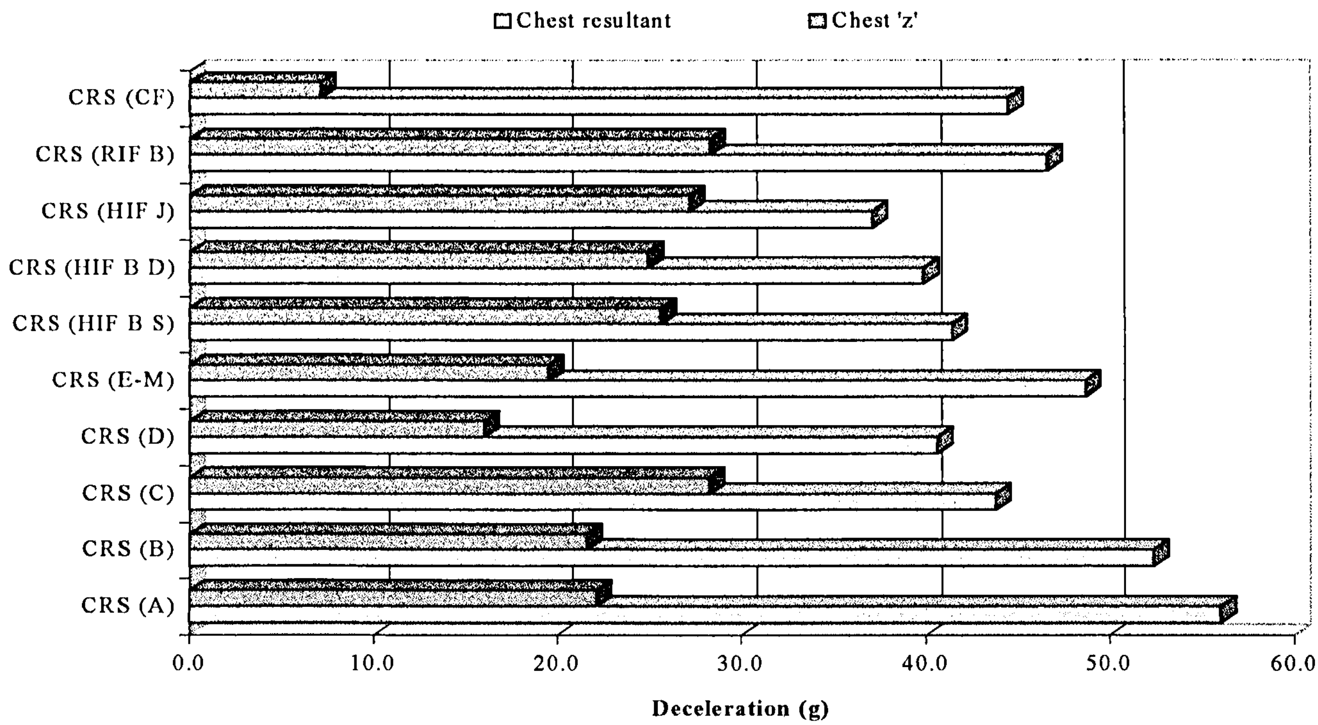


Figure 11 Comparison of chest accelerations.

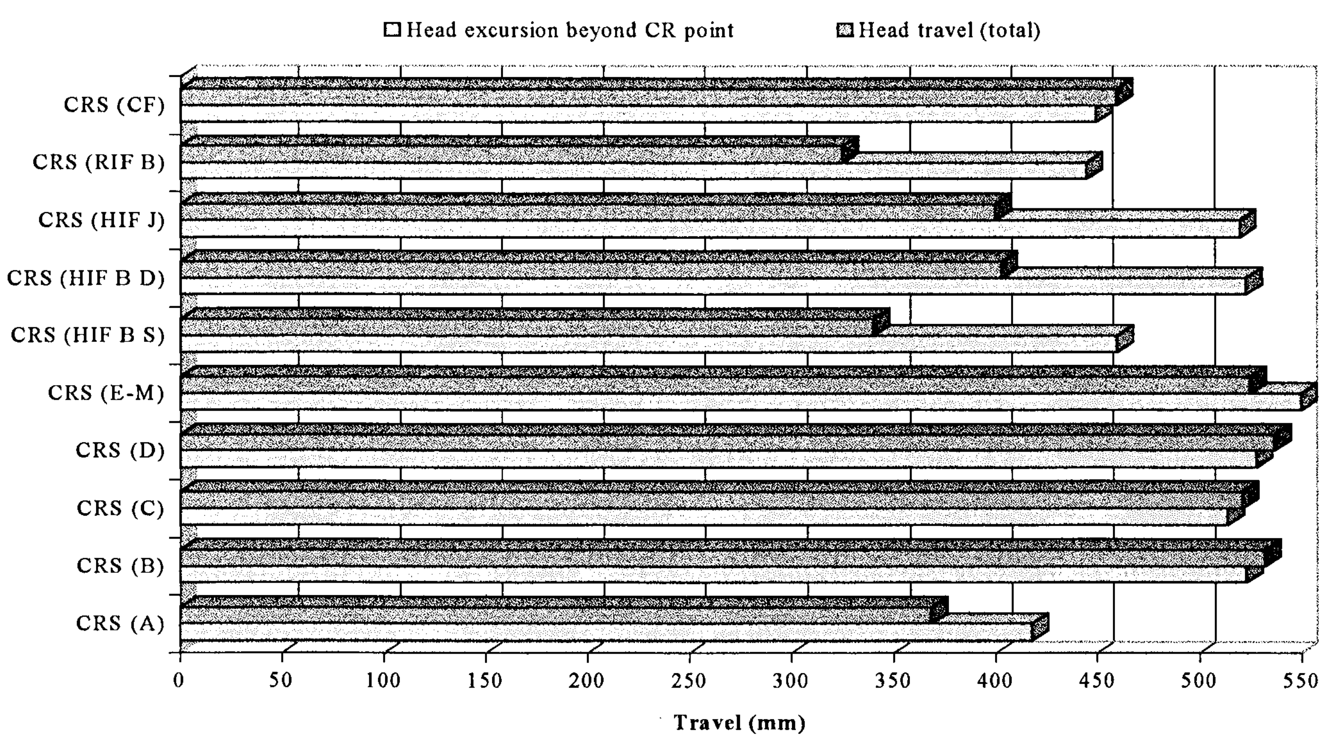


Figure 12 Comparison of head excursions.

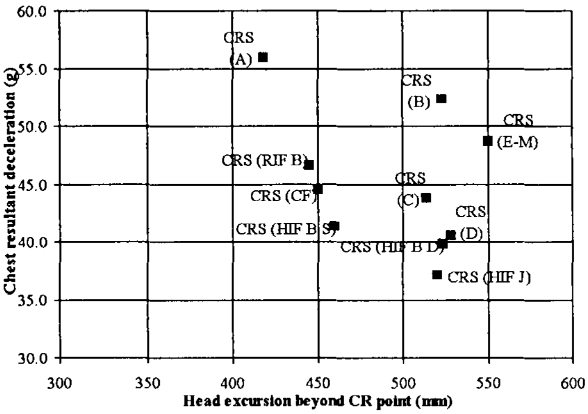


Figure 13 Comparison of CRS head excursion verses chest resultant acceleration.

in a variety of vehicles, this results in the occupant's head being positioned ahead of the seat Cr point (45.0mm on the ECE R44 02 seat).

It can be seen that plastic deformation of the ISOFIX frame during the impact offers advantages to the occupant with respect to chest deceleration, by acting as an energy absorber. However, the advantage is gained at the expense of slightly increased head translation.

CANFIX two point + top tether

Review of the semi rigid CANFIX system indicates that whilst not offering the full frontal impact performance advantages of ISOFIX, if fitted as designed, it does offer a high level of safety. Unfortunately if the top tether is not used or not tensioned to the manufactures recommendations the level of impact performance could be much worse. The head travel advantages of CANFIX are due primarily to the top tether pulling the occupant's head close to the seat squab, hence maximising the ride down envelope in front of the child. The seat cushion will however still play a role in the performance of this system, which in addition has limitations for rear facing applications.

Summary of performance comparisons

It is considered that the most important compliance requirements of ECE R44 02 describing CRS performance are forward head excursion beyond the Cr point of the vehicle seat and peak chest resultant deceleration. To compare easily the different CRS the two performance factors have been cross plotted. Figure 13 indicates the performance of the main CRS groups referred to in this paper (tested to the ECE R44 02 pulse), showing those systems which produce lower excursions and those which produce lower chest resultants. A lower head excursion may be more desirable in a small vehicle where occupant rear seat ride down space is limited. In a larger vehicle it may be beneficial to select a CRS which uses the greater available space ahead of the occupant. A graphical representation of this type has been previously proposed by one of the authors.[4]

A numerical comparison of the 'overall CRS performance' when installed upon the ECE R44 02 test seat can be compiled, based upon the product of chest resultant deceleration (m/s^2) and head excursion beyond the seat Cr point (mm), with the caveat that neither parameter can exceed the maximum ECE R44 02 acceptance limit. Figure 14 shows this assessment. CRS (A) is omitted since it exceeded the maximum permissible chest resultant deceleration.

An alternative assessment of CRS performance can be based upon the ratio of chest resultant deceleration (g) / head excursion beyond the seat Cr point (mm), normalised to the R44 approval limits. A ratio <1 would indicate a CRS favouring lower chest decelerations at the expense of head excursion, a ratio >1 would indicate the opposite. Figure 15 presents the data on this basis.

It is apparent from Figures 13, 14 and 15 that even with the design shortfalls previously mentioned with these early prototype CRS, the ISOFIX concept offers an improvement over current designs of restraint.

CONCLUSIONS

Four point ISOFIX CRS design

The four point ISOFIX restraint system offers an improvement in performance in frontal impacts compared to existing restraint designs. The particular samples of ISOFIX CRS tested located the childs' head further forwards than appeared desirable. If the childs' head was initially located further back on the base frame even greater improvements in performance, with resect to head excursion would be possible.

Further improvements in performance might be possible with the inclusion of controlled energy absorption in the system. This might be included in the attachment frame, child seat or harness design.

Two point plus top tether CANFIX design

The CANFIX two fixed point framed child seat, complete with a 'correctly adjusted' top tether, provides a level of performance similar to the ISOFIX four point fixing concept when tested to ECE R44 02 test requirements in frontal impacts, exceeding the performance of existing CRS designs of vehicle belt retained seats. However this improvement would not be achieved if the top tether were not used and adjusted correctly.

ISOFIX mounting pin forces.

Peak forces on a (single) mounting pin were found to be in the order of 6kN, This force level is felt to be within the acceptable working limits of the 6.0mm diameter x 25.0mm long mounting pins specified within the draft ISO standard.[5] It should be noted that the pins used for these tests were rigidly mounted. With body shell deformation the forces in a vehicle could be attenuated.

ACKNOWLEDGEMENTS

The authors wish to acknowledge the support of both Mr G T Savage and Mr C T Witherington for their assistance in the dynamic testing and subsequent data analysis at Middlesex University.

The bulk of the work described in this paper forms part of a Transport Research Laboratory research programme funded by the UK Department of Transport.

The views expressed in this paper are not necessarily those of the Department of Transport.

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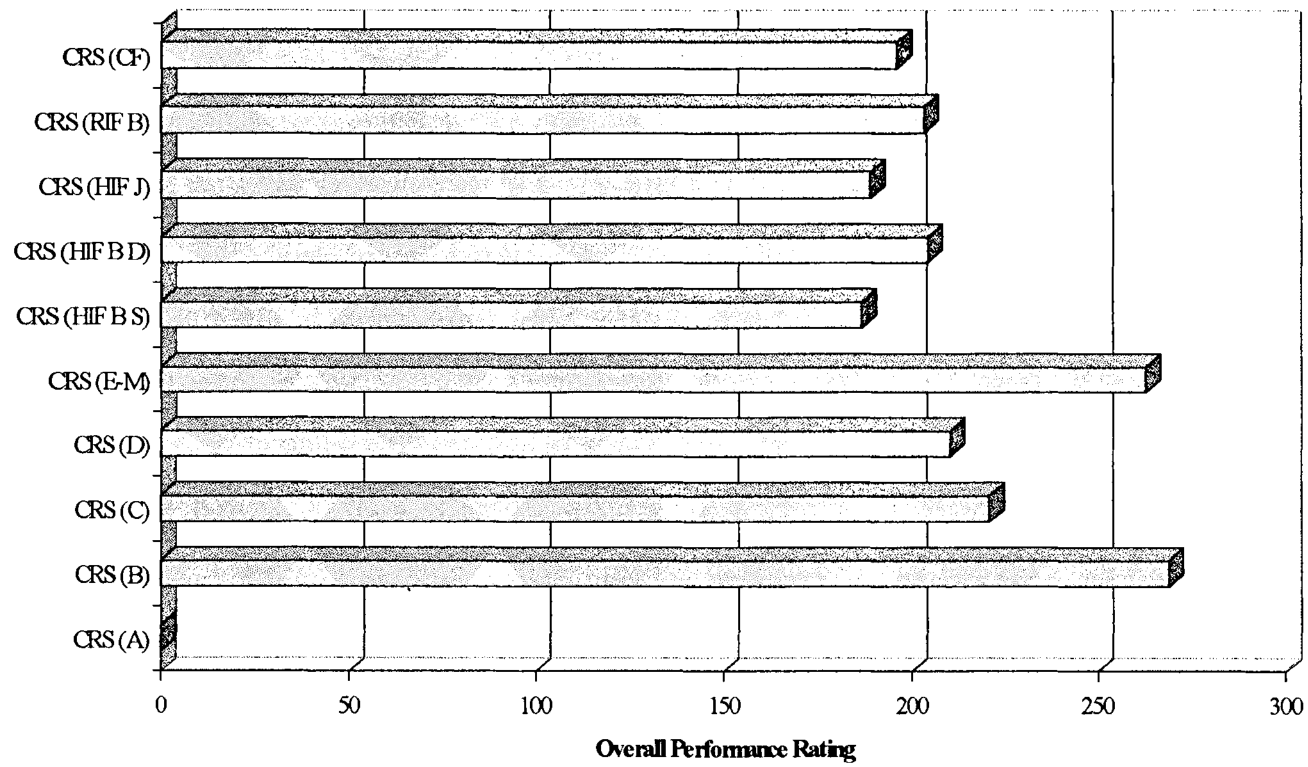

Figure 14 Numerical CRS comparison.

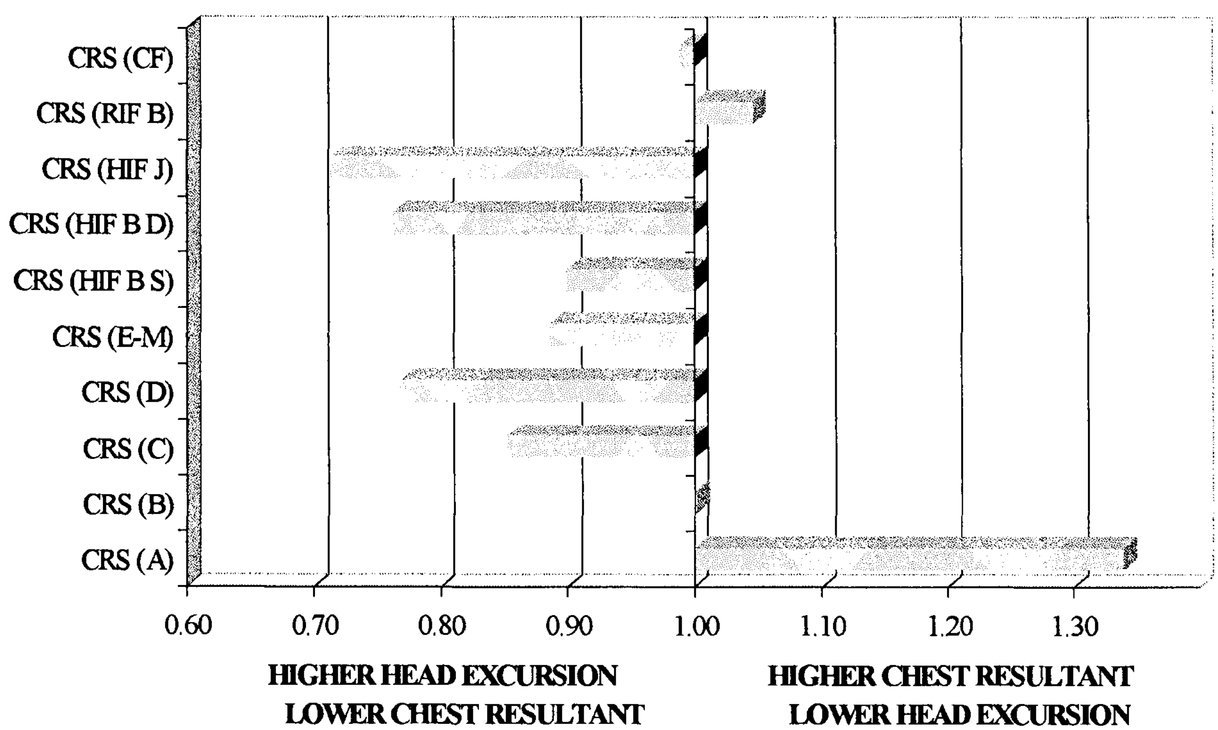

Figure 15 Numerical CRS performance.

A CABLE-SUPPORTED FRONTAL CAR STRUCTURE FOR OFFSET CRASH SITUATIONS

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ABSTRACT

The improved frontal crashworthiness of cars necessitates totally new design concepts, which take into account that the majority of collisions occur with partial frontal overlap and under off-axis load directions. Realistic crash tests with partial overlap have shown that conventional longitudinal structures are not capable of absorbing all the energy in the car front without deforming the passenger compartment. The reason for this is that the structure of the longitudinal members is specifically designed for meeting the more relaxed requirements of the compulsory full overlap test, in which both longitudinals are loaded axially.

Increased protection for the entire collision spectrum can be obtained by a frontal structure consisting of two special longitudinal members, which combine a higher bending resistance without increasing the axial stiffness. In addition the longitudinal members are supported by a cable connection system for symmetric force distribution. If only one of the longitudinal members is loaded during a partial overlap crash, the cable connection system will force the other longitudinal member to crumple as well. This results in normal programmed energy absorption. With this revolutionary concept it is possible to design a frontal car structure with the same stiffness for all overlap percentages and impact angles, resulting in an optimal crash pulse. The influence of various crash situations on the amount of energy absorbed by this system will be demonstrated by means of simulations and analyses.

INTRODUCTION

For improved frontal car safety it is necessary to design a structure that absorbs enough energy in each realistic crash situation. To protect the occupants, the passenger compartment should not be deformed and intrusion must be avoided to. To prevent excessively high deceleration levels, the available crush distance in front of the passenger compartment must be completely used for a chosen crash velocity. This implies that in a given vehicle concept the structure must have a specific stiffness. Normally, most of the crash energy will be absorbed by the two main longitudinal members with a progressive folding deformation of a steel column. The main problem is that in real car collisions these two longitudinals often are not simultaneously loaded axially. The majority of collisions occur with partial frontal overlap, in which only one longitudinal is loaded, or with an off-axis load direction. This implies that most longitudinals fail under a premature bending collapse rather than a much more energy absorbing progressive folding pattern. This gives rise to two design conflicts. The first conflict is that the same amount of energy must be absorbed with either one or with two longitudinals. The second conflict is that the same amount of energy must be absorbed in the case of an off-axis impact angle as in the case of a normal incidence impact. These problems can not be solved by just increasing the stiffness of the longitudinals in such a way that each longitudinal is capable of absorbing all of the energy. To absorb enough energy, a stiff longitudinal is needed for the offset crash in which normally only one longitudinal is loaded. The same longitudinal must be more supple in case of a full overlap crash, since both longitudinals must not exceed the desired deceleration level [1]. In addition, a stiff longitudinal is needed to absorb enough energy in an off-axis load direction (e.g. a crash test with a 30 degrees barrier) resulting in a higher bending resistance to help transform off-axis loads into axial loads and to prevent a bending collapse, and the same but more supple longitudinal is needed in the case of a normal axial load to avoid overly high deceleration forces.

To solve this design problem with its contradictory requirements, a new approach is needed in which the design for the frontal car structure is decomposed into separate parts each fulfilling its own function. The combination of these parts yields a complete vehicle structure which meets the requirement that in each crash situation (off-axis, offset and full overlap) nearly the same energy is absorbed and a similar deceleration level is obtained.

The next section presents a design solution based on this approach. It consists of a longitudinal with conventional axial stiffness but offering a much higher bending resistance. Furthermore, a new cable-supported system is supplemented to the designed longitudinals to provide a solution for the offset problem mentioned.

A NEW DESIGN CONCEPT FOR FUNCTIONALLY DECOMPOSED LONGITUDINAL MEMBERS

In designing a longitudinal member for optimal energy absorption, it is important to obtain a maximum value for the product of the mean crushing force and attainable crush distance. To guarantee that the required average force level is maintained over the entire available stroke of the column, a steady progressive collapse pattern must be maintained in the column. It is very important that the folding pattern rapidly converges to a stable, repeatable mode. If the progressive folding pattern is not sufficiently stable, the folding process can easily be disturbed and the structure will fail under a premature bending collapse. In this case no further energy absorption by progressive folding is possible.

To reach a stable progressive collapse pattern for many progressive lobes, an asymmetric folding pattern is preferred. As seen in earlier numerical simulations [2], a square cross-section folds in a stable asymmetric mode in contrast to hexagonal, octagonal and circular cross-sections, which fold in a less regular symmetric mode, despite having higher energy absorption at the same mass. Because stability is more important than energy efficiency, the square cross-section is used as baseline element for further improvements.

To determine the optimal wall thickness in relation with the optimal perimeter of the square cross-section, the following crash behaviour aspects obtained in earlier research [3] have to be considered:

1. The highest volume specific energy absorption is reached with a large wall thickness in conjunction with a small perimeter value.
2. The highest bending resistance (necessary to prevent a bending collapse in case of an off-axis load) is reached with a large wall thickness in conjunction with a large perimeter value.
3. The bending resistance decreases less in a fold if the ratio of the folding wave length to the profile width is smaller. This is optimal (lowest relative disturbance) for a small wall thickness in conjunction with a large perimeter value (due to a smaller decrease of the cross-section width during deformation).
4. The most stable force level is reached with a large wall thickness in conjunction with a small perimeter value.

From these results it is clear that it is impossible to select a profile that simultaneously meets all four mentioned design criteria. See Figure 1.

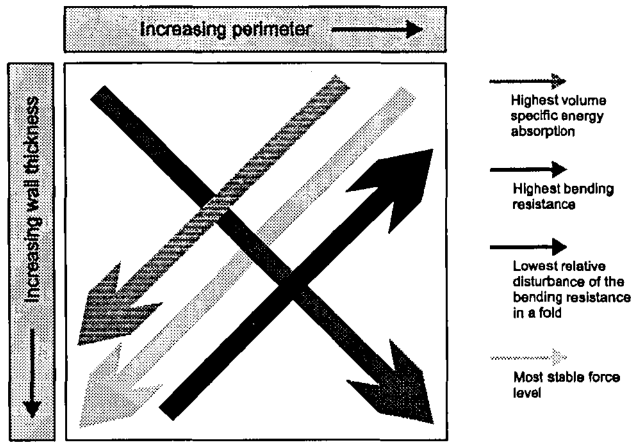

Figure 1. Design paradox, four design requirements resulting in different perimeter / wall thickness choices.

Also, the addition of flanges or ribs to the basic square cross-section for more bending resistance should be avoided, as each attachment to the profiles can disturb the normal stable folding pattern. These disturbances can result in an unstable folding pattern, from which the irregular perimeters introduce weaknesses, which in turn result in a premature bending collapse. For optimal folding behaviour, the column should be ideally positioned without any fixed attachment to disturb the natural folding process.

The new concept is based on the design philosophy that an optimal longitudinal member must be functionally decomposed into two separate systems: the first, called the crushing part, guarantees the desired stable and efficient energy absorption. The other, called the supporting part, guarantees the desired stiffness in the transverse direction. (Figures 2 and 3). This latter part is necessary to allow enough energy absorption during an off-axis collision and to give enough support with a sliding wall to protect the crushing part against a bending collapse.

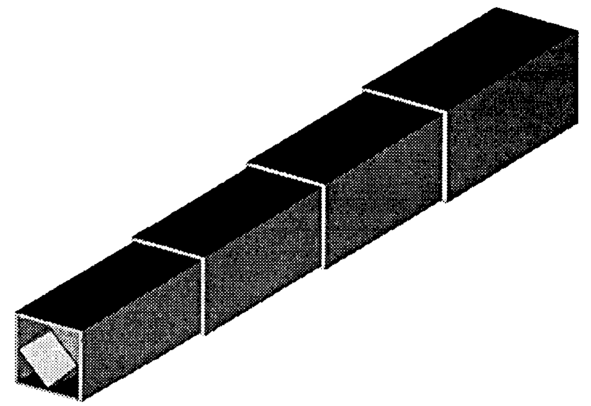

Figure 2. 3D View of the longitudinal member with the crushing part inside the supporting part.

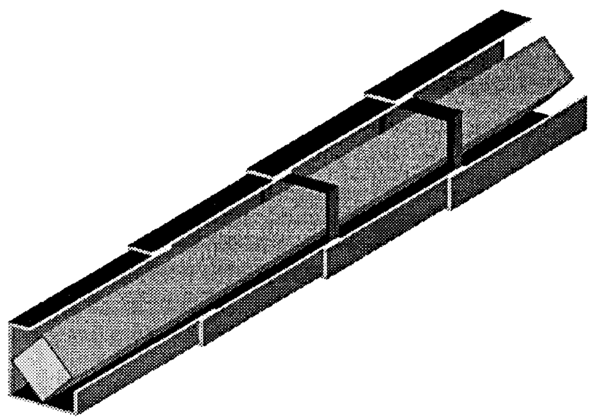

Figure 3. Interior view of the longitudinal member.

A square profile is chosen for the crushing part with a width of 70 mm outside and a thickness of 2 mm. The width dimensions of the crushing part are limited, as it has to fit within the available interior dimensions of the supporting part, depending on the available space between the engine and wheel envelope. The total length is 1160 mm. With a theoretical effective deformation of about 72.5 per cent of the length not deformed [4], the deformation that can be obtained measures 840 mm, an acceptable value for a middle class car [1] in the case of a 30 - 40 per cent overlap crash. The supporting part consists of four very stiff square profiles that fit into each other and may slide each over the other, like a telescope. Flanges prevent the telescope from falling apart. Two supporting squared rings are necessary to prevent a bending collapse of the crushing part in the larger rear parts of the telescope. The same length of 1160 mm can be shortened to 360 mm. See Figure 4 for more details. The space between the corners of the crushing part and the inside of the supporting part is only 0.5 mm. At both ends of the longitudinal member the two functional components are connected with a rigid plate.

The unusual angular orientation of the crushing part along the longitudinal axis of 45° with respect to the orientation of the enveloping support part has several advantages. At its corners the crushing part is supported by the enveloping square. No material deforms to the outside at the corners, which implies that at this position contact with the supporting part does not disturb the folding process. The narrow position of the crushing part in the enveloping supporting part gives a continuous sliding force as a support against bending. Note that during the deformation process the first part of the supporting structure with the smallest inner dimensions slides together with the folding front to the rear. After full deformation all the folds are packaged in the first supporting part. Figures 5 and 6 show the lobes of the crushing part inside the supporting part after deformation.

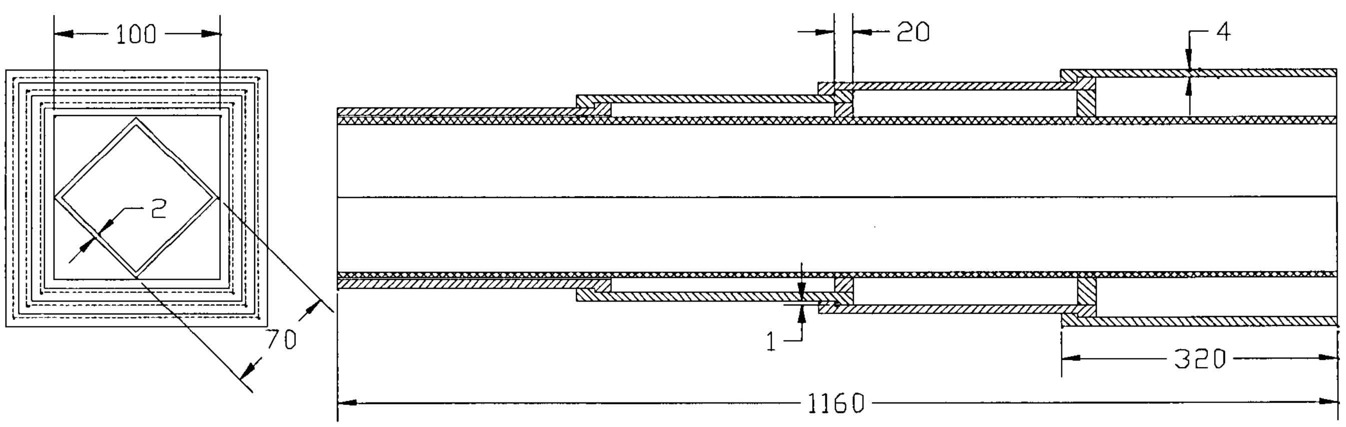

Figure 4. Drawing of the longitudinal member.

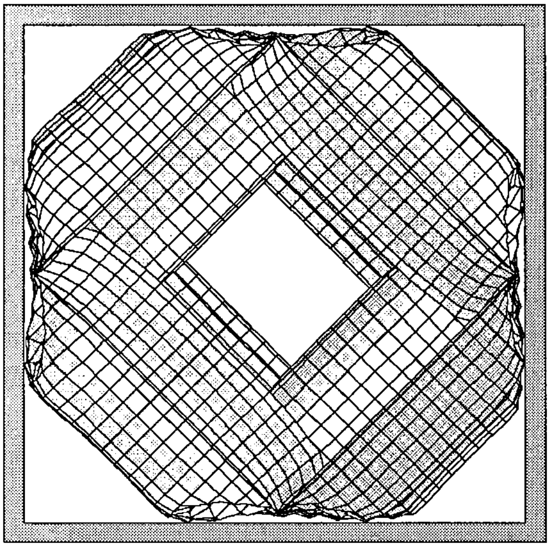

Figure 5. Front view of the crushing part inside the boundaries of the supporting part after deformation.

The space for undisturbed folding is always guaranteed. The width growth of an asymmetric fold is nearly half the width not deformed, this is possible due to the rotated orientation within the enveloping square.

The result is a longitudinal member with a conventional stiffness for stable energy absorption during a full overlap crash. It also has an extremely high bending resistance to absorb enough energy during an offset or off-axis collision,

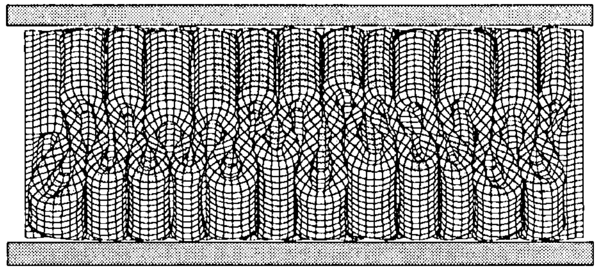

Figure 6. Side view of the crushing part inside the boundaries of the supporting part after deformation.

because a transverse load component can be transferred to an axial load.

Combining these two longitudinals with a rigid connection beam (a cross member) at the front ends, results in a frontal structure. It is supposed to be fixed to the stiff fire-wall at the rear ends. A numerical simulation with an axial 30 per cent offset load showed that the loaded longitudinal deforms very regularly with a maximal energy absorption and the unloaded longitudinal deforms by bending, due to the tensile force in the cross member. For a more realistic simulation the rigid engine block is modelled between the two longitudinals. See Figure 7 for the final deformed state after an offset load.

As a result of the high transverse stiffness of the supporting part of the bending longitudinal, it absorbs a considerable amount of extra energy in comparison to a

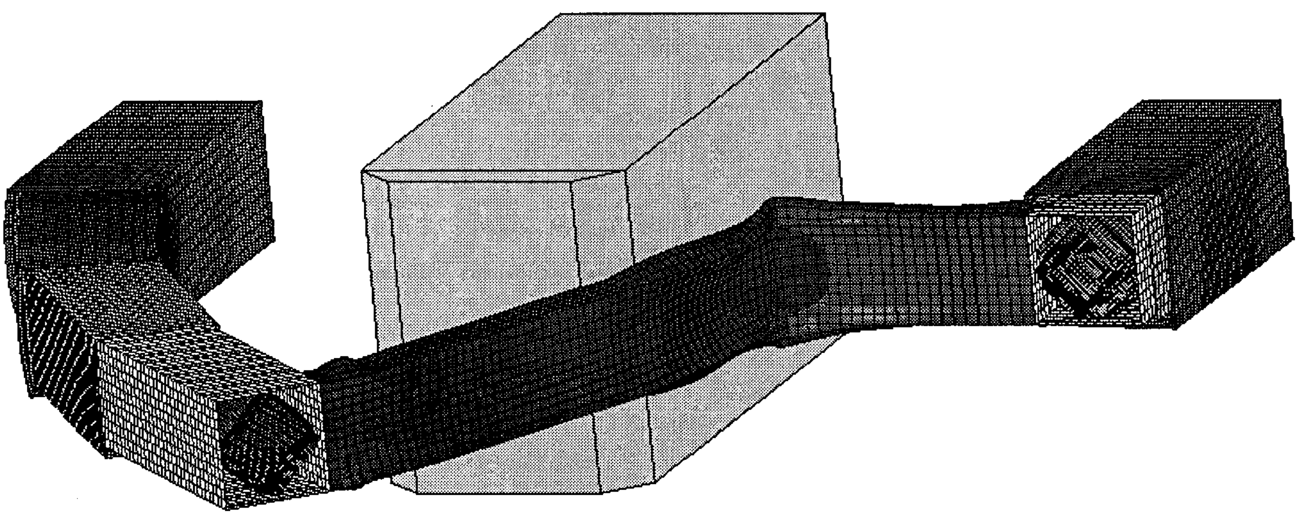

Figure 7. Thirty per cent offset crash of the new design concept after 800 mm axial shortening. For video see References.

conventional longitudinal. Nevertheless, it is better to reach the same amount of energy absorption in the case of an offset crash compared with a full overlap crash. Although this can be reached by further increasing the wall thickness of the supporting part, this is unfeasible if an acceptable mass for the total structure has to be obtained. Energy absorption by bending is a very inefficient method. A considerable amount of energy absorption is only possible with heavy structures. The only way to reach the same energy absorption and deformation length compared to a full overlap crash for an offset crash is to force the unloaded longitudinal to crumple as well, by means of a stable axial folding process. This is not possible with a very rigid transverse structure in the front. Bending moments will be introduced that are always too high. Also, the engine between the longitudinals reduces the possible shortening after a rigid transverse beam hits the engine. An interesting solution is the addition of two cables and two bars to the designed longitudinals. This cable connection will force the unloaded longitudinal of an offset crash to perform axial shortening with a tensile force to the rear. This new design idea will be introduced in the next section.

THE CABLE CONNECTION SYSTEM FOR A SYMMETRIC FORCE DISTRIBUTION

In Figure 8 a schematic sketch of a cable-supported frontal car structure is given. The system consists of two bars, two cables and four cable guides. The stiff bars are placed within the longitudinal members. At the front of the vehicle they are connected with the cross member. The bars are longer than the longitudinals and extend beyond the vehicle's fire-wall. A cable is connected to the end of each bar. This cable is guided to the front end of the other longitudinal via two cable guides, where it is connected to the cross member. The working principle is rather simple: if one longitudinal is loaded and starts deforming, the bar moves backwards and pulls the cable, which leads the crushing force via the cable guides directly to the unloaded longitudinal. The transmission of force is without loss of energy. Note that if both longitudinals are loaded (full overlap crash), the cable construction has no influence on the crash behaviour.

This cable concept could be built into every car with a conventional frontal structure. However, the new design concept described offers two important advantages, which make it very suitable for combination with the cable construction. First, the bars need to have sufficient space to move back-wards. Because intrusion of the passenger compartment is not desirable, the bars must move under the vehicle. This means that the longitudinal members need to be positioned under a slight angle (higher on the bumper side, lower on the fire-wall side), due to the prescribed compulsory height at which the forces must be led into the structure. The new design concept is well suited to be positioned under an angle. Its high bending resistance guarantees that the structure will not collapse in a premature bending collapse, unlike most conventional

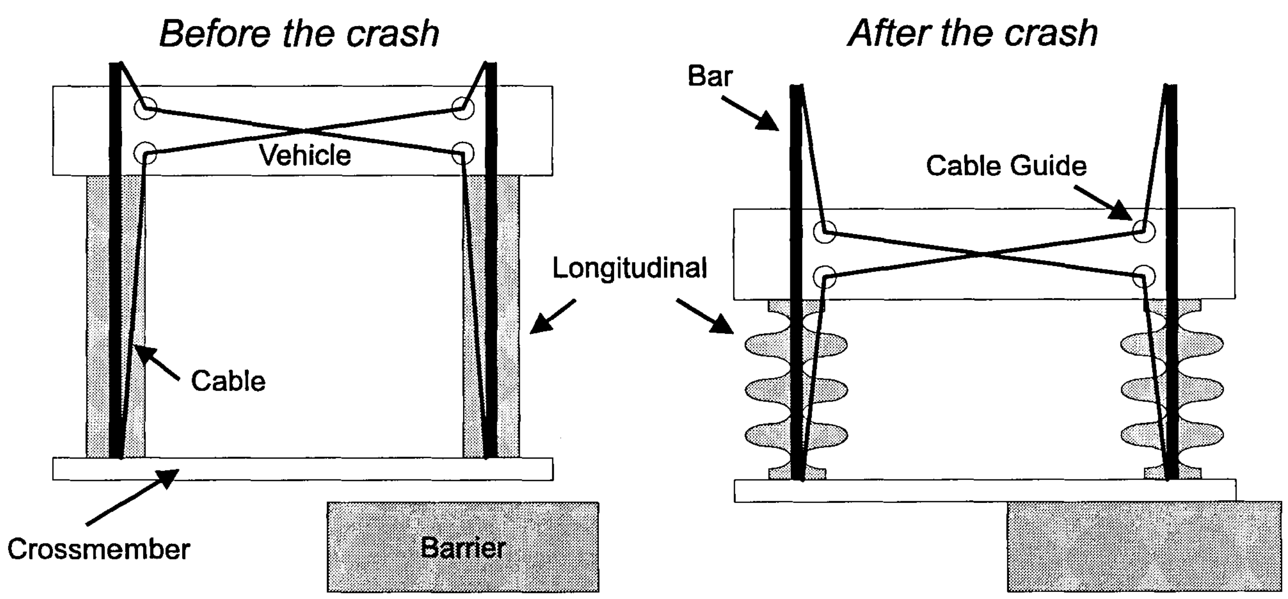

Figure 8. Principle sketch of a cable-supported frontal car structure.

longitudinal members. Second, the new design concept guarantees stable folding of the crushing part under all circumstances. Most conventional longitudinals have all kinds of connections with other parts under the bonnet, which can easily disturb the folding process. A stable folding process is necessary, because the bar is placed within the crushing longitudinal and should always be free to move back-wards. (Unstable folding would prevent the bar from sliding within the narrow space of the crushing profile. This would cause the cable system to stop working correctly). To avoid any transverse forces on the sliding bar, the cable is guided through the centre of the bar. This is possible if the two bars are formed like a U-profile and the cable guides fit into these U-bars. See Figures 9 and 10 for more details.

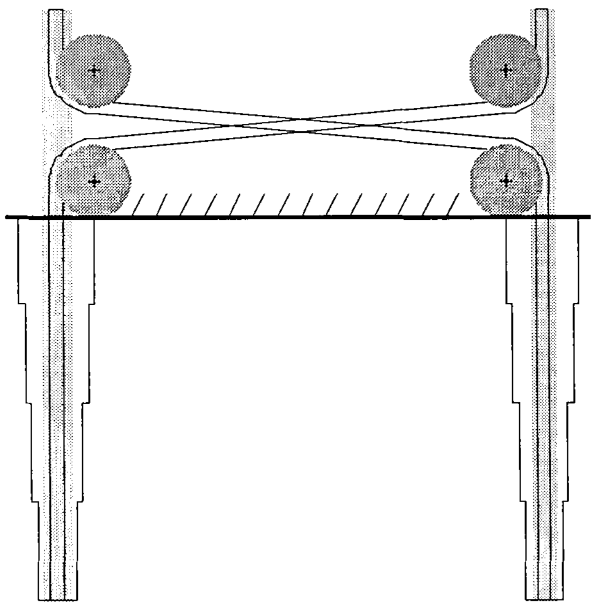

Figure 9. Top view of the cable-supported frontal car structure.

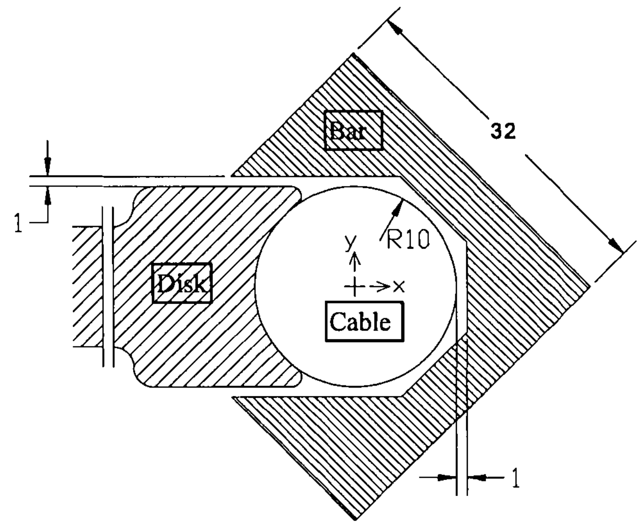

Figure 10. Cross-section of the cable around the cable guide disk inside the bar.

A steel 8-string wire-rope cable with a diameter of 20 mm has a fracture force of 279 kN. That means that a force up to 167 kN does not deform the cable plastically. Simulations [5] showed that in a real offset crash the peak load is below this value.

The free space inside the stable asymmetric deformed square crushing part is decreased to about half the original width. This was confirmed by simulations and experiments in our laboratory and done by others [6]. This 50 per cent decrease means that the inner dimension of the square crushing part after forming folds will be nearly 33 mm. To prevent each disturbance of the regular folding process, and to guarantee enough sliding space, an outside width of 32 mm of the square sliding bar is chosen.

The cable guide disk has a minimised height of 20 mm, the same as the cable diameter. This is important because the bar must not be weakened more than necessary.

The buckling load of the bar was calculated to be 171 kN. This is also more than the expected peak load during a crash. For the buckling load, the free length of the bar is the same as the longitudinal length. Behind the fire-wall extra leading support for the sliding bar is necessary to ensure that movement only occurs in the axial crush direction. During the crash the free buckling length decreases by additional support from the formed folds.

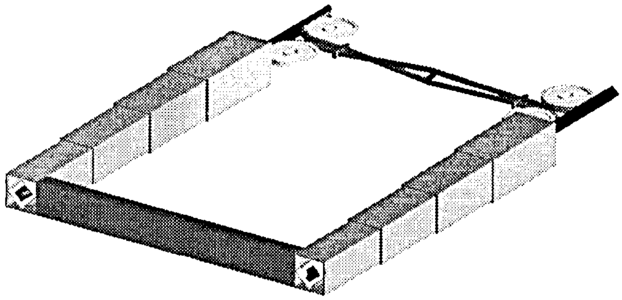

Figure 11. 3D view of the cable-supported frontal car structure.

In Figures 11 and 12 the assembly of the new design concept with the cable connection system can be seen. Also, extra cable guide rings preventing the cable from slipping off the disk and a pin mounted to the fire-wall at the crossing point of the cables are showed. Note that the centre lines of the cable, bar and longitudinal fall together yielding axial forces only. The position of this cable-supported structure built inside a car is showed in Figure 13. Note that the two bars can move freely to the rear under the car floor during a crash due to the position under a slight angle.

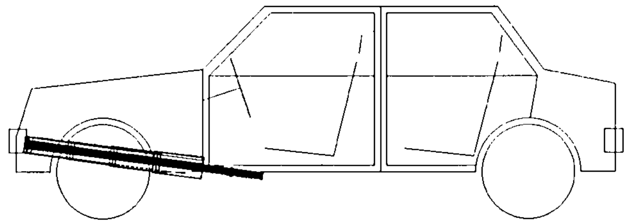

Figure 13. Side drawing of the cable-supported frontal car structure positioned in a car.

COMPUTER SIMULATION OF THE CABLE-SUPPORTED FRONTAL CAR STRUCTURE

To evaluate whether the designed cable-supported structure really works, and whether the absorbed energy is about two times the energy absorption of one longitudinal during an offset crash, a numerical model was built with the finite element program PAM-CRASH. The model consists of more than 50000 elements and many contact definitions to describe the folding and sliding process accurately. To avoid unrealistic simulation results, and to focus on the longitudinals only, the calculation was stopped at the moment the rigid engine block became involved. In this model a shortening of 510 mm is used as the deformation position to compare the amounts of absorbed energy between various crash situations.

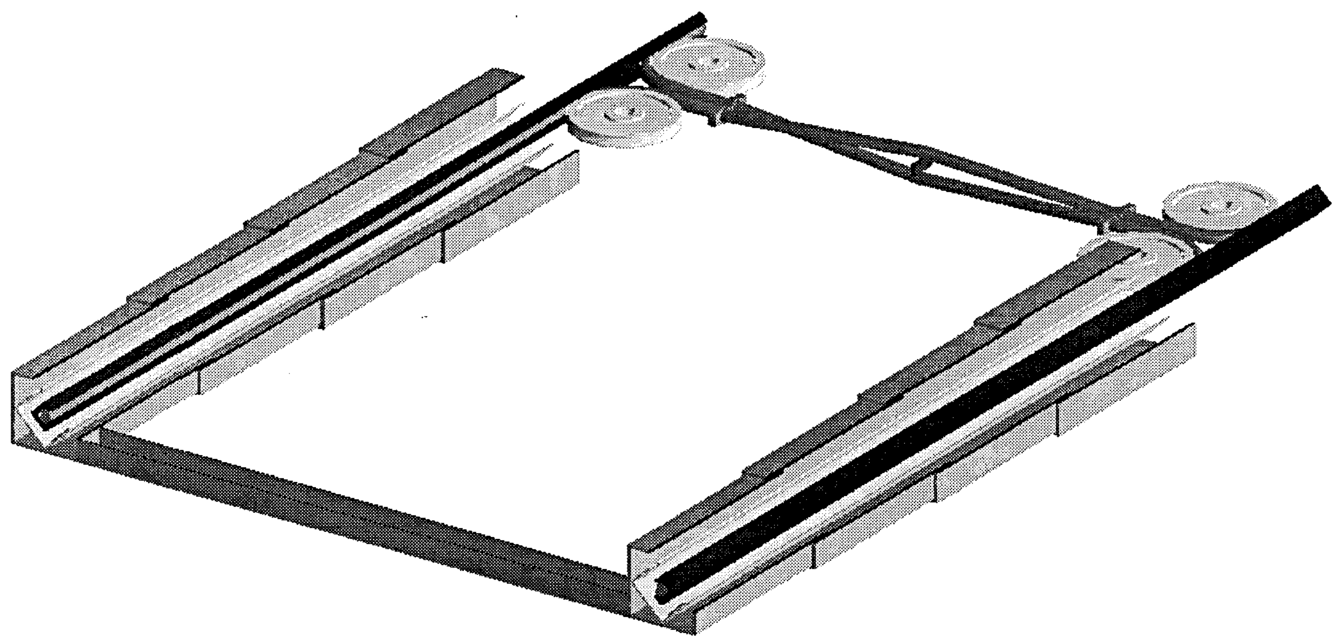

Figure 12. 3D open view of the cable-supported frontal car structure.

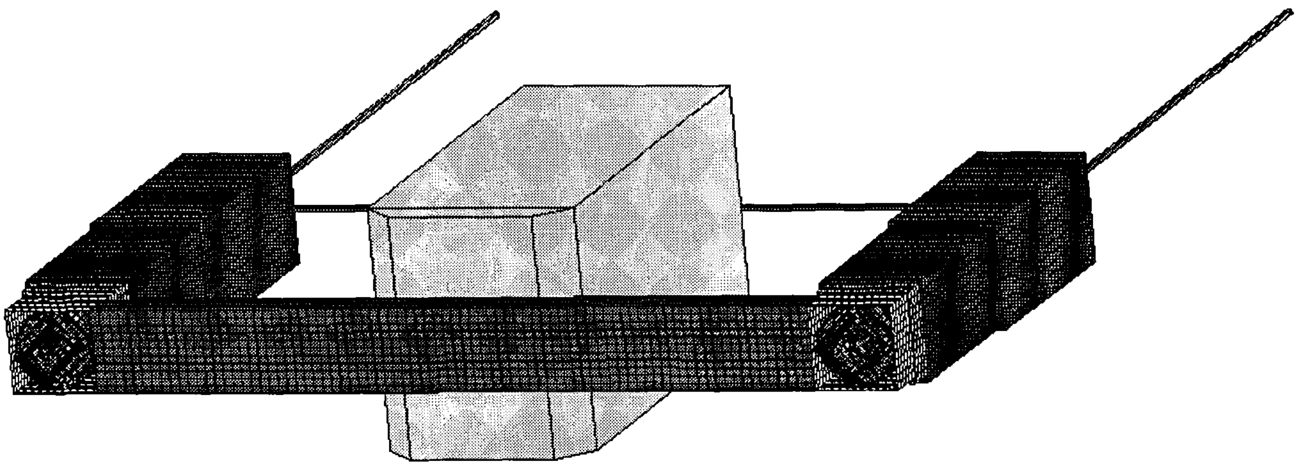

Figure 14. 3D view of 30 per cent offset crash of the cable-supported frontal car structure after 510 mm shortening.

Figures 14 and 15 show the deformed state of an offset crash with a moving rigid barrier of 1100 kg and an initial velocity of 56 km/h.

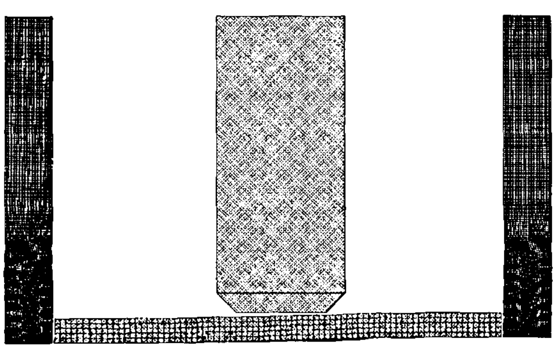

Figure 15. Top/inside view of 30 per cent offset crash of the cable-supported frontal car structure after 510 mm shortening.

It is clear that both longitudinals are deformed equally despite the asymmetric external offset load. The crushing force in the loaded longitudinal is transmitted to the unloaded longitudinal. In Figure 15 the supporting parts are afterwards removed to see the regular folding pattern in both crushing parts.

The energy absorption of this 30 per cent offset crash is 58580 J. This is plotted in Figure 16 with the solid line. This energy absorption is compared to the energy absorption of the new design concept (NDC) without the cable system during a 30 per cent overlap (dashed line, 35672 J) and a 100 per cent overlap (dash/dot line, two times the energy absorption of one NDC longitudinal without internal sliding bar, 61271 J). Also the energy absorption of one axial loaded NDC longitudinal is plotted (dotted line, 30635 J).

Although the high bending resistance of the new design concept structure gives an increment of energy absorption by nearly 17 per cent compared with a single NDC longitudinal during an offset crash, resulting in 58 per cent of the full overlap crash energy, the addition of the cable system increases the energy absorption by more than 91 per cent, resulting in 96 per cent of the full overlap energy. These are percentages for the same axial shortening. The difference of 4 per cent is caused by the fact that a longitudinal with an internal sliding bar absorbs about 4 per cent less energy than the same longitudinal without the internal bar as first used in the new design concept. The reason for this is that a square tube does not naturally deform with an exactly square inner space. The space is a little likely to be rectangular. The inside folds can use the bar as a support in forming the following outside fold with a little lower force level. Of course the energy absorption with the same cable-supported structure for full overlap or with offset crash is the same.

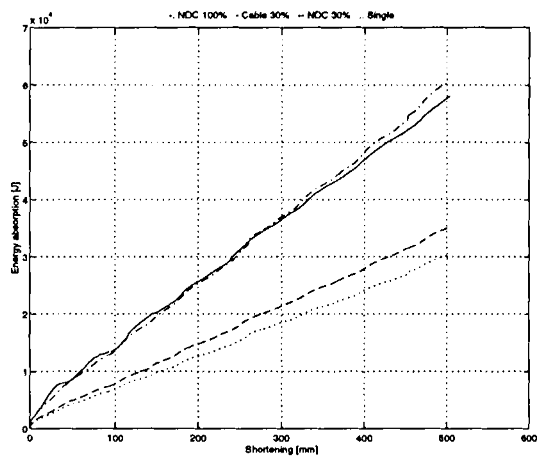

Figure 16. Comparison of the energy absorption between four situations.

CONCLUSIONS

Initially, a new design concept is proposed for the vehicle front, made up of two interconnected longitudinal member systems, each consisting of an interior crush component that resides inside a telescoping outer support component. This concept is capable of providing regular axial stiffness values in conjunction with very high transverse stiffness values. This is necessary to also absorb a considerable amount of energy during an offset or off-axis crash situation instead of the compulsory full overlap crash. However, to obtain the same level of energy absorption during an offset crash as during a full overlap crash without an extra deformation length to protect the passenger compartment, extra measures are necessary to better involve the unloaded longitudinal in the energy absorption. In the second stage a structure consisting of two stiff sliding bars and two cables connecting the rear of one bar inside one longitudinal to the front of the other longitudinal is added to the new design concept to transmit the crushing force from the loaded to the unloaded longitudinal. Numerical simulations showed that both longitudinals have a stable folding pattern during an offset crash over the full deformed length, resulting in a doubling of the energy absorption with regard to the energy absorption of only one axial loaded longitudinal. This means that during offset collisions, the cable-supported frontal car structure is able to absorb as much energy as during a full overlap collision with the same axial shortening. This is a very important result, because with this revolutionary design the same deceleration level of the car is reached for each crash overlap percentage. Now it is possible to design a frontal car structure with one optimal stiffness that hardly varies for different crash situations. Hence, one optimal occupant deceleration level yielding the lowest injury levels is obtained over the entire collision spectrum.

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Note: A video animation of the new design concept as showed in Figure 7 can be found as three Mpeg files on the internet site:
http://www.tue.nl/urc/visualisatie/rc_vis_exampl.html

A COUPLED APPROACH OF SIMULATION AND OPTIMIZATION TO DESIGN SAFETY SYSTEMS

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Paper Number 96-S3-W-27

ABSTRACT

This paper presents a methodology for designing safety systems during early concept phases by coupling crash victim simulation and optimization techniques. According to a global strategy, the different steps achieved to implement such a tool are described and illustrated on a typical frontal crash example.

INTRODUCTION

ECIA benefits from supplying various car components such as seats, steering-columns, dashboards, etc., which play a significant part regarding the safety definition of a car. Consequently, these components have to be globally optimized in this respect. The proceeding traditional way consists in using statistical techniques in conjunction with either experimental or numerical simulation-based approaches. However, the associated analysis remains limited to a discrete set of parameters levels (typically a low value and a high value). When the behavior of the system to be considered tends to be very complex, a continuous analysis turns out to be much more appropriate. One of the means to investigate such methodology is to combine numerical simulation and optimization techniques. With the exceptions of a general presentation (Gabler 94) and another very recent one (Hou 95), such a coupling has not been addressed in the litterature of automotive safety. This paper presents our approach in the context of frontal crash (which is generally considered to be the most significant in terms of potential benefit). It deals more precisely with the optimization of the direct environment of the dummy.

LEVEL OF ANALYSIS AND CHOICE OF THE MODEL

The expected level of analysis corresponds to an early design phase. At this stage, a global model of the system to be considered has to be found in which components are represented by few global parameters. The advantage of such a formulation compared to a detailed description, is to be independent from the technology. Thus it allows a good understanding of the global system behavior and is more convenient for conceptual improvements. As the level of analysis is induced by the level of the model, the appropriate type of this latter has to be chosen among classical formulations by applying the following criterion: provided that it remains sufficiently representative (allowing the major phenomena representation), it has to be as simple as possible (to give a global description). In dynamics, among the usual formulations: lumped mass-spring modelling, multi-body modelling and finite element modelling, we decided to select the second type which verifies best this criterion (except, of course, regarding airbags) and for its well recognized capabilities, the MADYMO program (Wismans 88, Huibers 94).

DEFINITION AND FITTING OF A BASIC SYSTEM

After the type of model is selected, it becomes necessary to fit it into the reality of tests. This is primarily achieved for a certain "basic" system which corresponds to an almost complete reduction of the dummy environment. The aim of such a methodology is to provide an objective assessment of the major parameters (of the system to be considered), which are then isolated as much as possible.

Definition

This typical system is illustrated figure 5. It includes: A rigid sled with an initial velocity of 49 km/h, decelerated by a mid-severity crash pulse given figure 1, an Hybrid III 50th percentile dummy, a standard seat (no anti-submarining device), a 3-point belt with retractor and webbing graber, and a rigid toe-pan (the feet are linked by rubber bands).

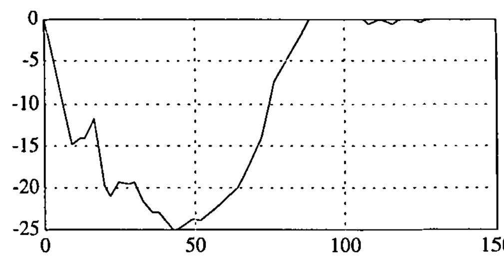

Figure 1. Crash pulse (ms - g).

Discussion on the fitting

The fitting of the corresponding model has been accomplished by making geometric measurements on the sled and tuning a lot of parameters according to our experience (Goualou 93).

Global kinematics - Figure 5 gives the position of the dummy at 3 different stages: Starting time, time of maximum pelvis displacement and time of the maximum impact level between chin and torso. We can then observe that the simulation compares well with the experiment regarding quality.

Trajectories - Figure 6 shows that trajectories of several markers are in accordance (we currently use a criterion to qualify the safety level, based on the maximum head displacement), except for the return of the dummy. We can explain this difference by the fact that in reality, the friction law on the seat pan is different during this stage from the one declared which corresponds to the forward motion of the pelvis (this distinction can not be introduced in the model).

Accelerations - Next 3 figures give the time-acceleration histories of the dummy (projections are done in local frames).

• Pelvis: The maximun levels are more or less correlated (a criterion is based on the resultant maximum level). But it turned out to be difficult to better reproduce the aspect of the longitudinal x-component. We attribute this problem to a more effective deceleration induced by the lap belt early in the experiment, which is followed by a transversal slip on the dummy (this phenomenon can not be reproduced by the current model).

• Upper torso: The curves are similar but the simulated longitudinal acceleration is a little too late and high compared with the experiment (no criterion at this level).

• Head: The x-longitudinal acceleration curves are in accordance. They both present a peak around 110 ms of 65g due to the chin impact on torso (as this latter also generates the maximum level of angular acceleration, it can be considered as an indicator of a too sudden restraint system). Concerning the vertical z-component, a computed smooth curve is obtained, which is reasonnably representative of the real one.

Belt tensions - Figure 10 reveals the good correspondance for the time-force histories of every belt segments (a criterion based on the lower shoulder part is used). Every information was measured during the test in order to precisely evaluate the parameters governing the behavior of a 3-point belt. Note that the shoulder belt unloads too soon in the simulation mainly because the sternum of the dummy is not modeled (for compatibility with other models).

Final remarks - For the given model, the results shown here, represent the best global fitting, i.e. an improvement of a particular output would signify a worse correlation for the others. However, as mentioned in parenthesis before, we have stressed the fitting on the outputs associated with our safety criteria set (to be defined further).

OPTIMIZATION OF A CURRENT SYSTEM

At this stage more or less sophisticated systems can be devised on the basis of the previous minimal definition, with the final purpose of evaluating design optimal solutions and inherent potential safety benefits by an optimization-simulation coupled procedure.

Design parameters

The design parameters set corresponds to the global caracteristics of usual or new components (regarding the basic system). They can be seen as degrees of freedom in the definition of the system.

Note: As optimization methods generally do not treat integers, "options" on the system have to be substituted by equivalent real parameters.

Objective function

Theoritical definition - We considered a single objective function approach, which represents a kind of global safety index (Goualou 95). It is built with the following formula:

$$F_{obj}(x) = \gamma \sum_{i=1}^{n} \beta_i[\alpha_i \max(0,(c_i(x)-c_{mi}))]^2 \qquad (1.)$$

Where x is the design parameters vector, n the number of criteria, c_i the ith criterion to be considered, c_{mi} the minimum or significant level of the criterion, α_i a scaling coefficient, β_i a user adjustment coefficient and γ a global scale factor.

This formulation can be seen as a usual penalty function of each criterion: When the level is below the significant value C_{mi}, it does not contribute to it (in addition such a definition implies some interesting mathematical properties).

Practical definition - The following step consists in devising the coefficients of the formula regarding some problems about the criteria sensitivities (to balance) and their relative dependancies. The criteria set we used in this study, corresponds to the HIC and the maximum of the following outputs (3ms filtered for accelerations and forces):

• HIC: Even if this criterion was introduced for head impacts only, we are used to consider it as a good indicator for its dynamics. In fact without impact, it is generally consistent with the maximum of the head z-component acceleration.

• Head angular acceleration: This criterion reflects the risk of rotational brain injuries. It is commonly consistent with the maximum of the head x-component acceleration and scarcely consistent with the HIC.

• Inferior torso compressive force: This in-house criterion corresponds to the projection of the lower shoulder belt segment force on the torso (local x-longitudinal axis). This force in conjunction with the equivalent one from the upper part belt segment (not considered because also applied on the shoulder) is responsible for the thoracic compression.

• Pelvis acceleration: This criterion is introduced to globally protect the lower limbs from hard impacts.

• Knee impact force: In the simulation, we prefer to consider the local forces applied on knees than the classical femur load. We observe that this latter is generally below the tolerance level and does not reflect sufficiently the severity of impacts (on dashboards and steering-columns).

• Head displacement: This criterion represents the risk for the head of striking hard components. It permits a qualification of a restraint system regardless of the associated parameters (like parameters of the steering-wheel for example).

• Risk of submarining: We consider a geometric criterion consistent with several studies on this subject (Haland 91, Nilson 93).

Optimization method

For the mathematical optimization procedure, we currently use the simplex method of Nelder and Mead (Nelder 65), according to the specifications of our potential problems:

• General: No constraints on variables except simple bounds; Low required levels of precision for the solution (concerning the objective function and variables);

• On variables (design parameters): Small dimension; Real variables;

• On the objective function (versus the design parameters): Of implicit nature; Evaluated on simple précision; Of convexity nature (local and global) not assured ; Not everywhere differentiable; Presents some noise (at differents levels), which does not allow a reasonable gradient evaluation.

The simplex method is only based on function comparison on a finite set of points in design space. If it is generally less efficient than other methods like Powell conjugate directions method for smooth functions, it proves to be very robust and more convenient for our purpose (Goualou 95).

Note: for precise constraints, the complex method can be used (Box 87).

Computing requirement

In order to be able to perform the optimization process, a complete automatic environment illustrated figure 2, has been built around MADYMO.

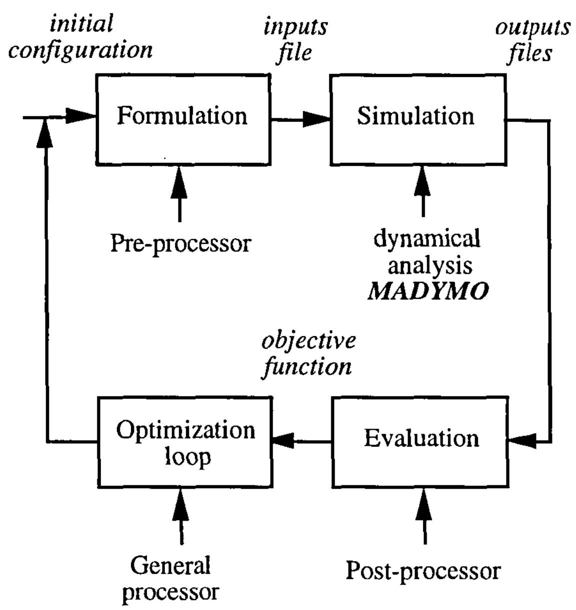

Figure 2. Computing organization.

Example of system optimization

<u>Definition of the problem</u> - In this example, we considered a system based on the previous minimal definition and the following design variations:

• Belt: webbing stiffness and friction coefficients at the buckle and D-ring are considered as parameters. This corresponds to the adjustment allowed here for the existing system.

• Anti-submarining bar: This classical component is added to the previous system. It is introduced in the model as an ellipsoid (with fixed dimensions). The associated parameters are: The position of the principal axis, i.e. the X-longitudinal and Z-vertical coordinates (the origin of the coordinate system is the H point); The stiffness caracteristics assuming the global conceptual behavior illustrated figure 3, i.e. the maximum force F and the limit of elastic deflexion δ.

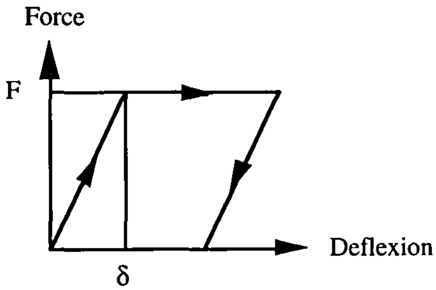

Figure 3. Conceptual behavior of the component.

• dashboard: This component is introduced to describe a real car environment. The same set of parameters as for the bar is used except for the Z positionning.

• Note: The initial definition of the system to be optimized is identical to the one attached to the basic system, i.e. the anti-submarining bar and dashboard are introduced but without initial stiffnesses (however, hand-dashboard impacts are taken into account).

__Results__ - Figure 4 illustrates the decrease of the objective function for the descent steps achieved by the algorithm. The total computation represents 220 iterations (3 hours on a workstation). We can comment the associated results which are detailed in table 1, in the following terms:

• The process did not manage to diminish the thoracic risk (such an improvement would require in fact the introduction of a pretensionner);

• The introduction of a firm bar clearly eliminates any risk of anti-submarining, it also reduces the global vertical motion of the dummy;

• In addition to the belt and seat contributions, ideally the dashboard has to participe in the dummy deceleration;

• The process appears to have modified the belt parameters in order to control as best as possible the head kinematics (see figures 11-12), i.e. to decrease both the maximum displacement and the HIC.

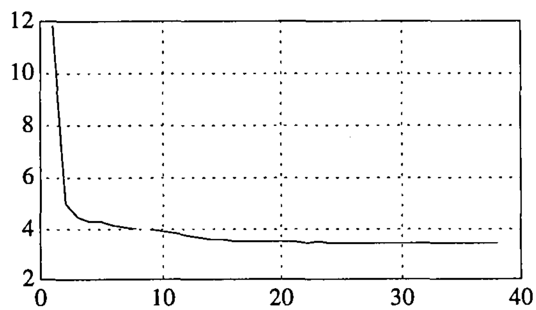

Figure 4. Optimization progress.

CONCLUSION

The methodology presented in this paper can be assimilated to a virtual prototyping for safety systems. Through a suitable global formulation of the problem, we have shown that it is possible to access to an optimized solution regarding the frontal crash concern. The main limit of such an approach is certainly the difficulty of interpreting this virtual solution in terms of technological parameters. It implies that correspondances between the different possible levels of analysis have to be established. In relation with the capabilities of this methodology, we feel confident in a close future to investigate more general systems (in particular with differents types of dummies).

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ACKNOWLEDGMENTS

The authors would like to thank Ms I. Nicolas for reviewing this paper.

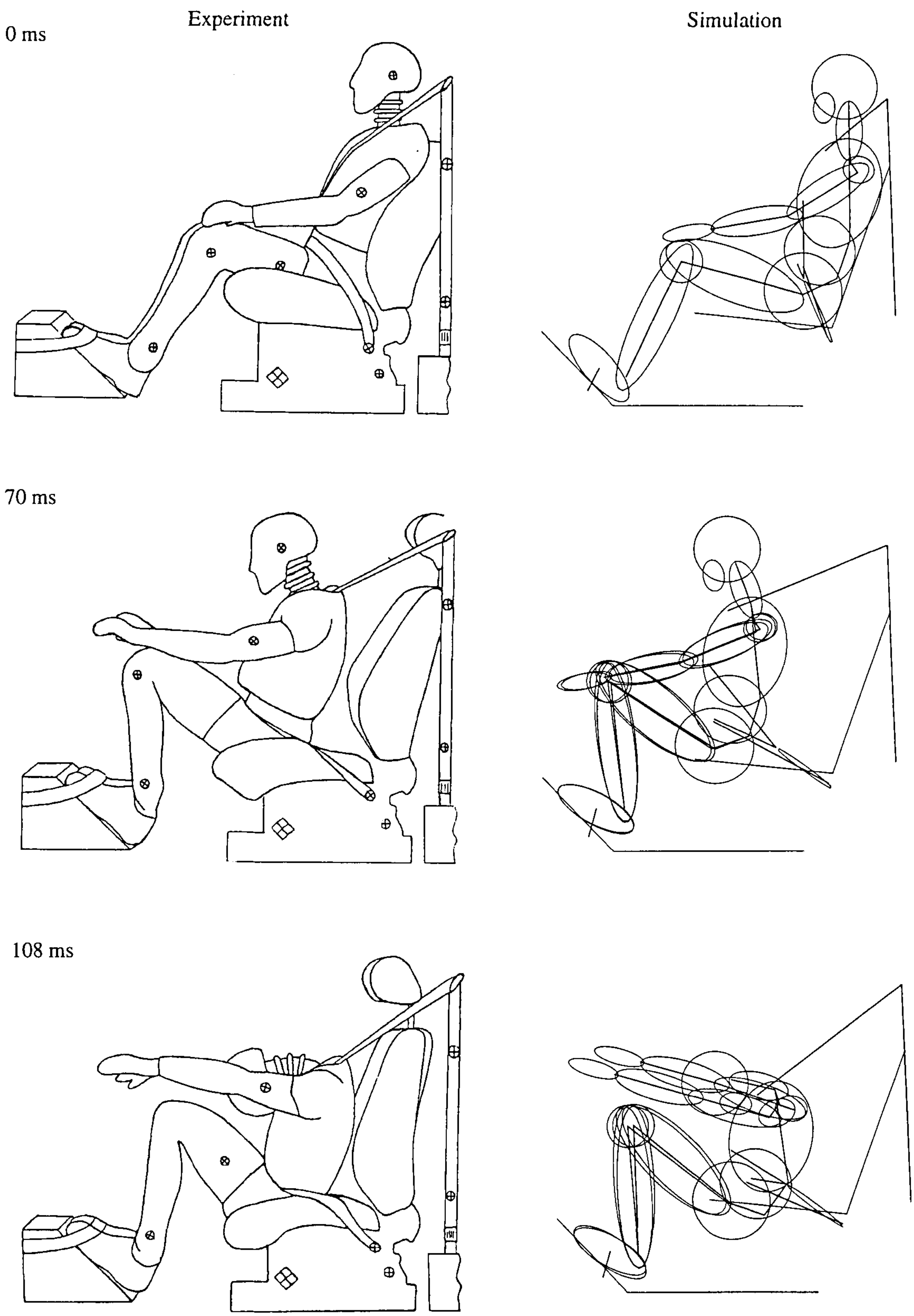
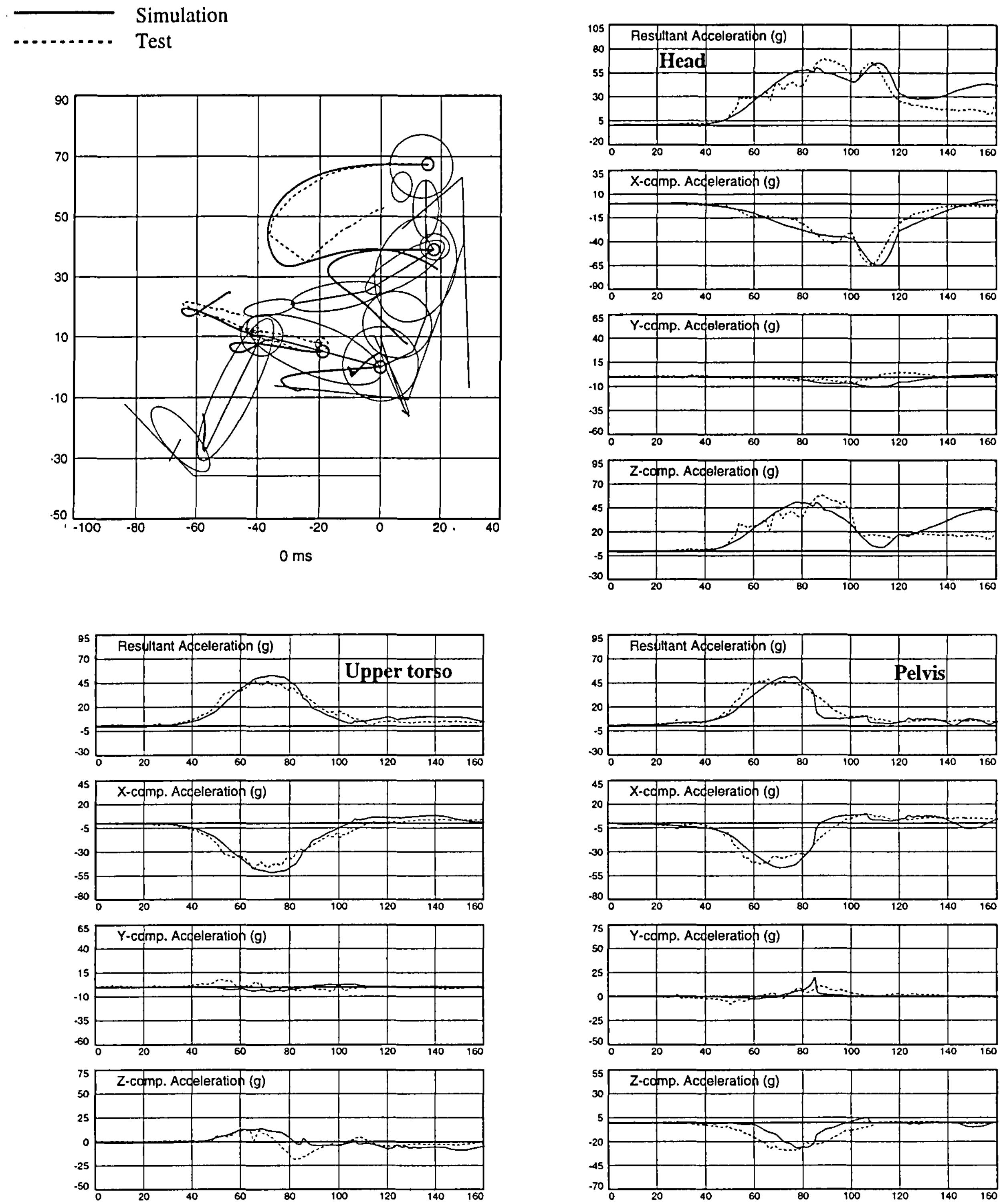


Figure 5. Compared kinematics for the basic system.



Figures 6-9. Compared trajectories and time-acceleration histories for the basic system. Notes: The straight line for the head experiment trajectory is due to the movement of the arm which hides the target; For all the curves, the time is expressed in ms; The x, y, z components correspond respectively to the local longitudinal, transversal and vertical axis.

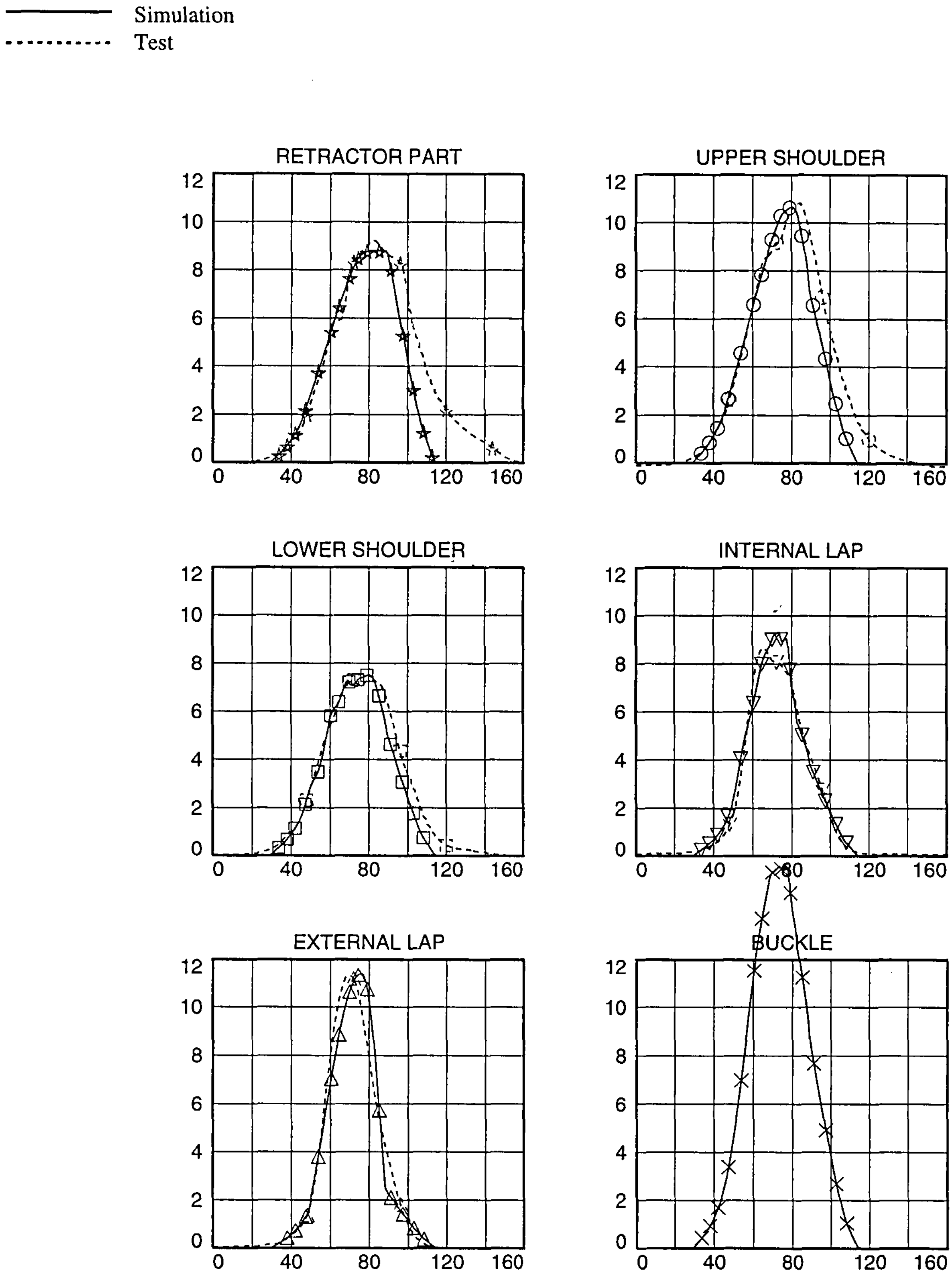
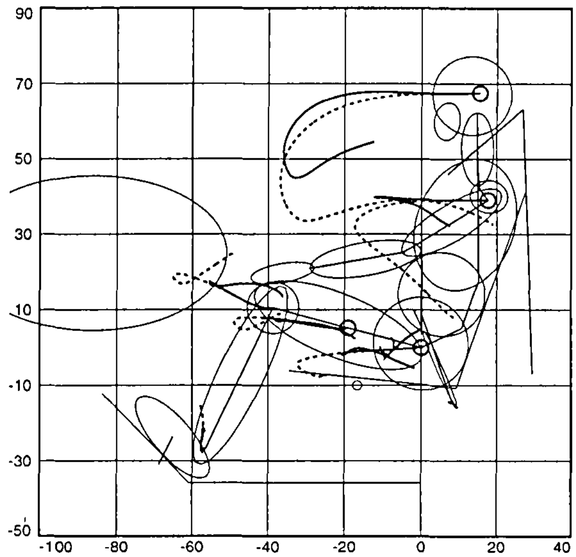
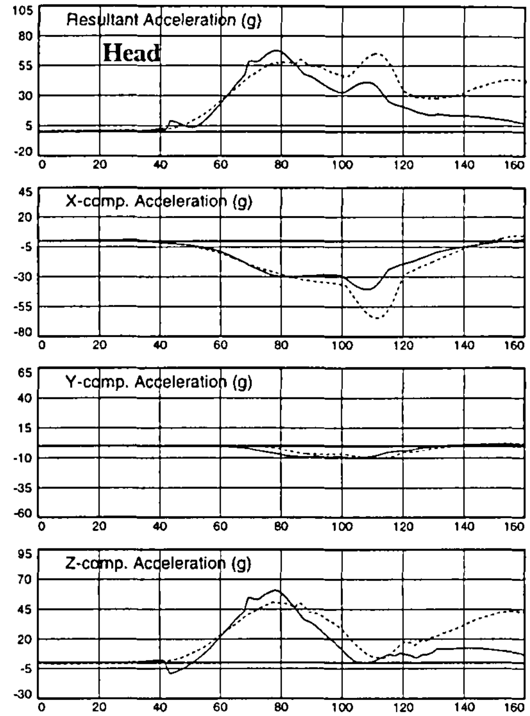


Figure 10. Compared time-belt forces histories for the basic system. The units are respectively the ms and the kN.

Optimized system
Initial system





Figures 11-12. Compared trajectories and time-acceleration histories for the initial and optimized definitions of the system. Notes: For all the curves, the time is expressed in ms; The x, y, z components correspond respectively to the local longitudinal, transversal and vertical axis.

Table 1.
Optimization results

	limit	unit	initial	optimized	variation
HIC (36 ms)	1000	-	937	699	-238
Head angular acc.	2500 ?	rad.s-2	4078	3826	-252
Torso inf. force	6000	N	5820	5800	-20
Pelvis acc.	75 ?	g	50	54	+4
Knee force	5000 ?	N	0	3073	+3073
Head displacement	600*	mm	575	522	-53
Submarining risk	0.2	-	0.36	-0.39	-0.75
	parameter				
Webbing	stiffness	%	12	7	
Buckle	friction	-	0.1	0.2	
D-ring	friction	-	0.1	0.3	
Anti-submarining bar	F	N	0	7000	
Anti-submarining bar	δ	mm	15	11	
Anti-submarining bar	X	mm	220	166	
Anti-submarining bar	Z	mm	-100	-100	
dashboard	F	N	0	3000	
dashboard	δ	mm	50	48	
dashboard	X	mm	900	862	

*conventionnal

Technical Session 4

Vehicle Aggressivity and Compatibility for Occupant Protection
Chairperson: Bernd Friedel, Germany

NHTSA's VEHICLE AGGRESSIVITY AND COMPATIBILITY RESEARCH PROGRAM

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Paper No. 96-S4-O-01

ABSTRACT

The National Highway Traffic Safety Administration's vehicle aggressivity and compatibility research program explores the global evaluation of vehicle crashworthiness designs as a means of minimizing injuries in the design vehicle while simultaneously minimizing injuries in the vehicle's collision partners. The program pursues both an analytic investigation of fleet wide vehicle performance as the basis for global optimization and pursues an experimental component as the foundation for validation of computer models and tools. This paper presents an overview of this research program along with a summary of the results achieved to date.

INTRODUCTION

The crashworthiness performance of passenger vehicles traditionally has been evaluated on the results of well defined laboratory crash tests. These tests, by their nature, focus on evaluating and minimizing injuries to the occupants in the subject vehicle. However, pursuing an optimal crashworthiness performance without regard to the crashworthiness performance of the collision partners can lead to very aggressive, incompatible vehicle designs. Particularly, design modifications which minimize injuries in one vehicle have the potential of actually accentuating injury levels in the vehicle's collision partner.

The purpose of this research program is to investigate the problems of vehicle compatibility in multi-vehicle crashes. The initial focus of the program is to identify and characterize compatible vehicle designs that will result in correspondingly large reductions in crash related injuries. While not a new idea, both the National Highway Traffic Safety Administration (NHTSA) and other international government agencies have recently renewed efforts to study compatibility as a means of reducing crash-related injuries below those levels achievable by equipping the fleet with safety belt and supplementary air bag restraints.

In the United States, there are several important research initiatives which are considering compatibility specifically and the overall fleet wide crashworthiness of vehicles more generally. First, within the last three years, the NHTSA Motor Vehicle Safety Research Advisory Committee's Crashworthiness Subcommittee established a special working group on Vehicle Aggressivity and Fleet Compatibility. This working group was established as a result of the concern about the structural modifications being made by vehicle manufacturers in response to frontal offset crash testing being conducted throughout the world. These modifications included strengthening the vehicle structure in order to reduce the level of intrusion observed in the offset crashing. The stiffened structures have the potential of increasing the severity of side impact crashes. Also, there has been a recent concern over the increasing use of light trucks and vans (LTVs) as personal use vehicles. In the United States, LTVs have accounted for over one-third of new vehicle purchases. This group of vehicles generally is heavier and has stiffer structures than the passenger cars. The working group is developing a system model for evaluating vehicle crashworthiness on a fleet wide basis with the goal of identifying desirable vehicle characteristics for the various vehicle types and weight classes that will lead to improved fleet performance in the fleet crash environment. A second initiative in the United States is the Partnership for a New Generation of Vehicles (PNGV) program, which has the goal of developing new technologies to triple fuel economy and to reduce exhaust emissions while maintaining crashworthiness performance. To achieve this goal, it is anticipated that a forty percent reduction in vehicle weight may be required in the PNGV vehicles. The introduction of such a downsized vehicle could lead to a safety problem due to the mass differences. Also, power train technologies are under consideration that may result in vehicle construction that is different than that utilized for present day vehicles. Finally, the Advanced Research Programs Administration (ARPA) of the Department of Defense is directing the development of new generation of electric vehicles (EVs) which will meet a strict zero emission vehicle criteria. Some of the vehicles under development have ultra-light and ultra-stiff structures which may prove extremely aggressive. ARPA and NHTSA are conducting a joint program to evaluate the crashworthiness of these EVs.

The NHTSA's vehicle aggressivity and compatibility program explores the global evaluation of vehicle impact designs as a means of minimizing injuries in the design vehicle while simultaneously minimizing injuries in its collision partners. The program pursues both an analytic investigation of fleet wide vehicle performance as the basis for global optimization, and an experimental component as the foundation for validation of computer models and tools. This paper presents an overview of this research program along with a summary of results achieved to date.

PROGRAM GOALS

The goals of the agency's research program are two-fold. The near-term goal is to identify and demonstrate the extent of the problem of incompatible vehicles in multi-vehicle collisions. The focus of this goal is to identify and characterize compatible vehicle designs with the overall objective that improved vehicle compatibility will result in correspondingly large reductions in crash related injuries. Based on the findings of the near-term efforts, the longer term goal will be to support improvements in vehicle compatibility. The longer term goal is develop test procedures that evaluate vehicle aggressivity and compatibility and that would lead to the development of appropriate countermeasures that reduce the aggressivity and increase the vehicle compatibility.

The objectives of the first phase of research are to identify those vehicle structural categories, vehicle models, or vehicle design characteristics which are relatively incompatible (i.e., too "hard" or too "soft") based upon accident statistics and crash test data; to develop a comprehensive computer simulation package for the system-wide crashworthiness evaluation of vehicle structures and occupant restraints; and to experimentally and analytically demonstrate the relationship between occupant injury and vehicle structural compatibility.

GENERAL TASK DESCRIPTIONS

This program is composed of the following six tasks: problem definition; global safety systems optimization model; frontal-side compatibility; compatibility of low mass, ultra-stiff electric vehicles; evaluation of compatible crush zone; and geometric compatibility. These are described as follows:

Problem Definition

Accident data is being examined to determine the extent of the aggressivity and compatibility problem and to explore the relationship between vehicle design and fleet compatibility through correlation of accident statistics and vehicle design parameters (e.g., hood profile, mass, and frontal stiffness) extracted from crash test data or physical measurements. For this study, aggressivity is defined to be the number of fatalities/injuries in the vehicles struck by the subject vehicle divided by the number of subject vehicle registrations. This metric will measure the probable outcome in the struck vehicle, given that a multi-vehicle accident has occurred with the subject vehicle. The subtasks include:

Aggressivity Ranking - This task is to use the Fatal Accident Reporting System (FARS) and the National Accident Sampling System (NASS) accident statistics databases to rank all passenger vehicles, cars and LTVs, by their relative aggressivity. The results of this rating will be examined to determine the relative aggressiveness of different vehicle body types, to quantify the effect of weight incompatibility, and to search for differences in aggressiveness among vehicles of the same weight class.

Development of Alternate Aggressivity Metrics - One obstacle to quantifying the aggressivity of a vehicle is the lack of an accepted measure of aggressivity. As previously mentioned, aggressivity has been defined as the number of fatalities/injuries in the vehicle struck by the subject vehicle divided by the number of subject vehicle registrations. Several improvements to this measure have been proposed. This task is evaluating several variations on the base aggressivity metric to include the effect of "other car" restraint usage, normalizing on accident severity, and restricting the metric to prescribed accident modes, e.g., frontal-side impacts and frontal-frontal impacts.

Correlation of Vehicle Design vs. Aggressivity - This task is examining the relationship between vehicle aggressivity and measurable vehicle design parameters. The study is focusing on mass, geometrical, and structural aggressivity factors. Geometrical factors include the hood profile, sill height, and bumper height. Structural factors include the frontal stiffness as determined from crash tests and engine location (transverse right or transverse left). Structural stiffness are being determined from SISAME model syntheses from frontal-barrier crash tests [1]. Sources of geometrical data will include the NHTSA Vehicle Parameter Database [2].

Frontal-Side Compatibility vs. Side Impact Injury - The task is examining the relationship between compatibility and occupant injury in side impacts. In the early 1980s, the Vehicle Research and Test Center (VRTC) conducted an extensive side impact crash test program in which Volkswagen Rabbits were side struck with modified moving deformable barriers (MDBs). This program showed a strong correlation between occupant response and MDB stiffness and profile. This study will determine if the accident statistics support a similar finding [3].

Door Sill-Bumper Height Incompatibility - This task is examining the effect of incompatibility between bumper height/vehicle hood profile and door sill height. This task was begun in 1990 at Volpe National Transportation Systems Center (VNTSC) under sponsorship of NHTSA. The effort investigated the correlation between occupant injury as reported in NASS and vehicle sill-bumper mismatch as inferred from vehicle specification sheets and measurement.

Global Safety Systems Optimization Model

In this task, a large scale systems model is being developed to evaluate vehicle crashworthiness based on the safety performance of the vehicle when exposed to the entire traffic accident environment, i.e., across the full spectrum of expected collision partners, collision speeds, occupant heights, occupant ages, and occupant injury tolerance levels. Optimal crash countermeasure designs must successfully balance two potentially conflicting objectives: (1) maximizing passenger protection in the vehicle under design, and (2) optimizing compatibility with other vehicles in the fleet mix. To meet these objectives, vehicle crashworthiness should be evaluated, not just on the basis of a few test configurations or test speeds, but also on the safety performance of the vehicle when exposed to the entire traffic accident environment; i.e., across the full spectrum of expected collision partners, collision speeds, occupant heights, occupant ages, and occupant injury tolerance levels. Note that, as in the real world accident environment, this will expose the design vehicle both to vehicles less compatible and vehicles more compatible with the design vehicle.

The means of evaluating vehicle crash performance on a system-wide basis was first accomplished by the Safety Systems Optimization Model developed by Ford Motor Company and later enhanced by the University of Virginia [4,5]. Starting with SSOM as a foundation, the VROOM (Vehicle Research Optimization Model) computer model, as proposed below, will take full advantage of recent dramatic improvements in vehicle and occupant models, newly developed injury criteria, and a comprehensive projection of the accident environment for the years 2000-2005. Where possible, VROOM will also explore the feasibility of implementing promising algorithms from the Volkswagen ROSI system-wide optimization model [6].

In this task, a large scale systems model is being developed for global evaluation of vehicle impact designs as a means of minimizing injuries in the design vehicle while simultaneously optimizing performance in crashes with its collision partners.

Enhanced Vehicle Models - During the last few years, the availability of the DYNA-3D finite element (FE) code has triggered a revolution in vehicle and occupant impact modeling. Unlike the lumped-mass models traditionally used in crashworthiness research, DYNA-3D models allow the complex dynamics of vehicle structural impact to be described with uncompromised detail and simulated with vastly improved fidelity to real world crash events. Both the vehicle manufacturers and NHTSA have comprehensive efforts underway to develop increasingly complex FE models of vehicle structures, occupant restraints, and occupants.

The objective of this task is to incorporate these promising new vehicle models into VROOM. The new vehicle FE models will be utilized in two ways. First, the models will be used to study vehicle-vehicle compatibility in a specific accident configurations, with specific collision partners, and specific impact speeds. However, while FE models are potentially very accurate and geometrically fidelic, FE models are prohibitively expensive to execute for global design optimization. A typical VROOM run requires 100,000 simulations. Even at an unrealistically fast 8 hours for a FE simulation of a car-to-car accident on a parallel machine, an optimization based exclusively on FE models would require nearly 800,000 hours (nearly 100 years) to complete.

The second application for the FE models will be to generate sophisticated, and faster running, lumped mass models for optimization. Optimization using lumped mass models will provide broad design directions (e.g., double the aft frame stiffness) for improved crashworthiness. After optimization, these lumped mass results can be used to design modified vehicle components and corresponding FE models for an optimized structure.

Because only a limited number of validated FE models are currently available to NHTSA, VROOM will initially consist of a mix of FE models and the more traditional lumped-mass models extracted from crash test data. To enable near-term analyses with VROOM, initial efforts are being focused on constructing lumped-mass SISAME models of late model passenger vehicles. Simultaneously, this task is developing and adapting FE vehicle models for use in VROOM. Specific subtasks are as follows:

1. **Develop Generic Models of Late Model Year Vehicles** - This task will construct DYNA-3D and SISAME models for generic vehicles in each of the five VROOM weight categories: <2,000 lb, 2,500 lb, 3,000 lb, 4,000 lb, >5,000 lb. A FE and a lumped mass model(s) are being developed for the reference vehicle, the Ford Taurus, to be suitable for use in simulating frontal-barrier, full frontal-frontal, frontal-frontal offset, and frontal-side impacts. Initial lumped mass models will be developed based upon available crash test data. After completion of FE models, enhanced lumped mass models will be extracted from FE simulations.
2. **Develop Models of the PNGV Vehicles** - The Partnership for a New Generation of Vehicles (PNGV) has selected three target vehicles: the Ford Taurus, the Dodge Intrepid, and the Chevrolet Lumina. This task is developing FE and lumped-mass models of these three vehicles.

All models will provide simulation of frontal-barrier, full frontal-frontal, frontal-frontal offset, and frontal-side impacts. The models will be exercised to develop approximating functions for input to VROOM.

Initial lumped mass models are being developed from the available crash test data. After completion of FE models, enhanced lumped mass models will be extracted from FE simulations.

3. **Vehicle-specific Models** - Rather than represent all vehicles with the generic vehicle models described above, this task will investigate the possibility of increasing VROOM accuracy by augmenting the generic models with models specific to high-volume vehicles (e.g., the Chrysler minivan or the Honda Accord). Under this systems model, each high volume vehicle would be characterized by its own model, while less frequently encountered vehicles would be represented by one of the generic models.

4. **Other FE Models** - This task will develop and extend other FE models for use in VROOM and for use as the basis for experimentally evaluating the relationship between aggressive structures and occupant injuries.

Enhanced Occupant Models - Improved occupant models are being constructed for installation in VROOM. Like the vehicle models described above, VROOM will initially consist of a mix of lumped-mass MADYMO models and DYNA-3D models. Initial efforts are concentrated on providing MADYMO models to complement the lumped-mass SISAME models. Simultaneously, modeling development will proceed to construct FE models of both crash test dummies and human occupants.

LTV Models - SSOM was developed in the 1970s when the fleet mix was dominated by passenger cars. Reflecting this fleet mix, the SSOM model accident environment is limited to passenger cars grouped into four different weight categories. This task is extending the VROOM package to include LTVs as well as passenger cars. The LTV segment will be disaggregated into several individual LTV body types to include pickup trucks, minivans, full-size vans, and sport utility vehicles. The following tasks are to be accomplished: (1) update accident statistics for the combined car/LTV fleet, (2) construction of generic vehicle and occupant models for each LTV category, and (3) validation of models against actual accident experience.

Updated Biomechanical Transforms - Injury criteria in SSOM are currently limited to Head Injury Criterion (HIC) and Chest Severity Index. This task will update the biomechanical transforms to include the Thoracic Trauma Index (TTI), pelvic fracture criteria, and lower extremity injury criteria. Injury criteria which are based on occupant age, gender, or stature may also require modification of the accident environment description to include the corresponding probability distributions.

Additional Impact Modes - The accident environment in SSOM is currently described by the majority of potential accident configurations or impact modes. However, a number of less frequent accident modes are not represented in the model. This task will add the following accident modes to VROOM: (a) Front-Rear, (b) Front-Side (oblique), (c) Front-Side (T-type collision), and (d) Side-Roadside object/barrier Collisions.

Additional Collision Partners - This task will add two new categories of collisions to VROOM: Heavy Trucks and Pedestrians.

Improved Accident Statistics - This task is developing a projection of accident statistics for the years 2000-2005 for use in VROOM. One challenging aspect of this task is the development of the distribution of impact speeds based on NASS delta-Vs.

Review of Safety Performance Requirements - Currently, SSOM optimizes vehicle designs without regard to FMVSS regulations, e.g., the FMVSS No. 208 frontal barrier crash test. Conceivably, SSOM could recommend a vehicle design which minimizes injuries but fails a FMVSS requirement. This task will evaluate the relationship between FMVSS regulations and optimal crashworthiness design. Should FMVSS regulations lead to sub-optimal designs, this task will evaluate various countermeasures to produce improved safety performance.

Frontal-Side Impact Compatibility

This task is developing a problem definition statement, and is developing test conditions and test devices for crash tests which explore the effectiveness of increasing compatibility in reducing occupant responses for the side impact crash mode. The objective of this task is to determine the relationship between occupant responses in side struck vehicles and variation in the striking vehicle front-end characteristics.

VROOM Evaluations - The effect of striking vehicle compatibility on side impact injuries will be evaluated using VROOM. The simulation will include both the effect of striking vehicle stiffness, weight, and profile as well as occupant height and age. VROOM evaluations and optimization of the striking vehicle front structure will be conducted to determine the effect of variations in stiffness and profile, and to suggest potential countermeasures.

Dummy Selection - A side impact dummy must be selected for use in side impact crash testing. The SID, BioSID, EuroSID, and SID2S could be considered. The primary criterion for this selection is the suitability of the dummies for this type of testing. Lowering of the striking vehicle profile may produce loading to the dummy below the thoracic region, perhaps in a direction different from that of the dummy's primary response axis. The response sensitivities of the dummies will be examined and compared through HYGE sled testing.

Testing - Based on the compatibility ranking and on the results of the VROOM optimization, at least two values for each of front end stiffness, bumper height, and hood profile will be selected. Up to eight different MDB fronts will be designed and fabricated which combine these characteristics. A side impact test will then be conducted using each of the MDB fronts. The struck car will have been previously tested with the appropriate dummy, and will marginally meet the requirements of the dynamic crash test of FMVSS No. 214. The same struck car will be used throughout this series of tests, and other impact conditions will be as specified in FMVSS No. 214. The results will be used to determine the effect that different striking vehicle front end characteristics have on side struck vehicle occupant responses.

Demonstrations - Based on the results of the aforementioned testing and the fleet characterization, specific front end characteristics will be selected for additional testing. At least two different vehicles will be selected (and modified if necessary) which combine these characteristics. The characteristics chosen will likely represent the upper and lower bounds of the side impact performance spectrum, as well as the optimal characteristics identified in the optimization studies, if different. If it is not feasible to modify existing vehicles to meet the required design requirements, then simulated fronts will be designed with the required characteristics, if different from those tested previously. Several frontal load cell barrier (FLCB) tests to determine front end stiffnesses will likely be required. A side impact crash test will then be conducted using each of these vehicles. The struck car, dummy, and test conditions will be the same as those used in the aforementioned testing. The results from these tests are intended to further validate the findings of the parameter study and the MDB crash tests. If vehicles are used, these tests also help to demonstrate practicability.

Compatibility of Low Mass, Ultra-Stiff Electric Vehicles

This task is to explore the aggressivity of the new generation of electric vehicles being developed under ARPA sponsorship in a joint research program. ARPA is directing the development of new generation of electric vehicles which will meet a zero emission vehicle criteria. Some of the vehicles being developed have ultra-light and ultra-stiff structures which may prove extremely incompatible with the fleet. In Europe, "city" cars are already at the prototype stage which weigh under 600 kg of mass, but are designed with ultra-stiff, ultra-aggressive frontal structures to protect the occupants. Under this task, NHTSA is conducting a joint research program with ARPA to evaluate the crashworthiness and compatibility of EVs.

Evaluation of Compatible Crush Zone

This task is investigating the feasibility of one recent European proposal to mandate front-end stiffness for the first 700 mm of crush as a means of regulating fleet compatibility. More recently, the European Experimental Vehicles Committee has convened a working group (Working Group 15 - Improvement of Crash Compatibility between Cars) to investigate the following topics: overall identification of compatibility problems with respect to injuries and countermeasures, determine parameters which can affect the compatibility, and determine the methods for evaluating compatibility such as the analysis of deformation patterns of deformable elements. This task will analytically investigate the feasibility of setting force-deflection requirements on the fleet by performing a VROOM optimization to determine the optimal force-deflection levels for this crush zone, and determining the expected benefits of a compatible crush zone regulation.

Geometric Compatibility

This task will examine the extent and consequences of geometric incompatibilities. This task will investigate the feasibility of adapting the VROOM methodology to determining the effect of geometric incompatibility. Studies to be conducted under this task include the correlation of door sill height and bumper height, and the correlation between occupant injury guard rail/vehicle frontal profile incompatibility. This task will investigate the use of three-dimensional lumped mass models and FE models to analytically evaluate geometric compatibility.

RESULTS

The aggressivity of a specific vehicle is controlled by its weight, its structure, and the driver behavior. Comparison of vehicle-vehicle aggressivity is challenging because these three factors vary widely between any two given models. However, by comparing only vehicles within a given vehicle category (i.e., subcompact cars, compact cars, midsize cars, large cars, minivans, full size vans, small pickups, full size pickups, and sports utility vehicles), the effects of weight can be minimized. Presumably all vehicles within a single vehicle category are of approximately the same weight.

But just as importantly, limiting the comparison of specific vehicles to within a vehicle category should reduce the complexity of the vehicle to vehicle variation in driver behavior (e.g., speeding). Presumably, the vehicles within a given category are operated by drivers sharing the same demographics. For example, minivans are typically driven for family transportation and the drivers are assumed to share similar driving behavior patterns. Similarly, sports performance cars are assumed to be operated by drivers who share similar driving characteristics.

This study first presents a fleet wide ranking in which all vehicle models are compared. This ranking will be dominated by the overwhelming effect of vehicle weight on the outcome of vehicle-vehicle collisions. In order to investigate structural aggressivity, a fleet wide ranking with model rankings for each vehicle category is presented.

Technical Approach

This initial study uses the Fatal Accident Reporting System Database (FARS) to rank order all passenger vehicles, cars, light trucks, and vans, by their relative aggressiveness. The ranking will show that vehicles of the same weight class and body type (e.g. minivans) will display approximately the same aggressivity. The results of this rating will be examined to (1) determine the relative aggressivity of different vehicle body types, and (2) to quantify the effect of weight incompatibility, and (3) to search for differences in aggressivity among vehicles of the same weight class.

This study examined the 1991-93 FARS database to tabulate, for each vehicle, the number of fatalities in the subject vehicle and in the other vehicle. FARS provides a comprehensive census of all U.S. traffic accident related fatalities. The scope of our analysis was constrained to cars, light trucks, and vans under 10,000 pounds in weight. The focus was further narrowed to two vehicle collisions in which the vehicles were either cars or LTVs in which a fatality had occurred.

The net result of the FARS analysis will be to provide absolute numbers of occupant fatalities resulting from multi-vehicle accidents. To develop an aggressivity risk factor or metric (rather than evaluate the subject vehicles by the absolute number of fatalities in the other vehicle), our study will normalize the absolute number of fatalities in the other vehicle by the size of the subject vehicle population. In particular, we have normalized the number of fatalities in the other car per million registrations of the subject vehicle as shown below:

$$\text{Aggressivity Metric} = \frac{\text{Deaths in Other Vehicle}}{\text{(Total Registrations in Subject Vehicle) / 1,000,000}}$$

Findings

This section presents the findings of the FARS analysis in terms of absolute numbers of fatalities in the other vehicle, as a rank ordering of all vehicles by Aggressivity Metric, as a rank ordering of all vehicle categories, and as a rank ordering of all models within each vehicle category.

Fatalities in the Other Vehicle - The results of the FARS analysis are presented in Figure 1 in rank order by total number of fatalities in the other vehicle for the top 20 vehicles. The fatality totals are an annual average computed over 1991-93. Note that four out of the top five vehicles on this plot are LTVs. LTVs are heavier and structurally stiffer than their passenger car counterparts, and might be expected to perform aggressively in a collision.

However, these absolute numbers must be used with caution, as they have not been normalized by the number of vehicle registrations. Although the Ford F-Series Pickup and the Chevrolet Pickup are the top two vehicles on the list, both trucks are extremely popular and have large populations in the fleet. To more accurately gauge aggressivity, we can not only measure total number of fatalities but divide this total by the size of the subject vehicle population.

Overall Fleet Aggressivity Ranking - In Figure 2, the total number of deaths in the other vehicle has been normalized by the estimated number of vehicle registrations over the 1991-93 time period. This plot is limited to current production vehicle models with at least 100,000 registered vehicles. For this study, a current production vehicle model is defined to be a vehicle model which was in production in the 1991-93 time frame. Note, however, that the totals for each vehicle include all vehicles of each model whether produced during the 1991-93 or earlier.

The most striking feature of Figure 2 is that 19 of the top 20 most aggressive vehicles are light trucks and vans. Of the nineteen LTVs, seven are sports utility vehicles, nine are pickup trucks, two are full-sized vans, and one is a minivan. The most aggressive vehicle of those surveyed was the full-size Chevrolet Blazer with an Aggressivity Metric of 122 other vehicle fatalities per million Blazer registrations. The Chevrolet Blazer is a large sport utility vehicle with an estimated curb weight of 4,700 pounds. The aggressivity of the Blazer is likely due to both its weight and the structural stiffness typical of a sports utility vehicle designed for off-road use.

Only one vehicle in the list of top twenty aggressors is a passenger car. The Chevrolet Camaro, with an Aggressivity Metric (AM) of 86, is a mid-sized performance sports car approximately 3,200 pounds in weight. The aggressivity of the Camaro may be more the result of the way a sports car is driven, rather than due to any structural or weight factor. Other metrics may provide a measure of aggressivity that may better handle effects such as driver behavior. In Table 1, two other metrics besides the measure selected for this study are presented for the identified 20 most aggressive vehicles. These measures are the ratios of (1) the other vehicle fatalities divided by the subject vehicle fatalities and (2) the other vehicle fatalities divided by the subject vehicle fatal accidents. Both of these measures rank the Camaro near the bottom for these 20 vehicles. In other words, the high Camaro fatalities may involve more frequent crashes and the vehicle itself may not be as aggressive as Figures 1 and 2 imply. Figure 3 provides a graphical representation of the measure using the ratio of the other vehicle fatalities divided by the subject vehicle fatalities. In this figure, the percent of the fatalities in the other vehicle and in the subject vehicle are shown. Any value of other vehicle fatalities above 50 percent indicates that the subject vehicle may be more aggressive.

Aggressivity Ranking by Vehicle Type - Aggressivity is a strong function of vehicle weight and vehicle type. Nineteen of the twenty most aggressive vehicles shown in Figure 2 are LTVs. One way to better illustrate the degree of crashworthiness incompatibility within the fleet is to compare the average Aggressivity Metric of the different categories of vehicle types. Figure 4 presents the registrations-averaged AM for each category of light truck, van, and passenger car. The categories assigned to each vehicle are as tabulated by the 1993 Automotive News Market Data Book [7]. Our study groups luxury, near luxury cars, and large cars into a single large car category.

As shown in Figure 4, full-size pickups were found to be the most aggressive vehicle category with an AM = 86. This category was followed closely by Sport Utility Vehicles (AM = 72), full-sized Vans (AM= 67), and Small Pickups (AM=59). Minivans were the least aggressive of all LTV groups with an average AM = 46. The AM of passenger cars was significantly lower and ranged from AM=24 for subcompacts to AM=42 for large cars.

Vehicle weight is not always the overriding factor dictating aggressivity as clearly demonstrated by Figure 4. Mid-sized cars (e.g. the Ford Taurus) and small pickups (e.g. the Toyota pickup) both have approximately the same curb weight of 3,000 pounds. However, mid-sized cars have a modest AM of 39 while small pickups have a dramatically higher AM of 59. We theorize that the higher aggressivity of the small pickup class is due to both its higher structural stiffness and its higher hood and bumper height.

Among cars, the Aggressivity Metric is a strong function of vehicle weight. AM for the large car category (e.g., Oldsmobile 98) is 42 and drops to an AM of 24 for the subcompact category (e.g., the Nissan Sentra). The conservation of momentum in a collision places smaller cars at a fundamental disadvantage when the collision partner is a heavier vehicle. The importance of car size in providing occupant protection has been demonstrated in several studies of the U.S. accident statistics [8].

Aggressivity Ranking by Vehicle Model - This section will compare the aggressivity of different models within a given vehicle category to identify specific vehicle models which vary significantly from the category average.

1. Minivans - The average AM of minivan vehicle category is 46. Figure 5 presents the within category variation for that group. AM varies from a high of 67 for the Chevy Astro Van to a low of 25 for Pontiac Trans Sport. Note the consistency of the AM across corporate twins: the Trans Sport and the Lumina APV are corporate twins and both have AM=25. The Plymouth Voyager and Caravan have AM=36 and 44 respectively. However, we have not controlled for all variations as two other twins, the Chevy Astro (AM=67) and GMC Safari (AM=47), do not have similar AM's. This dissimilarity may reflect the fact that the Chevy Astro Van is a popular cargo van as well as a minivan.

2. Vans and Pickups - Figure 6, 7, and 8 show that only modest variation is observed in the full-sized van, small pickup category, and full-size pickup categories. In the van class, the exception is the VW Vanagon which has AM = 31 significantly lower than the full-sized van average AM of 57. However, the VW Vanagon also has a significantly different structure than other full-sized vans in the group.

3. Sports Utility Vehicles - As shown in Figure 9, the Sport Utility Group displays dramatic variation in aggressivity between specific models. The most aggressive vehicle of those surveyed in the entire car/truck fleet is the Chevy Blazer, a member of the Sport Utility group, with an AM of 122 (71 percent above the group average). The least aggressive of the sport utility vehicles is the Isuzu Trooper with an AM of only 41 (43 percent below the group average). Future studies will explore the vehicle design variations which account for this tremendous variation.

4. Passenger Cars - Figures 10, 11, 12, and 13 show the within-group variation for the subcompact, compact, mid-sized, and large categories of passenger cars. As hypothesized earlier, comparison of cars in this manner should minimize the effect of vehicle weight and driver behavior, and allow the examination of structural differences between models. Within group variation is presented below:

Passenger Car Category	AM Low	AM Hi	AM Avg
Subcompacts	15	52	24
Compacts	18	62	38
Mid-Size	11	86	39
Large	14	61	42

In all but the large car category, the most aggressive vehicle in each car category was a sports/performance car. In the subcompact category, the most aggressive car was the Geo Storm. In the compact category, the most aggressive car was the Ford Mustang. In the mid-sized category, the most aggressive car was the Chevrolet Camaro. It is interesting to note that the large car with the lowest AM (14), the Volvo 240 (weight = 3000 lb) had an aggressivity metric slightly lower than the least aggressive subcompact (AM = 15), the Geo Sprint (<2000 lb). This demonstrates again that vehicle weight is not always the overriding contributor to aggressivity.

FUTURE WORK

This paper has presented the first results of a NHTSA study which is attempting to characterize the problem of fleet incompatibility and vehicle aggressivity. As further steps in investigating vehicle aggressivity, a number of areas for future work have been identified:

Alternative Metrics

One obstacle to quantifying the aggressivity of a vehicle is the lack of an accepted measure of aggressivity. For the purposes of this initial study, aggressivity was defined as the registration-weighted number of fatalities in the 'other' vehicle. In ranking the top 20 most aggressive vehicles, two other metrics (determined by using the ratio of the other vehicle fatalities divided by the subject vehicle fatalities and the ratio of the other vehicle fatalities divided by the subject vehicle fatal accidents) were presented to demonstrate other possible metrics. Several improvements to these measures have been proposed and will be evaluated in future efforts. Proposed variations on the basic aggressivity metric include:

1. Normalizing by number of accidents instead of number of registrations
2. Normalizing for the effect of restraint usage in either vehicle
3. Normalizing for accident severity
4. Examining the metric in prescribed accident modes, e.g., frontal-side impacts or frontal-frontal impacts
5. Examining rollovers and full ejections from either vehicle
6. Limiting the other vehicle fatality count to cases where the subject vehicle was the striking vehicle

Ranking Refinements

NHTSA currently has a research effort under way to further refine the aggressivity ranking presented in this paper. This second phase effort will investigate 1991-95 FARS, and will perform a more refined breakdown of vehicle models. The current study groups vehicle models simply by nameplate. The follow-on study will group vehicles by platform design within nameplate. This will allow the study to capture, for example, any design differences between the 1986 Ford Taurus and the 1996 Ford Taurus.

Correlation of Vehicle Design with Aggressivity

This task will examine the relationship between vehicle aggressiveness and measurable vehicle design parameters. This future task will focus on mass, geometrical, and structural compatibility factors. Geometrical factors will include hood profile, sill height, and bumper height. Structural factors will include frontal stiffness as determined from crash tests and engine location (transverse right or transverse left). Structural stiffness will be determined by extracting discrete element models from frontal-barrier crash tests.

CONCLUSIONS

This paper has investigated the problem of vehicle aggressivity in two-vehicle traffic accidents. Using the other vehicle fatalities per registered subject vehicle as a measure of a vehicle's aggressivity, the examination of U.S. accident statistics shows a striking incompatibility between LTVs and passenger cars crash performance. As measured by this aggressivity metric, LTVs as a class are twice as aggressive as passenger cars. This mismatch in crash performance has serious consequences for the traffic safety environment as approximately half of all passenger vehicles sold in the U.S. are LTVs. The effect of this tremendous degree of fleet incompatibility is not measured directly by frontal-barrier crash tests and will be the focus of future NHTSA research.

The aggressivity metric used in this study provides an initial analysis of the data. Other metrics will be examined in order to control for confounding factors such as driver behavior, crash severity, and other considerations which may affect the fatality outcome.

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Table 1. Top 20 Aggressors

Vehicle	1991-93 Fatal Accidents	1991-93 Subject Vehicle Fatalities	1991-93 Other Vehicle Fatalities	Other Vehicle Fatalities per Million Registered Vehicles	Other Vehicle Fatalities / Subject Vehicle Fatalities	Other Vehicle Fatalities / Fatal Accident
Chevrolet Blazer-full size	213	54	206	122	3.81	0.97
Toyota 4-Runner	94	32	78	105	2.44	0.83
Dodge Dakota	191	71	157	103	2.21	0.82
Jeep Comanche	49	14	44	99	3.14	0.90
GMC Jimmy full size	47	11	49	99	4.45	1.04
Nissan Pathfinder	62	19	48	97	2.53	0.77
Ford F-series PU	2474	606	2341	94	3.86	0.95
GMC C,K,R,V-series PU	599	166	561	86	3.38	0.94
Chevrolet Camaro	601	322	433	86	1.34	0.72
Ford Explorer/ Bronco	329	101	298	82	2.95	0.91
Ford Bronco full size	203	42	191	81	4.55	0.94
Chevrolet C,K,R,V-series PU	2131	601	1942	80	3.23	0.91
GMC G-series Van	117	28	120	80	4.29	1.03
Ford E-series Van	671	140	628	77	4.49	0.94
Ford Ranger	702	337	491	71	1.46	0.70
Isuzu P'up/PU	111	51	75	70	1.47	0.68
Dodge D,W-series PU	384	118	352	67	2.98	0.92
Chevrolet Astro Van	228	78	187	67	2.40	0.82
Chevrolet S-10,T-10 PU	577	315	379	66	1.20	0.66
Chevrolet S-10 Blazer	307	122	249	66	2.04	0.81

FIGURE 1. FATALITIES IN OTHER VEHICLES

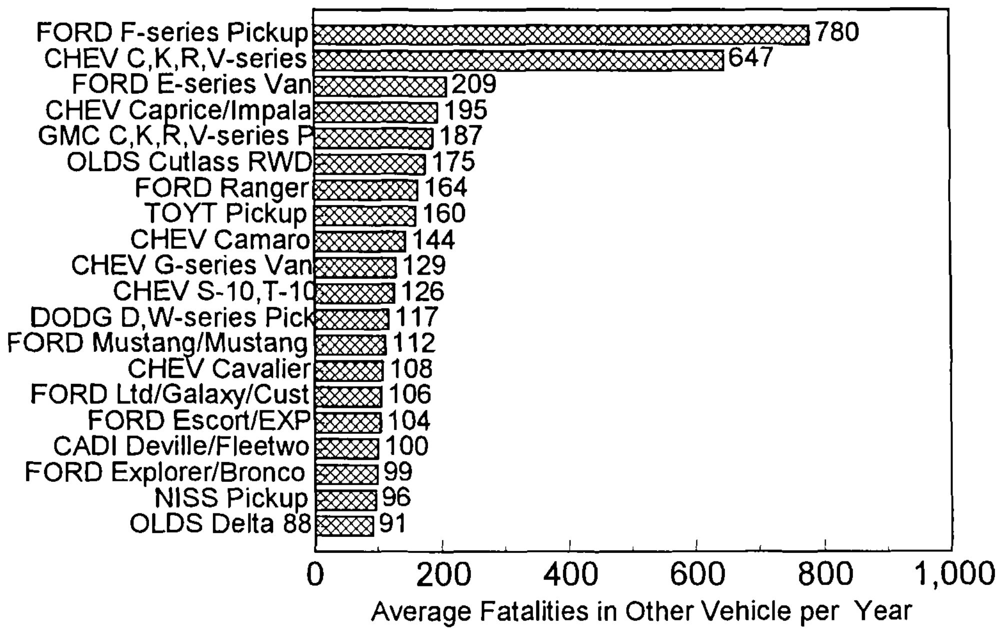


FIGURE 2. AGGRESSIVITY RANKING: TOP 20 VEHICLES

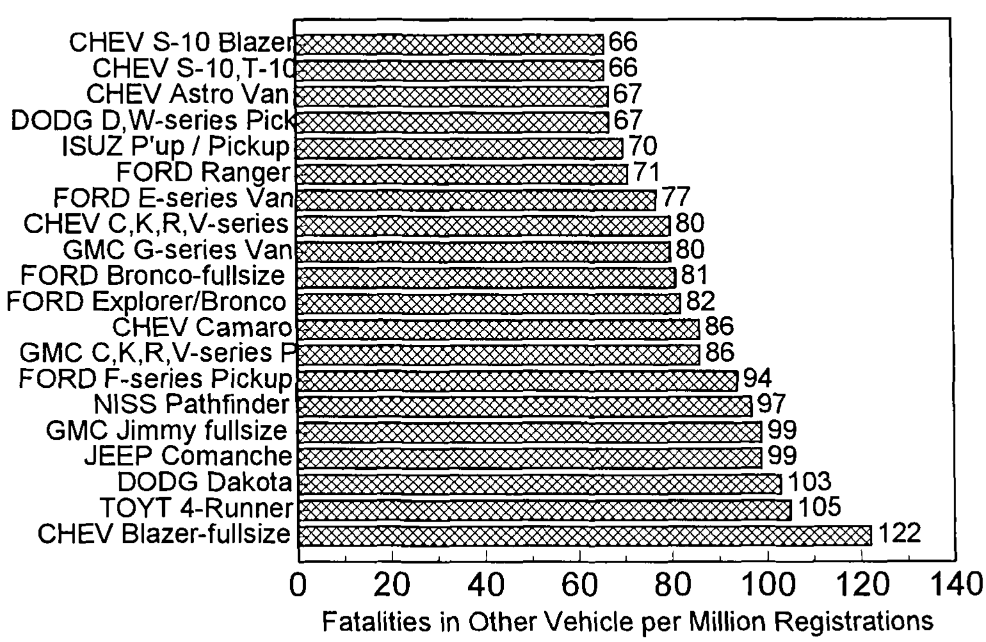


FIGURE 3. TWO VEHICLE COLLISIONS: OTHER VEHICLE/TOTAL FATALITIES

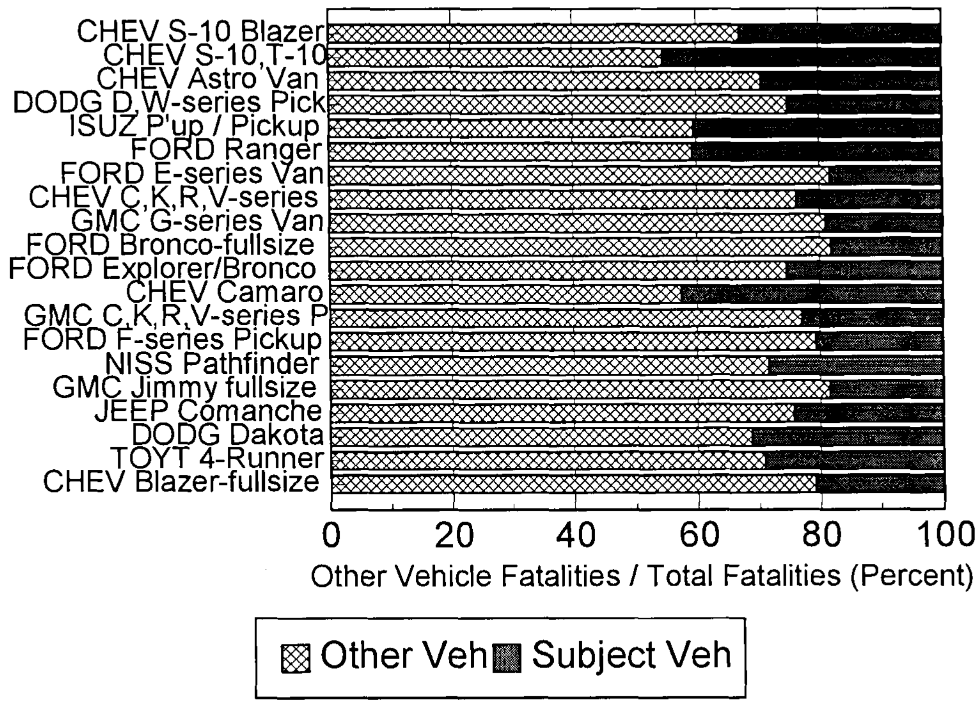


FIGURE 4. AGGRESSIVITY RANKING: LTVs vs CARS

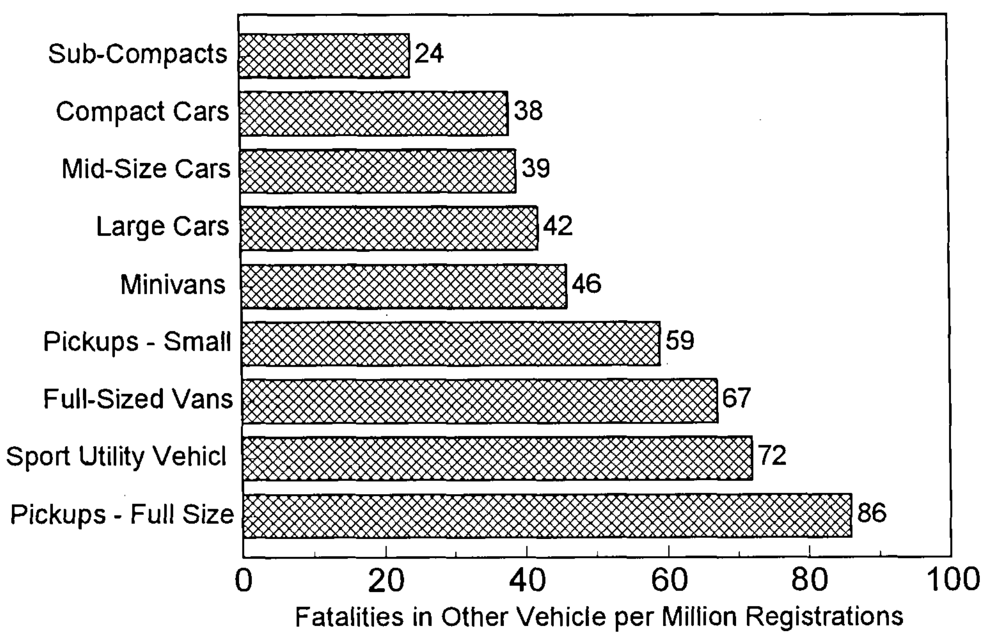


FIGURE 5. AGGRESSIVITY RANKING: MINIVANS

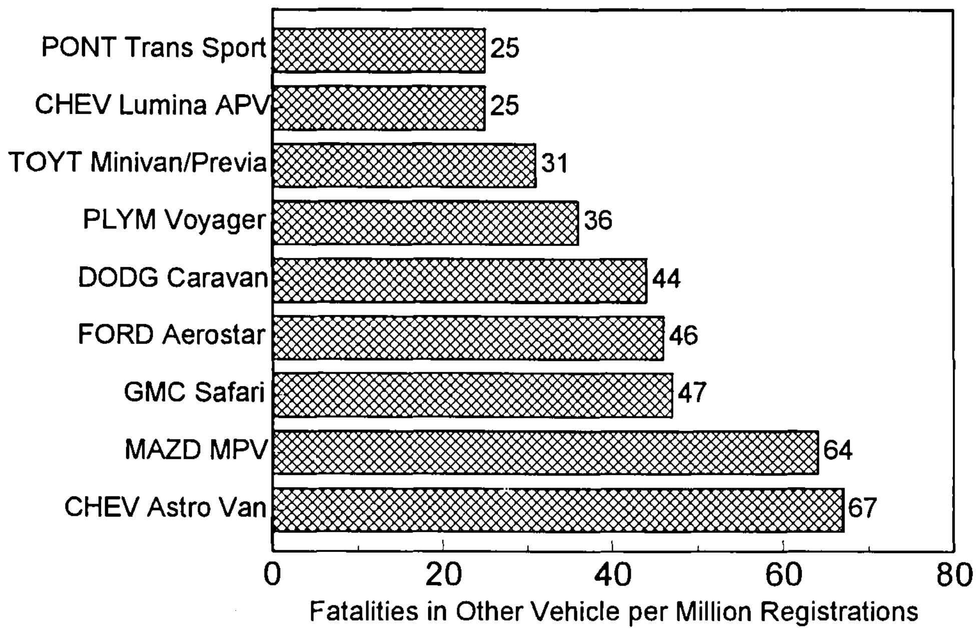


FIGURE 6. AGGRESSIVITY RANKING: FULL SIZED VANS

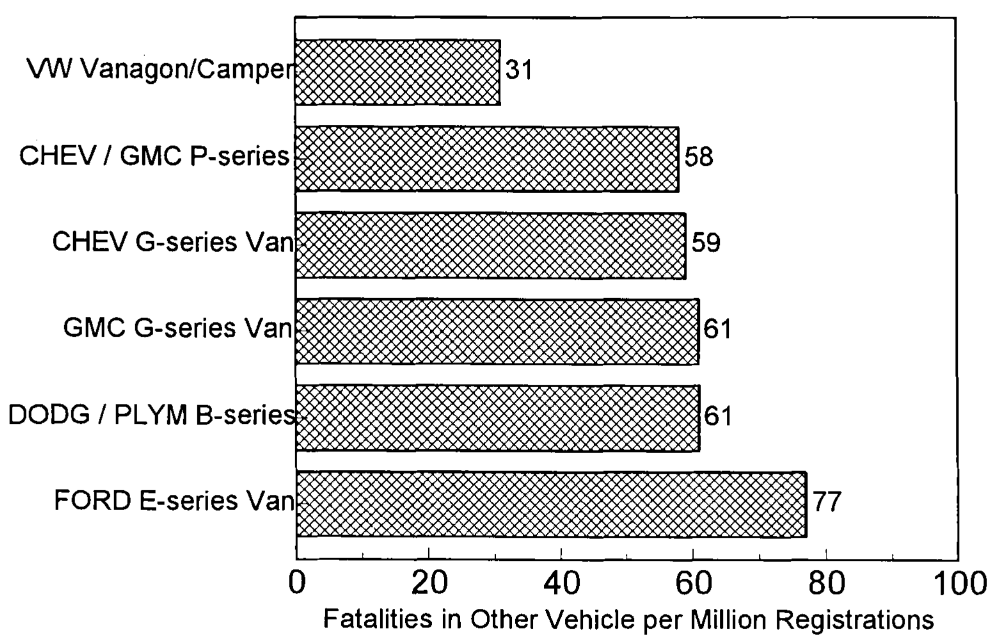


FIGURE 7. AGGRESSIVITY RANKING: SMALL PICKUP TRUCKS

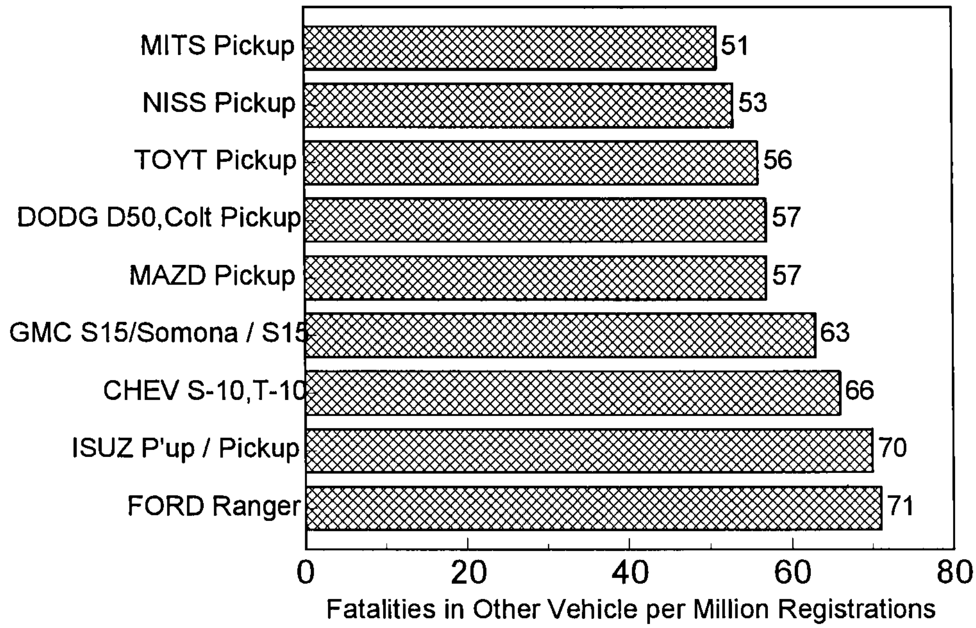


FIGURE 8. AGGRESSIVITY RANKING: FULL SIZE PICKUP TRUCKS

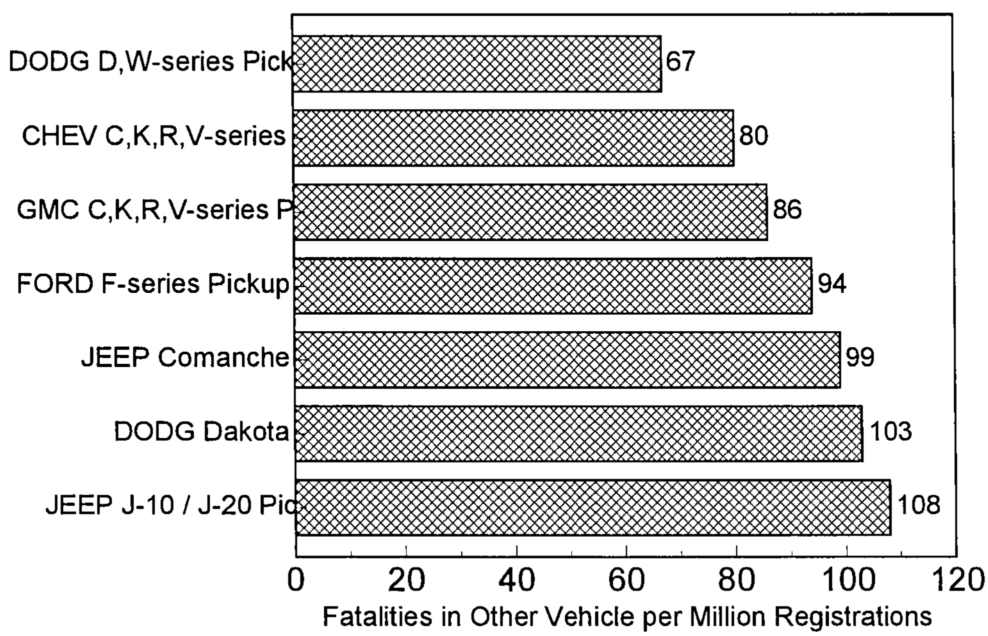


FIGURE 9. AGGRESSIVITY RANKING: SPORT UTILITY VEHICLES

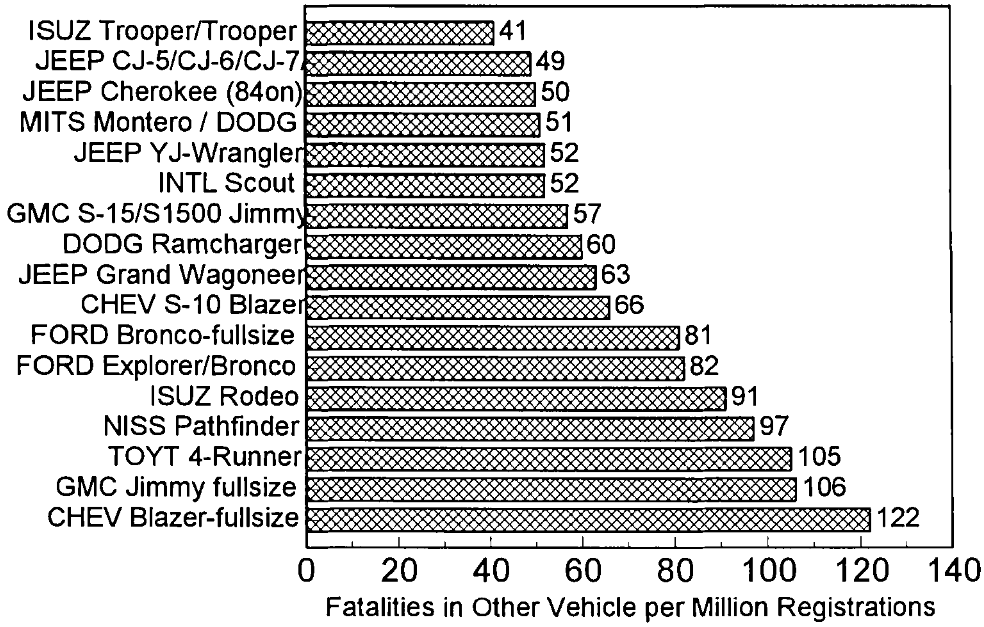


FIGURE 10. AGGRESSIVITY RANKING: SUBCOMPACT CARS

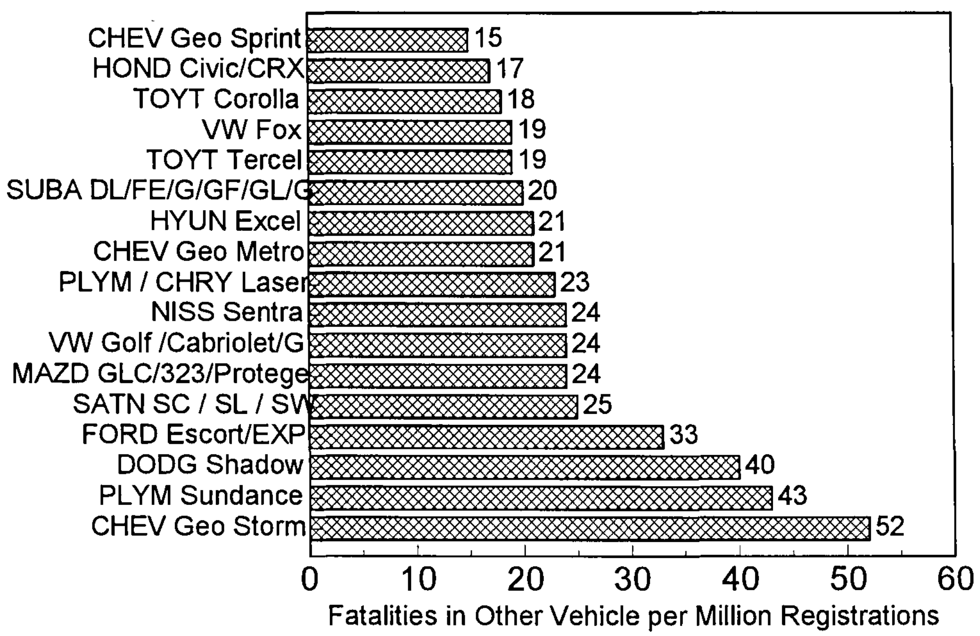


FIGURE 11. AGGRESSIVITY RANKING: COMPACT CARS

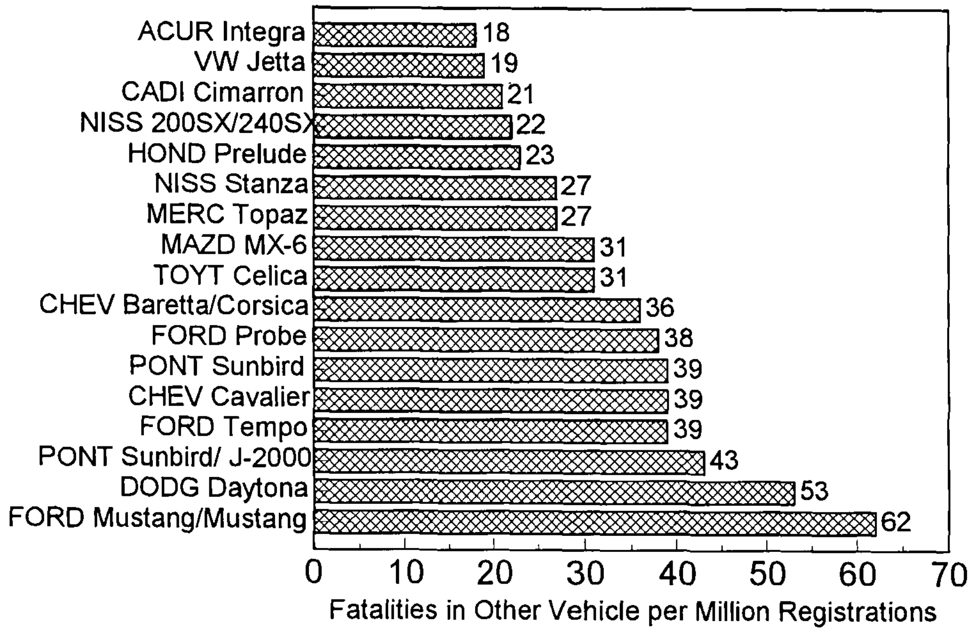


FIGURE 12. AGGRESSIVITY RANKING: MID-SIZE CARS

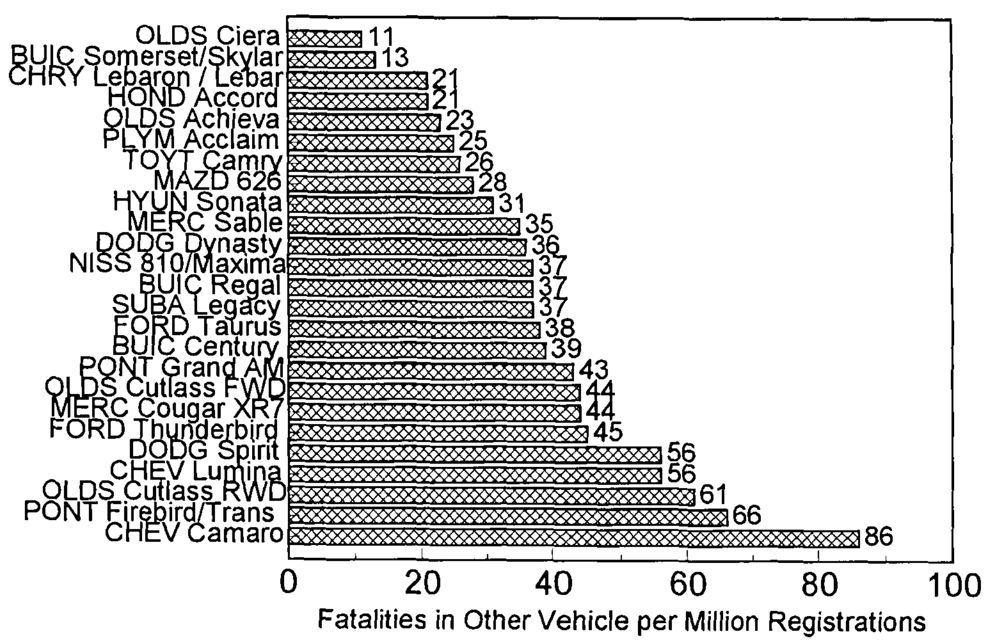
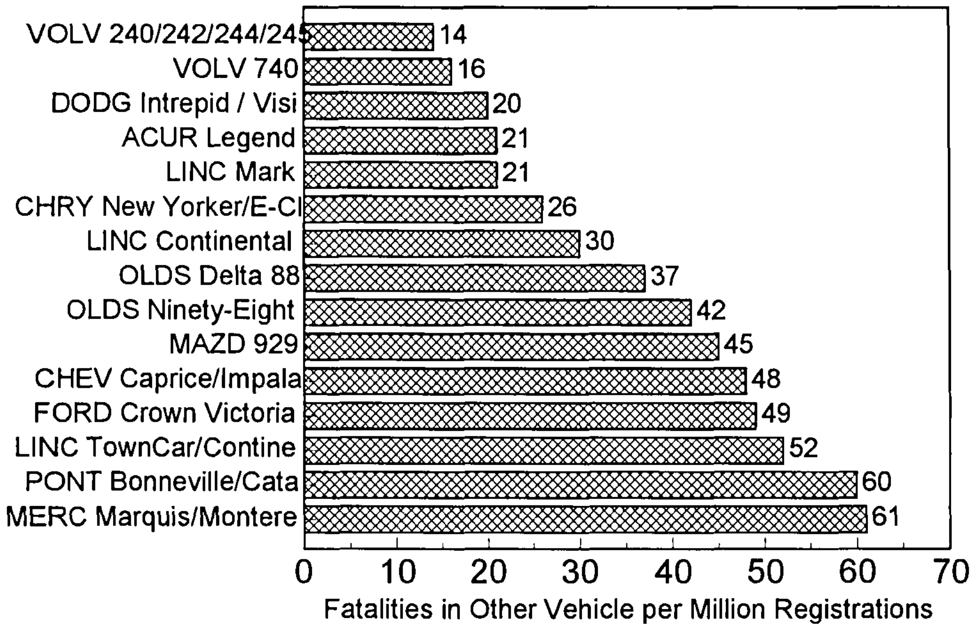


FIGURE 13. AGGRESSIVITY RANKING: LARGE CARS



BUMPER STRUCTURE FOR PEDESTRIAN PROTECTION

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Paper Number 96-S4-O-02

ABSTRACT

In previous studies, foam materials have been used as the energy absorbing structure on automobile bumpers to reduce the severity of leg injuries in pedestrian-to-automobile accidents. One obvious problem, with the foam material is that some portions of the foam remain uncrushed even though being fully compressed. To ensure effective impact absorbing performance while reducing the space required for the bumper, the portions remaining uncrushed must be minimized. In other words,crush efficiency, defined as the quotient obtained by dividing the crushed length by the original length prior to impact, should be improved.

This report describes the results of research using a skin only bumper structure with high crush efficiency. The results were applied to the Honda ASV-3, a research vehicle developed as a part of the Advanced Safety Vehicle (ASV) project*.

Lateral knee bending angle, knee shearing force and force acting on the tibia were adopted as leg injury parameters. Based upon these parameters, a vehicle front profile which might reduce the parameters, was studied and assessed using computer simulation. Computer simulations were performed to ensure the effectiveness of the skin only bumper structure. The structure effectiveness was confirmed in reducing leg injury parameters with impact tests using a rigid legform impactor and a pedestrian dummy.

INTRODUCTION

The continuous rise in traffic accidents has become a major social problem. Of great concern nowadays is that as the number of aged people further increases, more and more pedestrians may inevitably be involved in accidents.

Accident data show that in Japan, approximately 40% of pedestrians injured, suffer leg injury (1). Injury classification based on severity, excluding those rated as AIS 1, reveals that leg injury is dominated by serious injuries such as bone fracture or knee injury that may result in permanent disability (2). Among those suffering serious leg injury, the majority, approximately 70%, were struck by the bumper, and approximately 10% by the hood edge (3).

This data suggests that the bumper is the area attention should be focused on, and should be improved to reduce the severity of leg injury. One of the effective ways of improving pedestrian protection performance is to optimize the vehicle front profile, including bumper height. Many studies have already been conducted to identify an optimum vehicle front profile (4)(5)(6). Also, the contact force between leg and bumper must be decreased to reduce the severity of leg bone fracture in an accident. To achieve this, several bumper structures using foam material have been reported by others (7)(8).

The aim of our study was to reduce the number of pedestrians who suffer from severe leg injury, due to bumper contact, by 50%. This report contains the results of the study on a skin only bumper that ensures favorable crush efficiency.

The research included; 1) the analysis of accident data to determine a target impact speed for the test, 2) the examination of a vehicle front profile that may effectively reduce the leg injury parameter, 3) computer simulation to evaluate force deformation characteristics of the skin only bumper, 4) confirmation of the simulation results by comparison with the impact test results, and 5) the application of the result to the Honda ASV-3, a research vehicle for pedestrian safety that was developed as a part of the ASV project.

*The ASV project is organized by the Road Transport Bureau of the Ministry of Transport of Japan. Nine Japanese automobile manufacturers participate in the ASV project. Each manufacturer has selected and studied advanced safety technologies that enable the prevention of accidents and reduction the severity of injury due to vehicle impact.

COLLISION SPEED OF VEHICLE

Figure 1 shows the relationship between the cumulative number of pedestrians who suffered from severe leg injury caused by bumper contact and the corresponding vehicle collision speed (3).

The research goal of reducing, by 50%, the number of pedestrians suffering severe leg injury by bumper contact was determined,and the vehicle collision speed was set at 35 km/h in accordance both with the result shown in the figure.

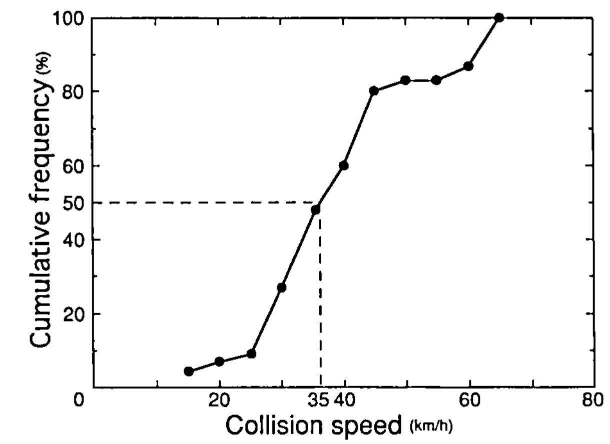

Figure 1. Relationship between collision speed and number of pedestrians suffering severe leg injury caused by bumper contact.

VEHICLE FRONT PROFILE AND KNEE INJURY PARAMETER

Influence of the vehicle front profile on the knee injury parameters, which has a high frequency of permanent disability, was studied using computer simulation. Various vehicle parameters, i.e., bumper height, hood edge height and bumper lead, were used to calculate knee shearing forces and lateral knee bending angles occurring on the dummy's knee joint. The lateral knee bending angle equaling 15 degrees or less and the knee shearing force of 4kN or less were adopted as the knee injury criteria. The former value was proposed by Cesari et. al. (9) and the latter by Kajzer et. al. (10).

Computer Simulation Model

The simulation model is shown in Figure 2. Both the bumper and hood edge of the vehicle model were represented by ellipsoid elements. In particular, the bumper was represented by two ellipsoid elements, considering the size of the leg-to-bumper contact area. Bumper height was defined as the vertical distance between the ground and upper ellipsoid element's center. Bumper lead was defined as the horizontal distance between bumper and hood edge, whose height was represented by the vertical distance between the ground and corresponding ellipsoid element center. The vehicle front profile was that of a standard compact automobile.

The force characteristics generated at the bumper and hood edge was determined to be 4kN constant. The applied value was derived from the fracture evaluation criteria (11)(12). The dummy represents a pedestrian impacted from the side while crossing a road. It consists of one leg with the upper body of the standard HYBRID-II dummy (50th percentile adult male) (13). A weight corresponding to the

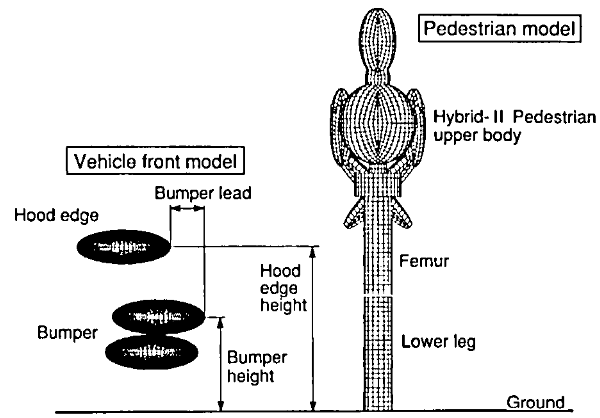

Figure 2. Simulation model of vehicle-pedestrian impact.

other leg was added to the waist part of the dummy.

As mentioned above, the impact speed was set at 35 km/h and the location of the impact was the center of the vehicle front. The lateral bending characteristics of deformable elements, which correspond to the pedestrian dummy's knee ligament is in accordance with the EEVC WG10 deformation characteristics (14), as shown in Figure 3.

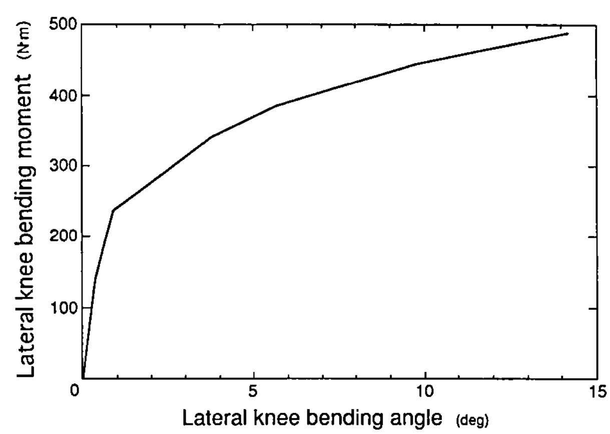

Figure 3. Lateral bending characteristics of deformable elements correspond to the pedestrian dummy's knee ligament.

Parameter Study

The combinations of vehicle specification parameters adopted for the simulation are listed in Table 1. Bumper height, bumper lead and hood edge height were varied between 300 and 500 mm, 50 - 250 mm, and 600 - 800 mm, respectively.

Figure 4 shows the influence of bumper height on the knee injury parameters. The results indicate that lowering bumper height reduces the lateral knee bending angle. This is attributable to the fact that with the lowered bumper height, the bumper-to-leg contact point becomes closer to the center of gravity of the lower leg, which may decrease the bending moment acting on the knee joint. Although the maximum knee shearing force was recorded at a bumper height of 400 mm, none of the force exceeded the knee injury criterion of 4 kN.

Table 1.
Combinations of Vehicle Specification Parameters for Computer Simulation

Bumper height (mm)	Bumper lead (mm)	Hood edge height (mm) 600	700	800
300	50		O	
	150	O	O	O
	250		O	
400	50		O	
	150	O	O	O
	250		O	
500	50		O	
	150	O	O	O
	250		O	

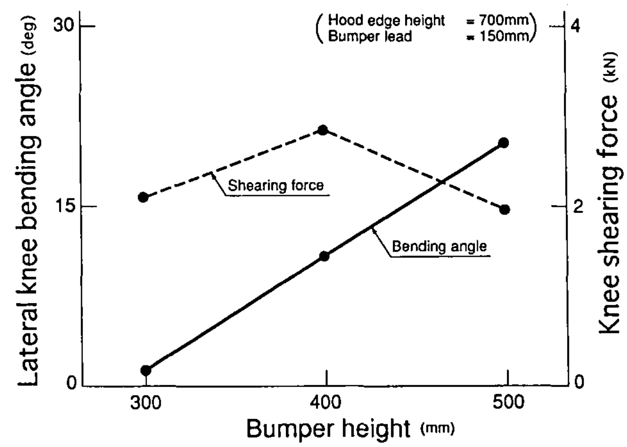

Figure 4. Influence of bumper height on knee injury parameters (computaion).

The influence of bumper lead on knee injury parameters is shown in Figure 5. When the bumper lead setting was changed to 150 mm from 50 mm, the lateral knee bending angle tended to decrease. Knee shearing force, in turn, tended to increase as the bumper lead became larger. None of the shearing force simulation results exceeded the 4 kN knee injury criterion.

The influence of hood edge height on the knee injury parameters is described in Figure 6. It is thought that hood edge height has little effect on both knee shearing force and lateral knee bending angle. The simulation results show a similar tendency compared with the test result obtained by Sakurai et. al. (13) using a legform impactor.

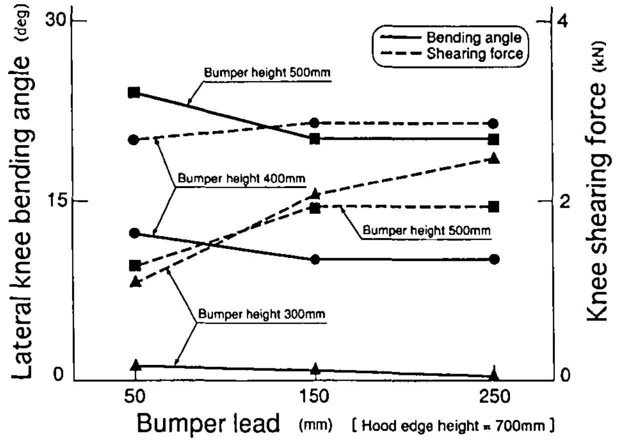

Figure 5. Influence of bumper lead on knee injury parameters (computaion).

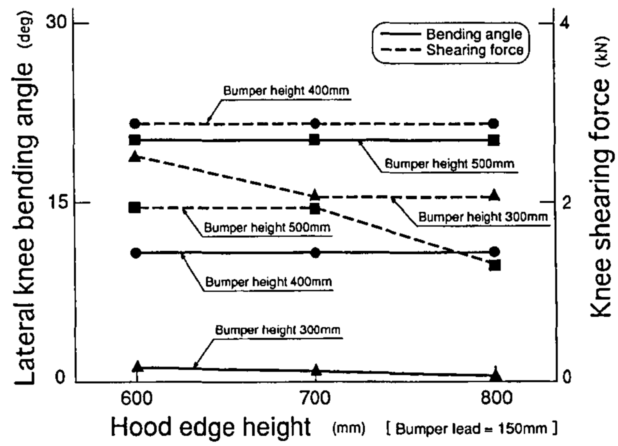

Figure 6. Influence of hood edge height on knee injury parameters (computaion).

The simulation result mentioned above indicates that a lowered bumper height is capable of reducing the knee injury parameters. An extremely low bumper setting, however, causes a problem of unnecessary bumper-to-curb contact. With this in mind, the bumper height setting of 430 mm from the ground was determined.

Neither bumper lead nor hood edge height greatly affects the knee injury parameters. Lawrence et. al. reported that it was preferable to obtain larger bumper lead and lower hood edge height, because they would decrease the impact speed of hood edge against the femur (12). Based on the simulation results and the Lawrence report, the dimensions of the vehicle front profile, i.e., bumper lead and hood edge height, were determined to be 160 mm and 710 mm, respectively.

BUMPER STRUCTURE

Aluminum honeycombs and foam have been used as materials for the energy absorbing structure of bumpers which are capable of reducing the severity of leg fracture. As shown in Figure 7, some portions (S1) of the material remain uncrushed even though being fully compressed.

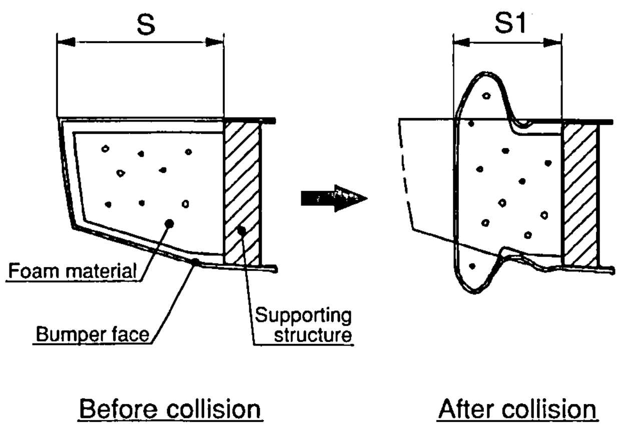

Figure 7. Deformation characteristics of foam filled bumper.

In order to reduce the space (S) for the bumper while ensuring a sufficient level of impact energy absorption performance, minimizing the uncrushed portions (S1) is required. To solve the problem, a bumper structure which absorbs impact energy through the buckling deformation mechanism of the bumper skin, not by a separate energy absorbing structure, was adopted. The energy absorption characteristics of the skin only bumper were examined with computer simulation.

Bumper Simulation Model

The simulation model is shown in Figure 8. The rigid legform impactor was simulated by a cylinder with a diameter of 50 mm, length of 300 mm, and mass of 10 kg (15). During the simulation, force acting on the impactor was calculated by varying the bumper skin thickness. The impact speed was set at 35 km/h and the location of impact was the center of the bumper.

Simulation Result

The impactor force variation in accordance with the changing bumper skin thickness is shown in Figure 9. It was found that the structure, designed to absorb energy by the buckling deformation mechanism of its skin, was under a constant force and had good crush efficiency.

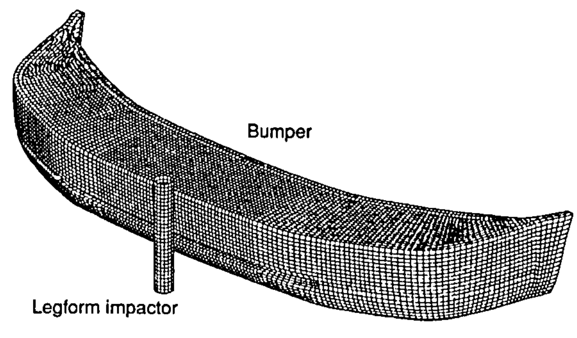

Figure 8. Bumper simulation model.

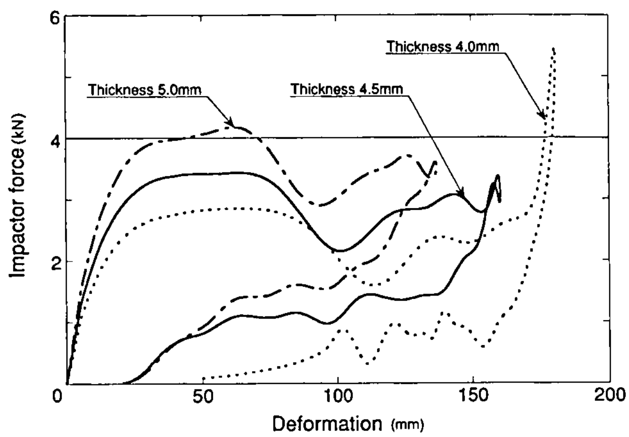

Figure 9. Variation of impactor force relative to the bumper skin thickness (computation).

BUMPER STRUCTURE AND KNEE INJURY PARAMETER

In the simulations that related to the vehicle front profile and knee injury parameter, the bumper and hood edge configurations were represented by ellipsoid elements and the force was a constant 4 kN. The bumper shape was represented by a mesh model, to enable calculations under different force characteristics conditions. With the model, the effect of the vehicle front profile on the knee injury parameters was studied.

The vehicle front end consists of bumper, hood edge and front grille. The hood edge, which comes into contact with the femur, features a double structure comprised of the skin and inner shell construction (hood edge inner) including vertical walls. The double structure was adopted, keeping in mind that the hood edge would come in contact with the femur diagonally, from above.Figure 10 shows the frontal structure of the vehicle.

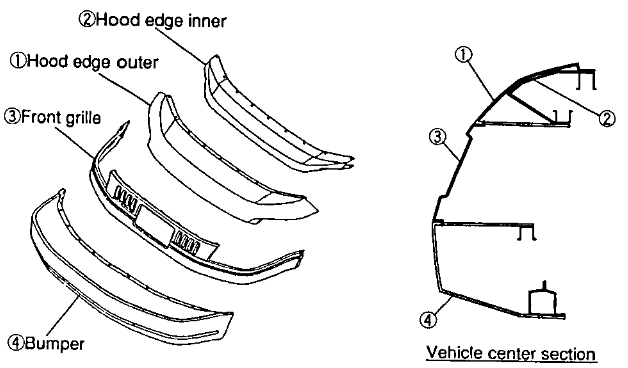

Figure 10. Frontal structure of the vehicle.

Computer Simulation Model

The simulation model of the vehicle front consisting of a bumper, hood edge, front grille and hood is shown in Figure 11. The pedestrian dummy that was utilized in the preceding simulations was used again here.Impact speed was set at 35 km/h and the location of impact was the vehicle front center.

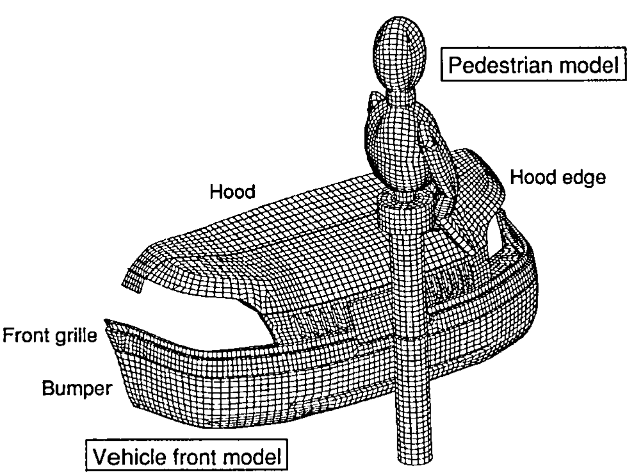

Figure 11. Simulation model of the vehicle frontal structure.

Simulation Result

Figures 12 and 13 present time histories for the lateral knee bending angle and knee shearing force, respectively.

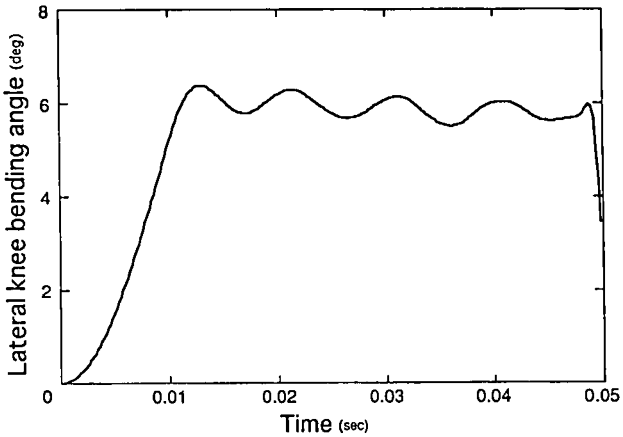

Figure 12. Time history of lateral knee bending angle(computation).

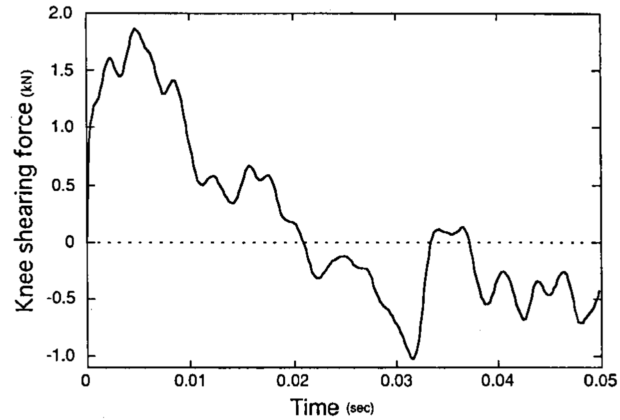

Figure 13. Time history of knee shearing force (computation).

Each value is below the corresponding knee injury criteria.

Compared with the simulation results achieved with the ellipsoid elements, both lateral knee bending angle and knee shearing force are smaller. The reason for this is that the force resulting from the contact between the bumper and lower leg, and the hood edge and femur, may be below the constant force of 4 kN.

The behavior of the pedestrian model, recorded 35 ms after the impact, is shown in Figure 14. The figure suggests that the lateral knee bending angle of the pedestrian model is relatively small.

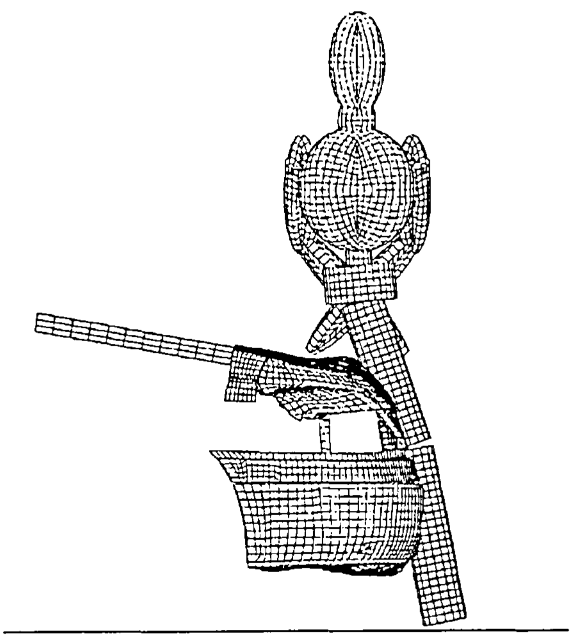

Figure 14. Behavior of the pedestrian model.

CONFIRMATION THROUGH TESTS

The test was conducted using a rigid legform impactor and a pedestrian dummy. The objective of the test was to confirm how effectively the skin only bumper structure would reduce the knee injury parameter. The specifications of the test vehicle, shown in Figure 15, were identical to those of the Honda ASV-3.

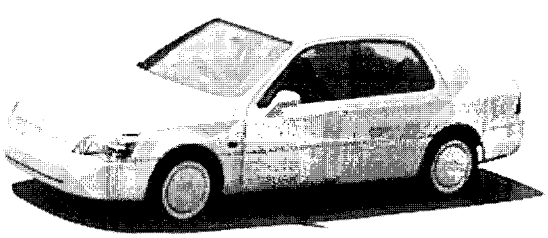

Figure 15. Test vehicle.

The bumper and hood edge are made of polyester elastomer. A polypropylene material was used for the front grille. Additionally, part of the headlight housings were made of polyester elastomer to absorb the energy of the housings.

Impactor Tests

Prior to the pedestrian dummy test, impactor tests were performed to confirm the force deformation characteristics of the bumper (15). The test conditions are depicted in Figure 16.

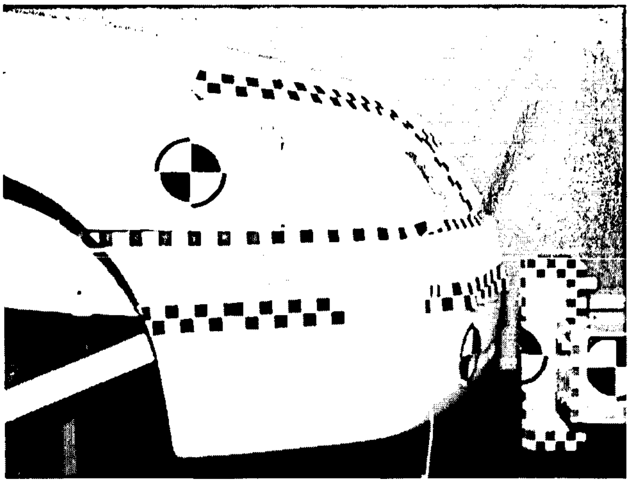

Figure 16. Impactor tests.

The impactor used, shown in Figure 17, was a steel legform with two uniaxial accelerometers mounted on the rear of the impact surface. The mass was 10 kg, including the impacting shaft. The projection angle was set parallel to the ground with the center heights of the impactor and bumper equal.

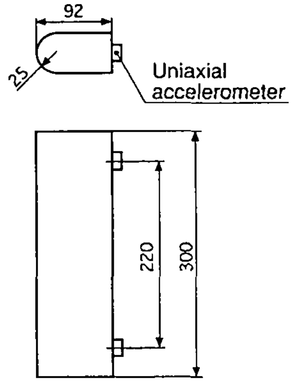

Figure 17. Legform impactor.

The tests were conducted at three locations, exactly at the vehicle center, 430 mm and 550 mm away from the vehicle center, as shown in Figure 18. The impact speed was 35 km/h.

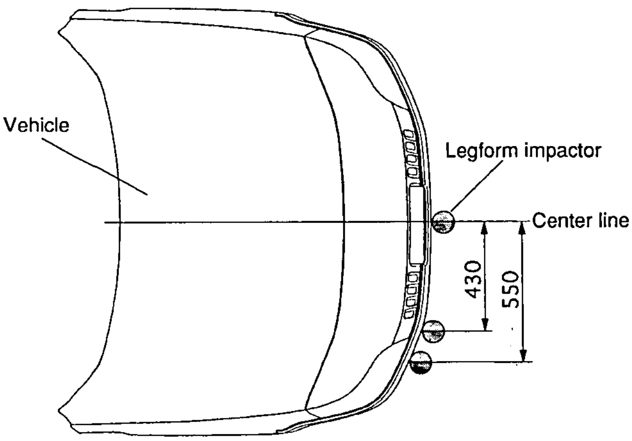

Figure 18. Impact locations (down view).

The test results are shown in Table 2. All the results show that the impactor forces generated at each of the three locations were equal to or below 4 kN. The impactor test confirmed that the skin only bumper structure is capable of reducing the leg fracture parameter to within the evaluation criteria.

Table 2.
Results of Impactor Tests

Test No.	Impact location (mm) (distance from vehicle center)	Impactor force(kN)
B 1	0	3.6
B 2	430	4.0
B 3	550	3.7

Force deformation characteristics when the impact location was at the center of the vehicle front are shown in Figure 19. As shown in the figure, the force characteristics are flat without bottoming out. The crush efficiency was deemed to be approximately 90%, considering that the distance between the bumper front leading edge and supporting structure was 220 mm when measured at the vehicle's longitudinal center line. These results show that the skin only bumper structure possesses high crush efficiency.

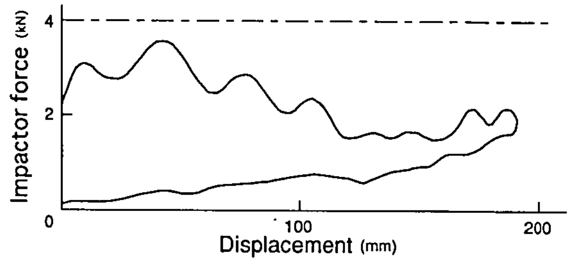

Figure 19. Impactor force deformation characteristics.

Pedestrian Dummy Tests

In order to confirm the effect of the bumper on the knee injury parameters, tests using the dummy whose specifications have been mentioned earlier were performed. The test conditions and set-up are shown in Figure 20.

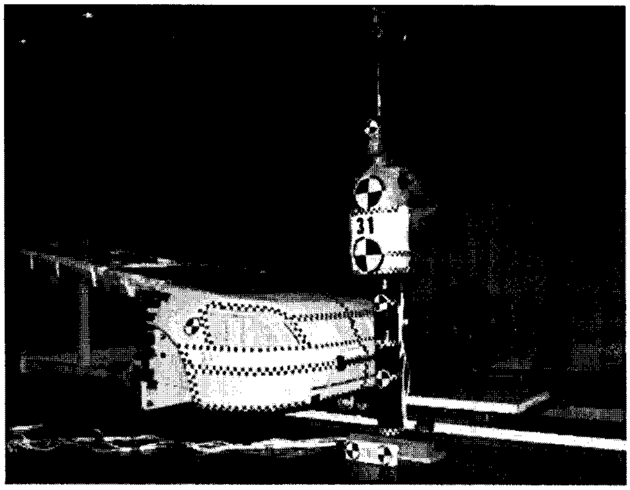

Figure 20. Pedestrian dummy tests.

A sled impact test apparatus was utilized for the test. The pedestrian dummy was set to receive a collision impact from the side. The dummy's arms were put behind its back and both wrists were tied together with rope. Abrasive paper was used to simulate the asphalt surface. The impact speed was 35 km/h. The tests were conducted at three locations, exactly at the vehicle center, 430 mm and 550 mm away from the vehicle center.

The results are shown in Table 3. At all three impact locations, the lateral knee bending angle was confirmed to be equal or less than 15 degrees and the knee shearing force was confirmed to be less than or equal to 4 kN. These results show a close correlation with the simulation results.

Table 3.
Results of Pedestrian Dummy Tests

Test No.	Impact location (mm) (distance from vehicle center)	Lateral knee bending angle(deg)	Knee shearing force(kN)
T1	0	4.9	2.3
T2	430	7.6	2.5
T3	550	6.3	2.2

The measurements of the lateral knee bending angles at the locations 430 mm and 550 mm away from center are comparatively larger than that obtained with the impact location set at the vehicle center. It is possible this resulted from the effect of a large impact force generated at the headlight and lower leg, which might increase the lateral knee bending angle.

CONCLUSIONS

Accident data was analyzed and a goal was set to reduce 50% of pedestrians that suffer severe leg injury due to impact with the bumper. Based upon the analysis, impact speed for all tests and simulations was 35km/h. Lateral knee bending angle, knee shearing force, and forces acting on the tibia were selected as the parameters for evaluating leg injury. The vehicle front profile was investigated and computer simulation was used to determine the optimum profile.Furthermore, computer simulations and impact tests were conducted with a skin only bumper structure.
The findings are as follows:
1) The bumper height setting of 430 mm ensured a lateral knee bending angle of 4.9 -7.6 degrees and a knee shearing force of 2.2 - 2.5 kN. A skin only bumper structure was proven capable of maintaining knee injury parameters within the knee injury criteria relating to lateral knee bending angle and knee shearing force.
2) A bumper structure that absorbs energy through the buckling deformation of its skin ensures a crush efficiency of 90% and flat force deformation characteristics without bottoming out.
3) The force acting on the rigid legform impactor, a parameter of leg fracture, was equal to or less than 4kN due to the use of a polyester elastomer skin only bumper structure. This bumper structure can reduce the leg fracture parameter to within the leg fracture criteria.

FUTURE STUDIES

In this study, bumper structures were developed with a focus on pedestrian protection characteristics. There are still many problems which must be solved before these technologies can be adopted on mass production vehicles, among these, temperature characteristics, durability of the materials, and the expense. To solve the remaining problems and develop practical pedestrian safe vehicles, further research is necessary.

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ACHIEVING COMPATIBILITY AT IMPACT

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Paper Number 96-S4-O-03

ABSTRACT

The concept of designing all road vehicles to be compatible in terms of their impact with each other at an agreed height above ground level came to the fore at the 14th ESV Conference. In particular this is to be done to minimise intrusion and consequent severe injuries, especially to occupants of the smaller vehicle involved, which is assumed to be a small car. The present paper discusses some of the more practical aspects of achieving this compatibility and follows on from the earlier paper (Neilson) and the ideas from other studies (Tarrière),(de Coo) and (Horii). These suggest that the front sections of all impacting structures at the fronts of vehicles should have similar crushing characteristics up to an agreed depth of crush, maybe 700mm, whether they be cars, coaches, vans or goods vehicles. The question is how can this structure be designed. It is proposed that perhaps 150mm behind the softer front face designed to alleviate impacts with pedestrians and then impacts with the sides of cars, there should be a rectangular frame supported by corner posts. The corner posts would crush back to provide the main impact resistance at the corners of the front. The sides would be strong in tension and would be the structures which would actually interact with the frame of the opposing vehicle. They would transfer the impact loads back to the corner posts which would control the crush characteristics. It is assumed that only the perimeter of the rectangular frame contains major impact resisting structures. The role of the engine is considered separately. The rationale behind this layout is discussed together with a number of details about applying it to various types of vehicles and to their sides and rears as well as to their fronts.

INTRODUCTION

It appears that notable reductions in injuries resulting from intrusion in offset frontal impacts are being made now that a number of recent car models are being designed in the light of the proposed offset frontal impact test procedure (Hobbs),(Lowne). This test encourages the strengthening of the front corner structures ahead of the footwell. Before these designs were introduced, cars tended to have weak and easily penetrated front corners, often providing little impact resistance (Foret-Bruno), until the suspension and front wheel were engaged. These might be overridden by the incoming structure of the other vehicle and the wheel might be pushed right back as far as the front seats. The earlier paper (Neilson) which included Figure 1, suggested that better compatibility at impact could be achieved by requiring the frontal structures of all vehicles for at least 700mm from the front to be similar to each other and based on structures suitable for small car fronts.

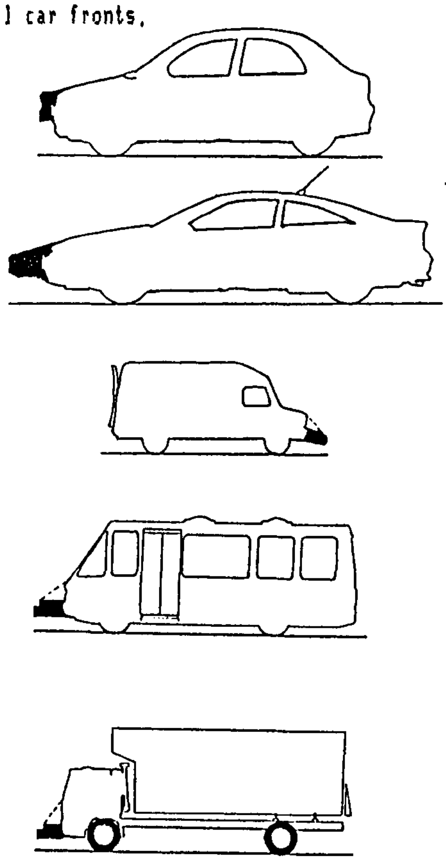

Figure 1. Profiles of road vehicles with compatible fronts.

One consideration is whether the front structures should be rectangular in plan or whether the corners should be tapered to permit vehicles to glance off and slide past each other when impacts are offset or angled (Foret-Bruno). Another aspect occurs when the front corner of one vehicle strikes the side of a car and is the first point of contact.

Tapered or Rectangular Front Structures

Tapered plan views range from pointed ones to almost rectangular ones with small tapers clipping off the corners. The longest practical taper is that which extends back as far as the front of the front tyre and the angle of a taper should be small if it is to modify frontal impacts that are highly offset. The decision to have tapers at the front corners depends on a balance between several considerations.

Tapering removes crush depth at the corner and is of value only if there is reduced friction so that vehicles do not lock into each other when impacts occur along the taper. This type of sliding impact may reduce penetration into the other vehicle and so reduce the possibility of intrusion and subsequent injury.

The effect of a taper in glancing head-on or side swipe impacts is to nudge the vehicles sideways so that they pass each other without a major impact. However the nudge accelerates the passenger compartment violently sideways and consequently throws any occupant adjacent to the other vehicle into the side. The risk is of serious head injury because cant or side rails on cars are not usually padded much and the head has a lower tolerance of side, than it does of frontal, impact.

A risk in all impacts between vehicles is that they swing out of their intended paths and strike other vehicles or objects at the side of the road. This risk may be greater when impacting vehicles do not lock into each other and so separate at the end of the impact. Although a severe primary impact causes most injury in many cases, in a few, the violent rotating motion causes severe injury and occasionally ejection.

Pedestrian Impacts.

The design ideas for protecting pedestrians and the corresponding tests to check them out are well known in Europe. Compatibility between vehicles at impact can be achieved without compromising the protection that can be provided for pedestrians, but there are one or two considerations. While it may be worthwhile to taper the front at the corners to assist with some vehicle to vehicle impacts, this is probably undesirable for pedestrian impacts. Because the human body is fairly flexible it does not follow that the head would be thrust out of the way of a car when the lower part of the body and the legs are impacted by a tapered corner of a car. There may well be a tendency for the head to strike the A posts while the rest of the human body slides past the side of the car. If this is shown to be the case, it is suggested that cars retain their rectangular shape in plan view to provide pedestrian protection up to the corners, even if in more severe impacts into the front corners, the corners fold back so that the oncoming vehicle slides past the front wheel rather than becoming entangled with it. For all vehicles larger than cars it is highly desirable that the protruding front section is rectangular in plan view, because the front corners are so often the first point of impact and the maximum amount of crush depth is needed there.

Many fronts are designed with suitable profiles over which an adult person can be struck without excessive injury. However the wrap-over length from the ground below the bumper, up and over the front compartment up to the base of the windscreen should not be less than 2 metres in length. A problem arises for cars with short bonnets or fronts because an adult head may then strike an A post or even the header rail. It is important that all larger vehicles meet the 2 m rule with a smooth profile in such a way that the head of a tall person does not hit a more vertical surface which may throw the head backwards and injure the neck even if the surface struck is suitably energy absorbing for such an impact. This may mean that low front structures for compatibility have to have a lightweight angled top cover (Figure 1.) which guides the upper torso and head of a pedestrian and suitably absorbs the impact with it. Such a sloping front may be desirable for overall styling purposes.

Side Impacts Into Cars.

In almost all side impacts into cars by vehicles with compatible fronts it is the special front that strikes the side of the car. As discussed previously (Neilson) this is one of the main reasons for having closely specified fronts in terms of height above ground and a suitably soft crush, although the crush depth of this may not need to be more than say 150-200mm as far as these side impacts are concerned. As it is the front corners of the fronts that usually strike the sides first the design of corners is especially critical. One requirement is that for this severity of impact the corners should not easily bite into the side structures, but rather slide along them, even if both corner and car side deform in the process. This is most desirable when the first point of impact is into the footwell of the car. Strengthening of the side structure from behind the front wheel to the A post is necessary and would be worthwhile when impacts into it are from compatible front corners of other vehicles.

Conceptual Design of Front of Car.

The main structure is placed immediately behind the first 150-200mm of the front which copes with impacts into pedestrians and car sides. The present paper proposes that in concept this main structure (Figure 2)

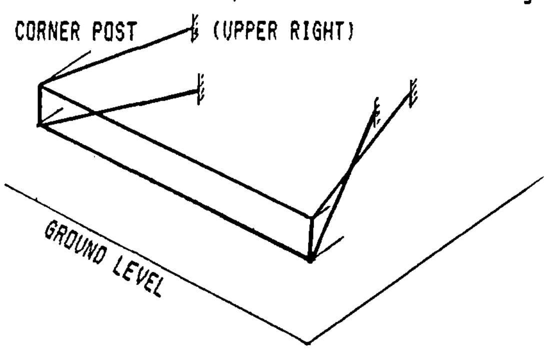

Figure 2. Rectangular Frame mounted on crushable corner posts.

should consist of a rectangular frame stretching from 350mm to 550mm above ground and across the whole width of the vehicle front (except possibly for the outer 100mm at each corner). The frame would be mounted on posts at each corner which would crumple at a predetermined rate as the structure is crushed backwards in the impact. The posts would protrude a little into the softer front so that when two vehicles impact frontally the two rectangular frames strike each other and cannot slide off each other as the fronts crumple. It is to be remembered that the frames will almost never strike each other exactly head-on, but one vehicle will dip its nose below the other one and the impacts will be offset to greater or lesser extent. The edges of the rectangular frames would be strong in tension and would not tear when one frame impacted another. They might deform so that the impact loading would be transmitted to the corner posts which in turn would load up the front structure of the vehicle back into the passenger compartment, cab or chassis. Non-alignment of the two rectangular frames would mean that one post in each frame would take a large share of the impact loadings, with its two adjacent posts taking the rest, leaving the fourth post almost unloaded. It is assumed that components within the frame are not strong in terms of impact resistance. The importance of the rectangular frame concept is that one vehicle could not over or under-ride the other and the compatible front would always be fully utilised. (Figure 3.).

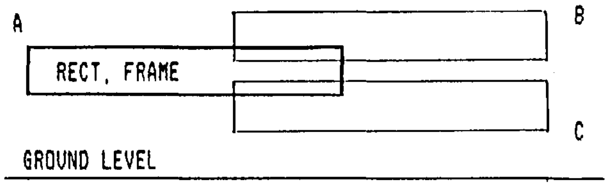

Figure 3. (A) Standard position of Rectangular Frame. Relative Frame positions for impacting mobile barriers for testing fronts of cars (B) or of heavy trucks (C).

A modification of the rectangular frame concept occurs when the engine of a car is fitted just behind the front, as is usually the case. Then the frame could have additional central posts which would load back on to the engine structure, so that there would be six or eight rather than four posts. If there were eight posts then the front could effectively have two rectangular frames, one on each side of the centre line. It might be cheaper to manufacture with six posts, but it would be easier and cheaper to replace one side of a front only, after a well offset type of impact if there were eight posts and effectively two separate frames.

Cars into the Rears, Sides and Fronts of other Vehicles

The worst impacts are into the fronts, rears and sides of large trucks and articulated vehicles. Under-run guards, bumpers or rails have done something to alleviate all these impacts (Robinson) and studies have shown that energy absorbing versions of these improve their protective performance for the occupants of the striking cars. Side guards were introduced to prevent pedestrians, cyclists and motor cyclists from penetrating the understructure of these vehicles and being crushed by their wheels. However side guards play a part in preventing some cars doing the same. The present paper seeks to unify side and rear guards with the proposed compatible frontal structures. Ideally this could be done by having two rails running around the sides and rears (except at the wheels) and fitting these at the same heights above ground as the horizontal sides of the frontal rectangular frames. Special consideration would be needed at the rear corners so that the corners would not be excessively stiff. Ramp angle considerations at the rear might require the lower rail to be free to pivot upwards and slightly rearwards when necessary. The rear rails should be of similar or slightly higher stiffness than that of the frontal rectangular frame so that the rear rails could not be forced forwards in impact enough for the windscreens of small cars to strike the rear structure. The side rails should be of slightly higher stiffness than that of the frontal frame. It would be necessary for the vertical links between the two rails to be set back from the rails so that anyone sliding along the rails after striking them would not be struck by the vertical links. Investigations might show that a single rail along the sides of these vehicles is sufficient. Their suspensions are relatively stiff and one rail may cope with the different heights above ground of loaded and unloaded vehicles. It is a matter of the probability of engagement between the various rails in all the different circumstances.

The sides and rears of buses, coaches, smaller trucks and vans have less need for side and rear rails, but nevertheless their structures should provide equivalent protection even if not by incorporating rails.

All these vehicles would need to have fronts fully compatible with the fronts of cars for at least the first 700mm or so of crush. There may be a ramp angle problem for the lower rail of the rectangular frame on some of these vehicles, but it should be possible to design these to be lifted upwards and forwards under the control of the driver.

Impact Testing for Compatibility

The proposed standard offside frontal impact test for cars by a deformable barrier would need only slight modification to check compatibility of frontal structures. The mobile deformable barrier would need to incorporate a standard frontal structure with a rectangular frame. The barrier would need to impact the test car with the barrier height above ground raised by possibly 150mm and without too much overlap, possibly 30% or less (Figure 3.). A suitable check for deformation of the front footwell would be needed as well as deceleration measurements.

Front structures on vehicles other than cars could be set up in a similar way, but with the barrier height above ground being dropped by up to 150mm from its standard height (350 to 550mm). The main check would be that the mobile barrier was crushed the correct amount and its rectangular frame was not unduly damaged.

CONCLUSIONS

The provision of adequate crush depth when vehicles impact each other is essential if intrusion into the occupants and their consequent injury is to be minimised. This would best be achieved by requiring all road vehicles to have similar frontal structures extending back at least 700mm and these would be similar to the fronts of small cars in terms of their impact characteristics. The present paper considers how this idea might be worked out in practice.

One question is whether front structures should be rectangular in plan or should be tapered. It is concluded that the very front face should be rectangular in plan to suit the needs for the protection of pedestrians. This would also be the case for the next section back which would meet the needs for impact into the sides of cars. The front corners of cars might be somewhat softer than elsewhere because they often strike the sides of cars first and they should not penetrate them but rather slide along them in glancing impacts. The main front structures further back should be basically rectangular in plan so that there is a maximum of crush depth available in frontal impacts that are angled. The only exception might be for the front corners of cars which might fold back over 100mm of width in glancing frontal impacts.

If the full value of frontal crush is to be gained, it is necessary that impacting vehicles should lock into each other at their front face, rather than penetrating each other at the points where their structures are strong and opposing structures are weak. The other aspect of engagement is that vehicles should not under or over-run each other. It is suggested that these requirements could be met by designing the front structure (perhaps 100-200mm back from the front face) to be effectively a rectangular frame mounted on corner posts. These posts would be the main structures resisting impact and they would be capable of being crushed right back without collapse. The purpose of the sides of the frame is that they engage with the sides of the opposing frame and transmit the impact to the corner posts. The sides of the frame must therefore be strong in tension even if somewhat flexible and must not easily be penetrated during impacts. The value of having two horizontal sides to the frame is that the frames will almost always engage with each other even with the misalignments that are typical of frontal impacts. Impacts involve one horizontal and one vertical side of each rectangular frame from the opposing vehicles in most cases.

In practice a rectangular frame on a car would be further supported along its horizontal sides by crushable posts mounted forwards from the engine.

The crushable frontal structures on heavy commercial vehicles may need to have their lower sides and corner posts capable of folding upwards and slightly forwards so that ramp angle problems can be dealt with when the vehicles are being driven up a ramp.

The lower side and rear structures of all vehicles would need to be compatible in height above ground and in their stiffnesses under impact with the frontal structures proposed.

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VEHICLE TO VEHICLE COMPATIBILITY IN REAL WORLD ACCIDENTS

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Paper Number 96-S4-O-04

Abstract

Vehicles in collision can be said to be incompatible if the deformation and structural characteristics mean the occupant loads are unequally distributed between the vehicles. Current crash testing does not consider vehicle safety in terms of the occupants of an opposing vehicle despite the obvious problems associated with incompatible vehicle sizes and structures. Real world accident data can be used to study the effects of vehicle structure on injury outcome in a number of impact types.

This study examines the structural features of car fronts and sides by reviewing real world accident cases where both vehicles were available for examination. Cases are studied to see how the structures previously classified interact. The residual damage to each vehicle is assessed and the load transmission paths identified. Cases with low injury outcome can be compared to those with more severe injuries to see if the vehicle structure influences injury outcome.

Introduction

Conventional thinking tells us that when a smaller car collides with a larger one, the occupant of the smaller car usually fares worse. The smaller vehicle undergoes a higher velocity change, has less structure to absorb the crush and potentially has less interior space to mitigate the effect of occupant contact. Several studies have looked at vehicle to vehicle compatibility, Evans[1], concentrating on the effects of mass, states " a driver in a 900 kg car in a head on collision with another 900 kg car is about 2.0 times as likely to be killed or injured as is a driver of a 1800 kg car in a head on crash with another 1800 kg car.". Thomas[2] looked at a number of accidents , grouped the vehicles according to mass and then looked at the risk of injury to the occupants of the struck car and the risk to the occupants of the striking car. Perhaps unsurprisingly lighter cars seem to present less of a hazard to others, heavier cars present more of a hazard. Interestingly, the paper notes that "the exterior risk associated with each model varied greatly
within a given mass category", this conclusion illustrates that crash compatibility is not only influenced by mass but also by vehicle structure.

Hartemann[3] considered the heights of both the sill and bumper for a range of vehicles and found that with increasing mass the structural elements increased in height. Hartemann concludes that this has the effect of penalising the occupants of the smaller struck vehicle as it leads to greater intrusion in those vehicles already at a mass disadvantage.

Richter[4] examined the question of vehicle to vehicle compatibility with a view to proving that a lightweight vehicle could demonstrate crash compatibility with a more massive vehicle. This paper supported the "traditional" view that lightweight vehicles need to be quite stiff and more massive vehicles need a considerable soft structure to be sacrificially crushed.

Neiderer[6] looked at experimental low mass rigid belt vehicles and demonstrated that although small vehicles experience greater deltaV's than more massive vehicles in the same impact, it is possible to engineer a very rigid low mass vehicle that can offer adequate occupant protection provided full use of inflatable restraints is made.

Neilson[5] proposed that greater compatibility between a wide range of vehicles could be achieved by matching up the points of impact between vehicles and by managing the shape and crush stiffness of the fronts of vehicles. This paper argued that additional structures on the fronts of vehicles could be designed to mitigate the consequences of impact between vehicles as diverse as small cars and heavy goods vehicles; a useful reminder that any changes made to the vehicle fleet as a measure to increase vehicle to vehicle compatibility will inevitably have consequences for all road users, including pedestrians, cyclists, and motorcyclists.

Methodology

The data has been collected within the UK Co-operative Crash Injury Study between 1992 and 1995. Detailed measurements have been taken of the exterior and interior crush to the vehicle and this has been combined with full injury descriptions. Only cases where two vehicles were involved have been considered so the model is not representative of the accident population.

Compatibility Issues - Geometric or Structural Compatibility

Looking at real world accident cases can provide an excellent opportunity to understand the compatibility problems associated with producing a vehicle for use on roads populated by a heterogeneous fleet. Real world cases can illustrate what is meant by incompatibility in the broadest sense as well as defining some of the more subtle aspects of compatibility.

Conventional passenger cars are frequently involved in impacts with larger vehicles, this often results in the more massive vehicle riding over the car as the structural elements of the vehicles are not aligned.

Figure 1. Mid to large size saloon showing damage from an underrun type impact.

Figure 1 shows a mid to large size saloon that was in collision with the van in figure 2. The longitudinal rails of the car were approximately 16 cm below those of the van. This accident type lies at one extreme of a spectrum, incompatibility in this case can be attributed to both the mis-match between the height of the stiff structures of the van and the height of the stiff structures of the car resulting in deformation to the passenger car at a high level. Whilst there is a significant difference in mass between the two vehicles (car 1280 kg, van 1890 kg), the geometric incompatibility is the dominant feature. Mitigation of this type of accident presents a considerable challenge for manufacturers.

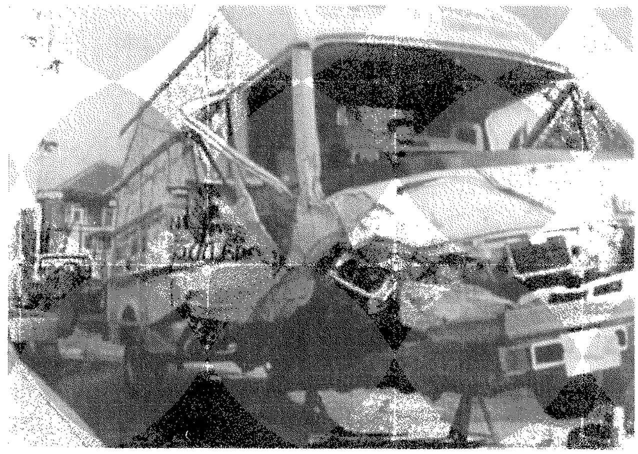

Figure 2. Large van illustrating damage from impact with vehicle in figure 1.

Accidents between apparently similar vehicles can demonstrate that height is not the only variable affecting compatibility. Figures 3 & 4 show a large estate car which has collided with another large estate car (figures 5 & 6). Much of the resulting deformation of this first vehicle occurs behind the A-post, indicative of a relatively soft structure. One result of this weak construction is that the passenger compartment shows a reduction in size and associated intrusion as the vehicle has crushed.

Figure 3. Large Estate car 1. Mass = 1292 kg EES = 53 kph.

Figure 4. Large Estate car 1.

The other vehicle in this impact shows considerably less crushing and its passenger compartment is largely undisturbed. This second vehicle is considerably stiffer and has dissipated energy at the expense of the other vehicle. Differences in compatibility in this case result from a combination of events. Whilst the longitudinals have aligned laterally (geometric compatibility), there is a clear miss-match of stiffness between the vehicles in the vertical plane. In this collision the dominant incompatibility is due to stiffness difference but a slightly different crash configuration might also introduce geometric incompatibilities.

Figure 5. Large Estate car 2. Mass = 1625 kg EES = 27 kph.

The driver of large estate car 1 suffered serious injuries including a dislocated right hip and was hospitalised for over five weeks, the driver of large estate car 2 suffered minor bruising and lacerations.

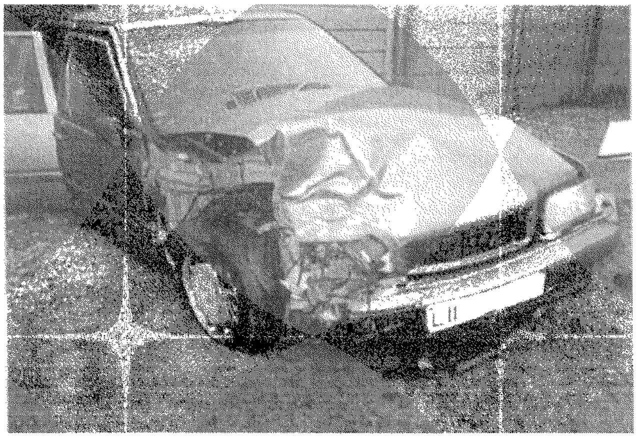

Figure 6. Large Estate car 2.

Real world accident cases demonstrate the wide range of response that vehicles exhibit to different striking vehicles. Studying one particular model in one scenario can furnish information about the response to different striking vehicles. Figures 7 to 10 illustrate a popular small/medium car when struck by two other popular models of similar size.

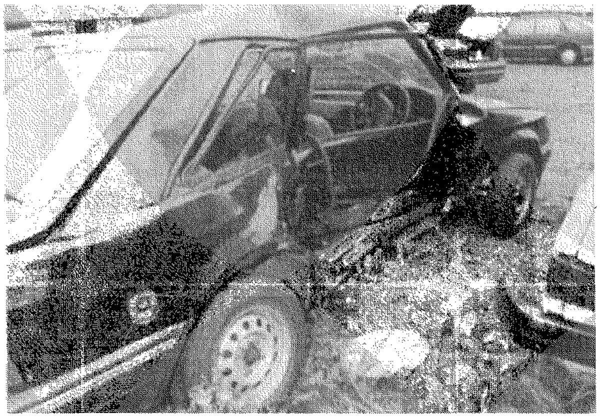

Figure 7. Small / medium car 1. . Mass = 979 kg EES = 35 kph

Figures 7 & 8 illustrate the response to an impact from a vehicle which appears very stiff in localised areas. The door of the struck car (figure 7) shows a concentration of loading just above the sill level resulting in the door overriding the sill and localised intrusion. The driver of this car was hospitalised for two days with bruising, abrasions and a minor renal trauma. This type of impact is location critical, that is, there is a risk of the stiff areas of the bullet car missing the load carrying parts of the target car, resulting in intrusion which may be very hostile to the occupants of the target vehicle. In this case the very stiff longitudinal members passed either side of the driver of the struck car; had the impact been centred further along the drivers door there would be a high probability of a very severe contact with the drivers pelvic area. Equally, had the striking car involved more of the B-post the occupant loads would have been reduced further.

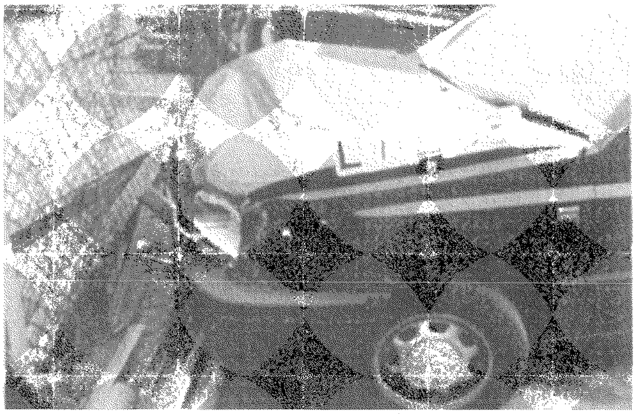

Figure 8. collision partner Small / medium car 1.. Mass = 1060 kg EES = 21 kph.

Figure 9. Small / medium car 2.. Mass = 1029 kg EES = 31 kph.

Figure 10. collision partner Small / medium car 2.. Mass = 1225 kg EES = 19 kph

Figures 9 & 10 are typical of an impact with a car which demonstrates a more homogeneous soft frontal area. The door and sill of the struck car (figure 9) show how loading has been applied very evenly to the vehicle side. Whilst still resulting in intrusion, this is of a less localised nature than the previous case. Moving the centre of this impact along the length of the struck car would be unlikely to result in a different injury outcome for the struck side occupant, an undisplaced fracture of the left pubic rami. In this case the front homogeneity of the car in figure 10 has given it more adaptability to variation in crash configuration.

Figures 11 to 13 show an accident case involving a popular small car and a popular medium sized car. The medium car has a mass advantage of 388 kg.

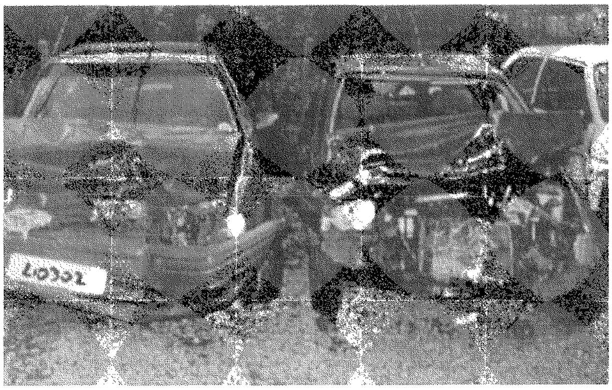

Figure 11.

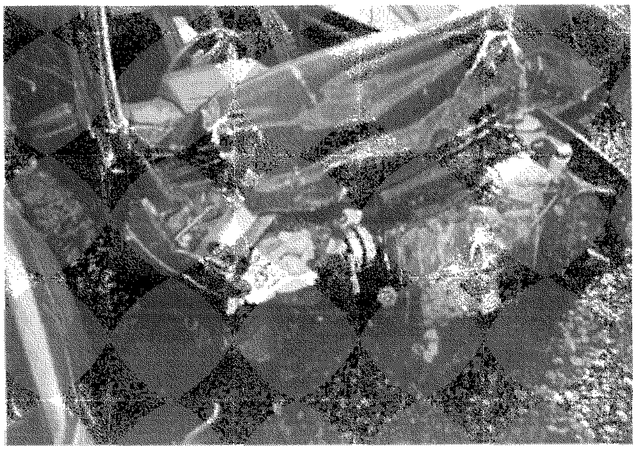

Figure 12. Mass = 891 kg EES = 45 kph.

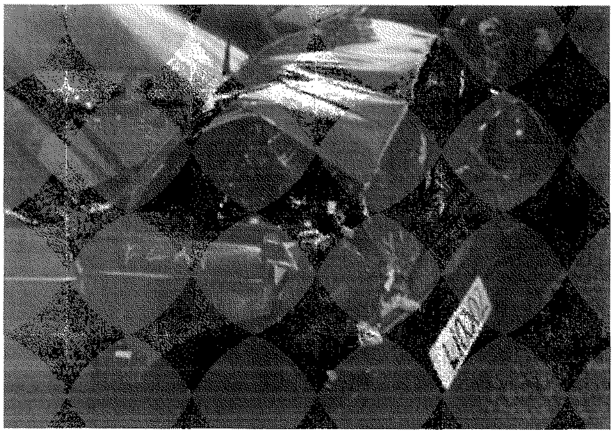

Figure 13. Mass = 1279 kg   EES = 37 kph.

This accident demonstrates good compatibility *between these two vehicles in this scenario.* There is no obvious sign of a disparity in stiffness in either a vertical or lateral plane. The engine of either vehicle has interacted with the offside longitudinal member of the other vehicle; due to the transverse engine configuration of both of these vehicles, this effect is not location critical. These vehicles exhibit geometric compatibility and are well matched for stiffness. Despite a significant mass difference the overall effect is that of compatibility due to well matched structures. The driver of the heavier vehicle, a 46 year old male, suffered a fractured sternum and a compression fracture of the 2nd lumbar vertebra resulting in hospitalisation for six days. The other driver, a 30 year old male, escaped with a minor laceration to the forehead and bruising to nose, chest and left knee.

Heights of structural elements

Hartemann[3] studying the relationship between compatibility and the heights of various structural elements measured 75 popular vehicles covering most of the French market at that time. The data were used to construct a diagram to show the relationship between increasing mass and the heights of vehicle structural elements, (reproduced here as figure 14).

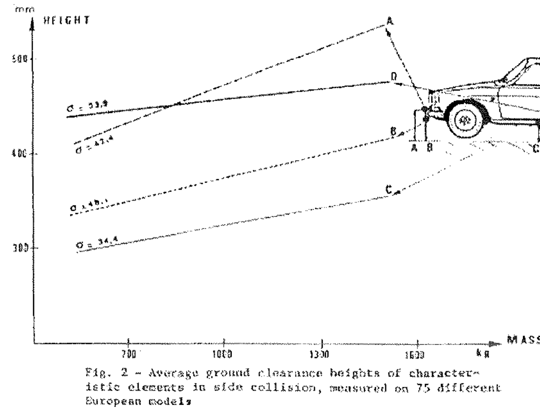

Figure 14. Heights of structural elements. Hartemann (1979).

This work provided strong support for the view that with increasing mass there was a corresponding increase in the ground clearance height of various structural elements. This effect would appear to further penalise the smaller vehicle in collision with a larger vehicle as override of the smaller vehicle is increasingly likely. In a repetition of this earlier work, this study measured the ground clearance heights of the sills (tops and bottoms) and longitudinal members (tops and bottoms) of 185 models of car including most in current production, to give a coverage of 82.5% of the models represented in CCIS, a significant proportion of the UK market.

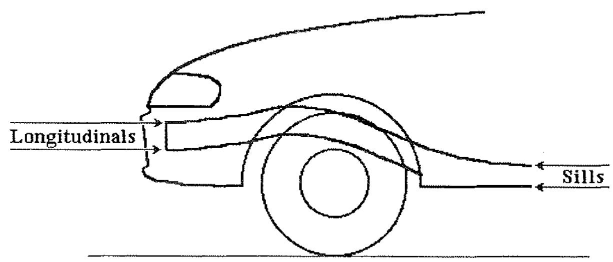

Figure 15. Measuring points for sills and longitudinals.

Plotting the ground clearance height of the structural elements in figures 16 & 17 reveals a different picture to that presented by Hartemann. Firstly, the sill top measurements in figure 16 do not show a simple relationship with mass, there is considerable scatter in the data with most data points falling into a clearly defined broad band. In most cars the height of the sill top is virtually independent of mass and is presumably mostly designed on access considerations.

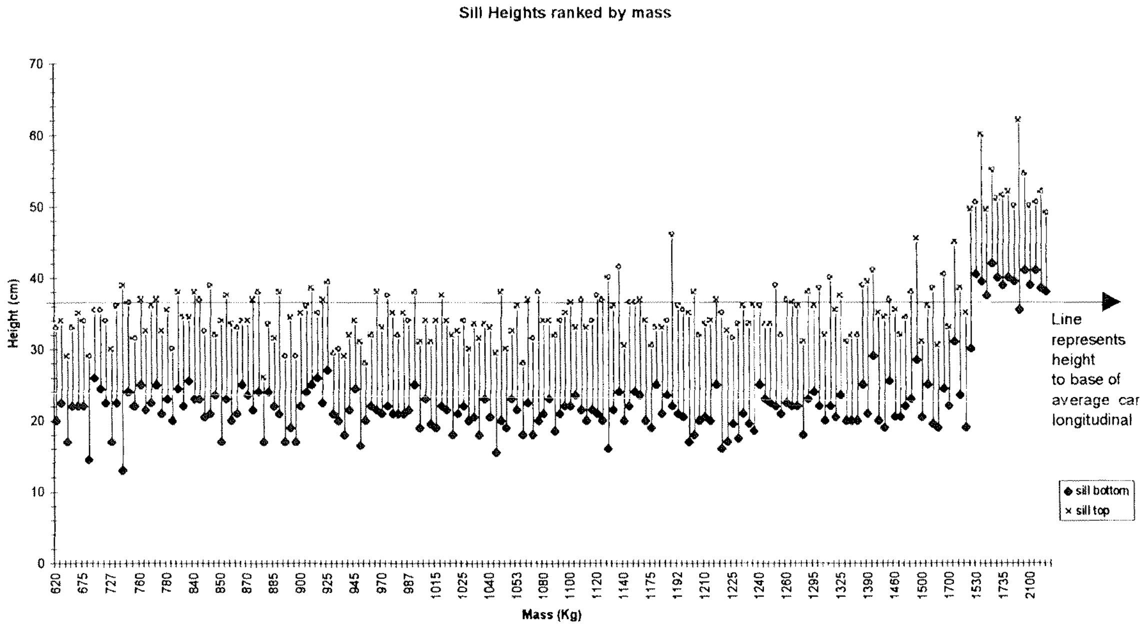

Figure 16. Sill top and sill bottom heights plotted against mass.

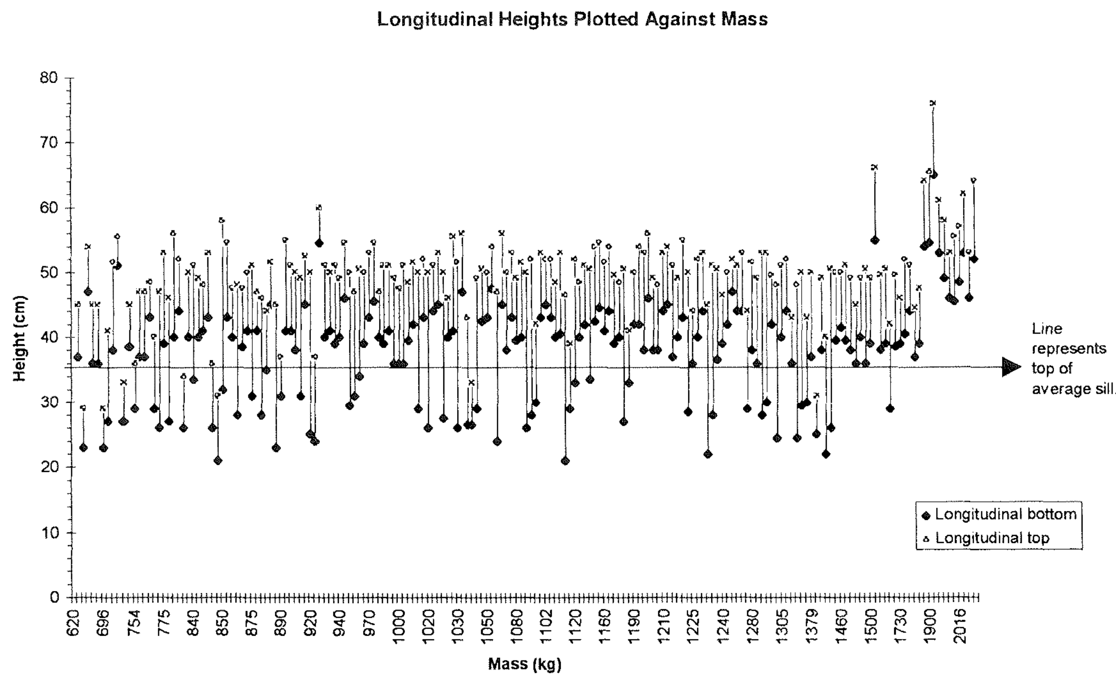

Figure 17. Longitudinal top and longitudinal bottom heights plotted against mass.

The longitudinal measurements represented in figure 17 show more scatter than the sill measurements (figure 16) but there is still no evidence of a relationship with mass. Most of the data points are above the line representing the top of the average car sill.

The high points on the right of figures 16 & 17 are all four wheel drive vehicles which hold an increasing market share in Europe, a phenomenon not evident when Hartemann wrote in 1979. These vehicles are characterised by having considerably more ground clearance than conventional passenger cars and frequently a rigid chassis construction. These vehicles can potentially present a considerable risk to the occupants of a conventional passenger car, not only because of the frequently greater mass they enjoy, but also by virtue of their potential to concentrate loading above the sills or longitudinals of the struck car.

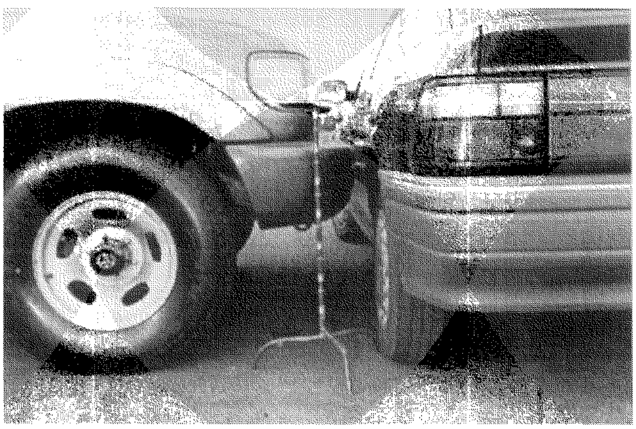

Figure 18. Four wheel drive vehicle compared with popular mid-size passenger car.

Furthermore, whilst these four wheel drive vehicles may have very stiff longitudinal chassis rails tied together laterally by a substantial cross member; they are often considerably softer above this level. This design characteristic may lead to a very aggressive concentrated loading pattern. Figure 18 highlights the vertical incompatibility, figure 19 is used to emphasise the problem.

The vehicles in the sample had an average of 37.3 centimetres ground clearance to the bottom of their longitudinals and an average ground clearance height of 36.1 cm to the top of the sill. When the 4x4 vehicles are removed from the sample the clearance to the bottom of the longitudinals is an average of 36.3 centimetres, whilst the average ground clearance height to the top of the sill is 34.6 cm.

Figure 19. Occupants view of approaching four wheel drive vehicle.

The line on Figure 16 represents the average of the "bottom of longitudinal" measurements for normal passenger cars (36.3 centimetres). The line shows that the average longitudinal would not interact with 64% of sills.

Figure 20. Illustrating the difference between car frontal structures and side structures.

Injury Analysis

A sample database of 469 CCIS cases involving 938 vehicles and drivers was examined. Every case involved two cars striking each other, no other impact types were considered. The most severe impact for each vehicle was determined and used as a classifying variable. The breakdown of impact directions is detailed in table 1.

Table 1.
Impact direction

Impact direction	$N^{o.}$ of cases	%
Frontal	592	63
Struck side	91	10
Non - struck side	83	9
Rear	60	6
Other	112	12
Total	938	100

For the 91 struck side drivers the angle of any door intrusion was determined. This was categorised as,

1: door intrusion greatest at chest level. (Chest > Pelvis)
2: intrusion profile vertical. (Chest = Pelvis)
3: door intrusion greatest at pelvis level. (Chest < Pelvis)

The incidence of these intrusion categories is shown in table 2.

Table 2.
Showing breakdown of intrusion categories for struck side drivers

Intrusion Type	None	Chest > Pelvis	Chest = Pelvis	Chest < Pelvis	not classified
Cases (%)	28 (31%)	9 (10%)	17 (19%)	35 (38%)	2 (2%)

Table 3.
Maximum AIS of 91 struck side drivers by intrusion category

	None	Chest > Pelvis	Chest = Pelvis	Chest < Pelvis	not classified
MAIS 0	11		3	3	
MAIS 1	14	3	8	16	1
MAIS 2	1	3	2	9	1
MAIS 3	1	2	3	1	
MAIS 4			1	3	
MAIS 5				2	
MAIS 6		1		1	
not known	1				
totals (%)	28 (31%)	9 (10%)	17 (19%)	35 (38%)	2 (2%)

Table 4.
Body regions injured by intrusion type

	None	Chest > Pelvis	Chest = Pelvis	Chest < Pelvis	not classified	Total
Head	8 (20 %)	5 (12 %)	5 (12 %)	22 (54 %)	1 (2 %)	41 (100 %)
Thorax	3 (11 %)	3 (11 %)	6 (21 %)	14 (50%)	2 (7 %)	28 (100 %)
Abdomen	0 (0 %)	2 (12.5 %)	2 (12.5 %)	12 (75%)	0 (0 %)	16 (100 %)
Pelvis and lower extremity	8 (17 %)	2 (4 %)	12 (26 %)	23 (50%)	1 (2 %)	46 (99 %)

Examining the case files shows that those vehicles which suffer more intrusion at the pelvis level are frequently loaded in a concentrated area, whereas those cases which show a vertical intrusion profile are often the result of diffuse loading. Intrusion which is greatest at chest level is less common in car to car impacts. Examining the Maximum Abbreviated Injury Severity scores of the drivers in these categories reveals the pattern shown in table 3.

Severe injuries (MAIS 4+) most commonly occurred when the maximum residual intrusion was at pelvis level. The total number of injuries in selected body regions, classified by intrusion slant are represented in table 4.

Whilst greatest intrusion at the pelvis level is present in just over 38% of the cases in this sample, this category is over represented in all of the body regions in table 4. Whilst this may indicate that this type of intrusion is more injurious than the other types, it may also indicate a phenomenon of higher energy impacts. Reference to the graphs comparing sill heights with longitudinal heights shows that this type of intrusion could be expected in a large proportion of side impacts.

Crash Testing

Critics of offset deformable barrier testing have argued that vehicles optimised for this test configuration will inevitably be aggressive. The asymmetrical loading of the car frontal structures may well lead manufacturers to produce very stiff designs, however, these designs are not necessarily any more aggressive than the models they replace. A recent series of crash tests at the Transport Research Laboratory illustrate this point.

Model **A** is a popular large car designed as a replacement for model **B.** An offset deformable barrier test of model **A** showed a very favourable result for this model. Examination of this vehicle reveals a very stiff construction and a considerable degree of tying together both laterally and vertically. The front end design of model **A** shows a response to offset barrier testing. Models **A** and **B** were then balanced for mass and crashed at 50 kph into the side of identical target vehicles carrying instrumented dummies. The dummy readings are summarised in table 5.

Table 5.
Readings from instrumented dummies struck by two different car models.

Bullet Vehicle		**Model A**	**Model**
HIC		251	321
Top Rib Compression	mm	29.4	35.9
Mid Rib Compression	mm	33.8	38.5
Bottom Rib Compression	mm	34.8	38.8
Viscous Criterion Top	m/s	0.74	1.49
Viscous Criterion Mid	m/s	0.93	2.49
Viscous Criterion Bottom	m/s	1.12	1.16
TTI Top	G	79	102
TTI Mid	G	83	96
TTI Bottom	G	66	80
Pubic Force	kN	7.43	5.69

Source TRL

The vehicle impacted by model **B** showed extensive localised intrusion at the base of the drivers door. Matching model **B** to the damaged area of the struck car it was clear that there had been no interaction with the sill of the struck car. The intrusion was largely a result of localised loading by the longitudinals of model **B**. There was no geometric compatibility in either a vertical or a lateral plane as the longitudinals of model **B** did not have any significant tie in with the rest of the vehicle structure.

In contrast to the vehicle struck by model **B**, the vehicle struck by model **A** showed a remarkably even loading with no localised intrusion. The sill of the struck car had taken a lot of the impact loads due to contact with stiff components low down on model **A**. This impact showed a good degree of geometric compatibility in both a vertical and a lateral plane. The stiffness of model **A** was eclipsed by its uniformity resulting in a less aggressive diffuse contact.

Comparison of the instrumented dummy results shows that model **A**, designed to pass the offset deformable barrier test, has the potential to be less injurious than its predecessor model **B** which was designed to pass the ECE Regulation 12 perpendicular block test.

Discussion

Compatibility is a broad issue which is difficult to confine to single parameters such as mass or stiffness. Any definition of compatibility needs to encompass the concepts of geometric compatibility in both a lateral and a vertical plane, as well as considering both stiffness and mass. Furthermore, compatibility may be location critical or indeed it may depend on which vehicles are colliding. A model may demonstrate good compatibility when in collision with other models of similar construction whilst the same model may show poor compatibility with models of a different construction. For this reason proposals to crash test for compatibility would be difficult to define in a meaningful way. Measurement of vehicle sills and longitudinals highlight the mismatch between these structures in side impacts. Large four wheel drive vehicles present special problems when they collide with ordinary passenger cars. Intrusion can be classified into different types and compared with real world accident cases. Examination of these real world cases can be used to show differences in load transmission paths and identify different degrees of compatibility in a number of parameters. Most vehicles studied showed a combination of compatible features, aggressive features and incompatible features, resulting in difficult to classify scenarios. Accidents which demonstrate overwhelmingly incompatible features are easy to identify, good compatibility is harder to demonstrate.

Acknowledgements

The Co-operative Crash Injury Study is managed by the Transport Research Laboratory (TRL) on behalf of the Department of Transport (Vehicle Standards and Engineering Division) who fund the project with Ford Motor Company Limited, the Rover Group Limited and Toyota Motor Europe. The data were collected by teams at the Vehicle Safety Research Centre at Loughborough University, the Accident Research Centre at The University of Birmingham and from the Vehicle Inspectorate.

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COMPATIBILITY OF CARS IN FRONTAL AND SIDE IMPACT

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ABSTRACT

For many years there has been an acceptance that the level of protection provided to car occupants was influenced not only by the characteristics of the car they occupied but also by the characteristics of the other car involved. Heavy and stiff cars have been said to be aggressive and with other aspects, such as the height of the front structure, these characteristics have been seen as the major factors influencing compatibility. Despite these concerns, comparatively little research has been carried out to identify the most important characteristics influencing compatibility or to quantify their effects.

For the UK Department of Transport, the Transport Research Laboratory has embarked on a programme of research which will attempt to answer these questions. This paper discusses what is currently known and details some of the early findings from the research. Already, some of the simplistic views held in the past are being questioned.

INTRODUCTION

It is generally accepted that the injuries suffered by car occupants are not only influenced by the protection afforded by their car but also by the characteristics of what their car hits. In some accidents, the object hit is obviously aggressive. Under-run impacts with large vehicles and impacts with roadside poles are obvious examples of poor compatibility. However, impacts between cars are more frequent and the possibilities for making improvements are potentially greater. Improving compatibility in such accidents is the first priority.

Despite the paucity of good information, much is claimed about what influences compatibility. There seems to be a general acceptance that what is thought to be obvious must be true. Previous experience, such as that gained from frontal and side impact, should have shown that acceptance of the obvious is frequently a poor indicator of truth and that there is no substitute for scientific research. For some, acceptance of the conventional wisdom has suited their interests in helping to support arguments against the use of offset and higher speeds for frontal impact testing.

For the UK Department of Transport, the Transport Research Laboratory has started a study of compatibility. Efforts will be concentrated on attempting to understand what influences compatibility in car to car frontal and side impacts. In both cases, some measure of the potential for casualty savings will be sought and related to those parameters affecting compatibility. Although the work will not initially attempt to address the problems in Heavy Goods Vehicle under-run and pole impacts, care will be taken to ensure that any proposals would not adversely affect protection in such accidents. Similarly, care will be exercised to ensure that pedestrian protection will not be compromised.

A staged approach to the study may be followed. Initially, the target vehicle may be left unchanged with the effects of changes to the bullet car only being considered. Later, it may become clear that changes to the bullet and target car, taken together, could bring about greater improvements. If it can be shown that significant benefits could result from improvements in compatibility, a method of assessment will be sought.

This paper discusses those factors which are currently seen as having the greatest influence on compatibility and gives the results from some of the work carried out so far.

CAR TO CAR IMPACTS

Two aspects dominate the literature on compatibility. The effects of mass and mass ratio on injury risk have been studied for many years. There is plenty of evidence that, in car to car impacts, the risk of injury in the heavier car is lower than that in the lighter car (1,2). Some analyses have shown that the risk of injury is lower when two heavy cars impact each other compared with two light cars (3). Others have claimed that, in impacts between cars of similar mass, lighter cars are as safe as heavy ones (4). For these analyses, mass is a convenient parameter which is recorded in the databases. However, mass is also a surrogate measure for other factors, which may themselves influence injury risk. Vehicle size, length of front structure, height, size of passenger compartment and presence and quality of safety features are all well correlated with mass. The influence of one or more of these might appear as a mass effect.

The second factor which is frequently claimed to affect compatibility is the stiffness of the car's front structure. It may appear obvious that a stiffer structure would be more aggressive but there is no information to show the effect of simply changing stiffness. Where comparisons have been made between cars of different stiffness, other related factors have had a more dominant effect.

CAR TO CAR FRONTAL IMPACT

A deformable barrier for frontal impact testing has been adopted in Europe because of the importance of the way a car's structure is loaded in an accident with another car. In the research carried out during the development of the test procedure, the way car structures interact was seen to be of great importance. The same is true for compatibility, the way car structures interact, their frontal stiffness and shape and the mass of the car and its major frontal components are all likely to have some influence on compatibility. The importance of each characteristic and their interactions have yet to be established.

Structural Interaction

In the past, most references to the effect of frontal stiffness on compatibility have been based on the assumption that the impacting front structures would interact properly with one another. This lead to the consideration of an average stiffness, often based on measurements taken from a loadcell wall impact.

However, the interaction of car structures in car to car impacts is unlike this. Car fronts often have local areas of stiff structure, within a much larger area of weaker structure. Usually there will be two main longitudinals, one each side of the engine compartment. These may be bridged across the front by a lateral beam. Upper longitudinals may also be present, running along the top edge of the front wings. Currently, it is unusual for there to be any substantial connecting structures between the fronts of these upper "shotguns" or between them and the main longitudinals. Some cars do have good interconnections in these areas, to reduce noise, vibration, harshness problems, to improve torsional rigidity or in response to deformable frontal impact tests. Because of this lack of stiffness uniformity at the front of cars, it is usual for the stiff parts of one car to penetrate the weaker parts of the other.

In addition there is usually horizontal and vertical misalignment of the stiff structures in the opposing cars. Even where they are initially aligned, this alignment may be unstable so that the structures move out of alignment during the impact. The effect of vertical mis-alignment is that the structure of one car over-rides that of the other.

In an offset frontal car to car impact carried out to study the effect of car mass, a small car was impacted with a medium sized car. However, the test showed that, in this instance, problems with the interaction of the cars' structures had a dominate effect on the outcome. The structure of the small car over-rode that of the larger car. The structure then bent upwards, absorbing only a limited amount of energy, and the passenger compartment collapsed. In comparison, the difference in mass and front structure length had only a secondary effect (Figure 1).

The adoption of a deformable barrier face for frontal impact testing should help to improve the interaction of frontal structures. It encourages greater horizontal and vertical connection between the stiff elements which results in a more homogeneous stiffness distribution. In an offset frontal impact between two identical cars, with well connected front structures, there was little structural penetration or over-riding (Figure 2). Although the deformable frontal test should aid compatibility, the extent

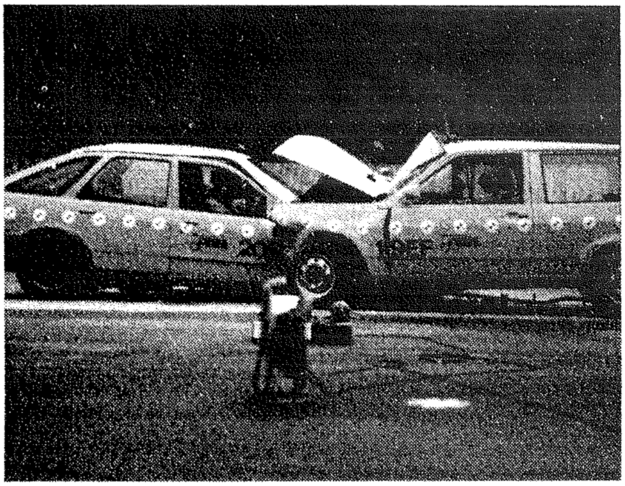

Figure 1. Small car over-riding larger car due to vertical misalignment of the front structures.

Figure 2. Car with well connected front structures experienced little penetration or over-ride in impact with similar car.

of its influence is not yet clear. In particular, when different models and sizes of car impact each other.

Structural Stiffness

Over the years, it has been generally accepted that increasing the frontal stiffness of a car would make it more aggressive. However, the relationship between frontal stiffness and injury risk has not been established. This is particularly important as the impact speed for frontal impact testing has been limited because of concerns about its effect on compatibility.

There are a number of ways in which frontal stiffness might influence injuries in frontal impact. Some are related to the relative stiffness of the two cars and some are related to their absolute stiffnesses.

Assuming the structures interact properly, the proportion of impact energy absorbed by each car, and its associated crush, will be related to their relative stiffnesses, with the weaker car deforming more. If the deformation extends to the passenger compartment, this will affect the risk of injury due to occupant contact with the car's interior. If the passenger compartment remains intact, the risk of injury in each car will be related to the ride down given by the total the deformation of both cars. In such circumstances, increasing the stiffness of either car will increase the acceleration seen by both cars and the risk of acceleration related injuries.

The development of more intrusion resistant passenger compartments, improved seat belt systems and airbags will increase the proportion of impacts where occupant contact with the interior, other than in the footwell, is prevented. In such impacts, non-contact injuries will result from the reaction of the restraint system. Comparatively little is known about the relationship between vehicle deceleration, seat belt force and occupant loading. However, for a well restrained occupant, chest loading and deceleration related neck injuries might be the most important.

Although it is clear that there is a relationship between vehicle deceleration and injuries, little is known about the effects of variations in vehicle deceleration time history and injury. There are a number of stages in this time history. Initially, little structure is involved and the vehicle acceleration builds up slowly. As the structure starts to collapse, the vehicle acceleration fluctuates as new parts of the structure become involved, load up and collapse. Later in the impact, the acceleration levels decrease. Throughout this period the occupant's restraint loading also varies. Initially the occupant moves forward with no restraint loading. When the slack has been taken up in the restraint system, occupant loading increases and then decreases. Changes in the car's acceleration time history: before the onset of belt load, as the belt load peaks and during the latter stages of the impact can be expected to influence injury risk. The demonstration car ESV 87, showed that it was possible to build a stiffer car with greater overall acceleration but with a smoother response and a lower peak acceleration (5). The individual effects of overall acceleration and peak acceleration are poorly understood. Peak acceleration could be expected to have a greater effect on rib acceleration, whereas the cumulative effect of overall acceleration might be expected to influence rib deflection more.

It is frequently suggested that benefits would result from small cars being stiff and large cars weak. This would allow for a greater proportion of the impact energy to be absorbed in the longer front of the large car when they hit each other. Unfortunately, there would appear to be some flaws in this argument. Firstly, the stiffness of large cars is somewhat dictated by high engine and suspension loads acting well forward of the passenger compartment. Secondly, large cars often have relatively full engine compartments with little free space to allow for additional collapse. Thirdly, the small stiff cars can be expected to hit one another, particularly if they are concentrated together in towns, and their occupants will then suffer from the resulting high accelerations.

Frontal Shape

The shape of the car front can be expected to influence the way in which the two cars interact in the impact. However the effect may be small compared with that caused by the stiffness distribution. It may be more relevant to consider shape as a constraint. In a frontal impact, it might be reasonable to expect that vertical, flat fronted vehicles would interact best. For good interaction with a heavy goods vehicle, a tall front might also help to prevent under-run. However, such a shape is poor for pedestrian protection.

Mass and Size

Mass ratio is the most studied of all the aspects relevant to compatibility. Studies of large accident databases have been used to show that the occupants of the heavier vehicle in an impact have a lower risk of injury than those in the lighter vehicle. For frontal impact, a simple consideration of momentum supports this as the velocity change of the heavier car will be less than that of the lighter car. For an unrestrained occupant, this velocity change will be directly reflected in the secondary impact velocity of the occupant with the car's interior. For a restrained occupant, the effect will be seen through the cars' acceleration.

However, the effects seen may not all be due to mass ratio. Mass is also a surrogate for vehicle size and some aspects of vehicle size may contribute to the effects observed. In impacts, larger cars have certain advantages. They have longer frontal structures with a greater potential to absorb the impact energy. The structure may be stronger, to take the engine and suspension loads, and stiff longitudinals will tend to be located higher within the structure. This makes it more likely that the structure of the large car will over-ride that of the smaller car. It is usually safer to be in the over-riding car as the other car tends to suffer from excessive deformation of the upper load path. These effects, together with others related to size, make the more massive car appear safer. However, it is not clear how much of the apparent benefits are due directly to mass.

Engine Mass

Usually the engine is situated near the front of the car where it may become intimately involved in the impact. The engine is effectively a rigid concentrated mass. Its deceleration will generate a significant force and the way the engine interacts with the opposing car will affect the loads imposed and the load paths. Consequently, the mass, width, height, location, support and shape of the engine can all be expected to have some influence on compatibility.

CAR TO CAR SIDE IMPACT

Structural Interaction

Research on side impact protection showed the importance of considering the separate load paths through the door to the occupant and through the stiffer sill and door pillars, to the main mass of the car. Controlling the separate responses of these two load paths was seen to strongly influence injury parameters (6,7,8). The structural geometry and stiffness characteristics of the front of a bullet vehicle will influence the way these different parts of the car side are loaded and so influence injury risk.

For good occupant protection, it is desirable for the main impact loads to be transferred to the target car through the side sill and the door pillars. Although controlled intrusion of the door is beneficial, the forces transferred through it need to be limited.

There are a number of ways in which the design of the bullet vehicle's front could help. Alignment of the front structure with the side sill should provide a direct load path below the occupant. To be effective, alignment would need to be maintained throughout the impact despite any vertical motion of the vehicles. To be fully effective, strengthening of the target car's floor may also be necessary.

Unfortunately, if all the impact loading is taken below the occupant, it may be difficult to maintain a vertical intrusion profile for the door and occupant loading may be delayed. Where the door intrudes more at the bottom, chest forces are concentrated on the lower ribs and occur late relative to those on the pelvis.

If significant impact loading occurs above sill height, it is important that it is well distributed. Concentrated loading, through the door to the occupant, from stiff structures in the bullet vehicle will be dangerous. Cars with a more homogeneous frontal stiffness will avoid this concentrated loading.

Structural Stiffness

As for frontal impact, it has been accepted that a bullet car with a stiff front structure would be more aggressive than one with a weak structure. However, frontal stiffness also affects the structural interaction and with current designs, the interaction effect is likely to dominate. A car with a stiff homogeneous front may bridge the gap between the door pillars. If the stiffness is lower, the bridging effect will reduce and loading through the door to the occupant will be greater. This was demonstrated in comparative tests using the EEVC and NHTSA side impact barriers. The stiffer NHTSA barrier bridged across the door pillars. The weaker EEVC barrier was unable to sustain the loading from the door pillars, collapsed locally allowing the barrier to penetrate the door opening. Loading to the vulnerable chest was lower with the stiffer barrier (6) (Table 1).

Table 1.
Side Impact Tests on Medium Sized European Car Showing More Aggressive Nature of Weaker EEVC Barrier Compared with Stiffer NHTSA Barrier

Barrier Face	EEVC	NHTSA
Chest Compression (mm)		
Top rib	36	26
Middle rib	40	26
Bottom rib	48	30
Viscous Criterion		
Top rib	1.04	0.58
Middle rib	1.02	0.22
Bottom rib	1.35	0.39
Thoracic Trauma Index		
Top rib	164	123
Middle rib	162	112
Bottom rib	169	104
Pubic Symphysis Force (kN)	6.9	15.8

There is also some evidence from car to car tests that the benefits from a more homogeneous frontal stiffness are greater than the adverse effects due to increased stiffness. Comparison tests have been made, using a modern mid-sized European car and its predecessor as bullet vehicles. The modern car performs well in the offset deformable frontal test whereas the earlier model performed badly. The newer car had a much stiffer frontal structure but had good horizontal and vertical connections improving its homogeneity (Figure 3). The weaker older car had little connecting its two front longitudinals which penetrated the side of the target (Figure 4). From these tests it could be seen that the new stiff car was less aggressive, to the vulnerable chest, than the older weaker car (Table 2). In fact the front structures of both cars were much stiffer than the side of the target car, such that there was no significant deformation of their main structures (Figure 5). If the front structures of cars do not deform, increasing their stiffness will have no direct effect on injury risk. There may be a benefit from adding a weaker structure ahead of that

Figure 3. Quite good load spreading from stiff car with well interconnected front structure.

Figure 4. Penetration of stiff longitudinals from weak car with poor frontal interconnections.

necessary for frontal impact protection. However, care would be needed to ensure that the structural interaction of the cars is not compromised.

Table 2.
Side Impact Comparison Showing the Benefit from Good Structural Interconnection of a Stiff Bullet Car Compared with Poor Interconnection of a Weaker Bullet Car

Bullet Car	STIFF	WEAK
Chest Compression (mm)		
Top rib	29	36
Middle rib	34	39
Bottom rib	35	39
Viscous Criterion		
Top rib	0.74	0.97
Middle rib	0.93	1.02
Bottom rib	1.12	1.06
Thoracic Trauma Index		
Top rib	126	142
Middle rib	131	136
Bottom rib	113	120
Pubic Symphysis Force (kN)	7.3	5.7

Frontal Shape

An obvious way to lower the load path is by ensuring that only low structure on the bullet car comes into contact with the target car's side. This could be achieved by the shape of the car front. Projecting bumpers with sloping or curved fronts above them might achieve this. However, as stated earlier, there may be problem related to the intrusion profile and delayed loading of the chest. Sloping fronts create problems in impacts with pedestrians. The lack of bonnet leading edge limits the locations where acceleration forces can be transmitted to the pedestrian, resulting in high head impact velocities. Curved fronts overcome this problem. Unfortunately, neither sloping nor curved fronts provide an upper load path at the very front of the car, so in impacts with heavy goods vehicles, there is no high structure to interact early in the impact and prevent under-run.

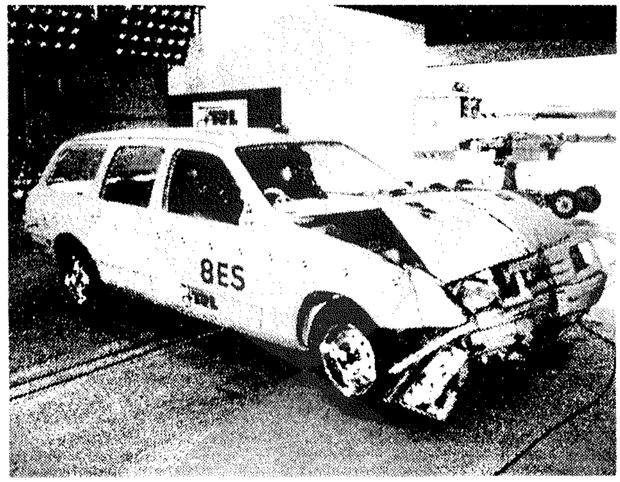

Figure 5. Front of weak car with poor structural connections showing no deformation of longitudinals.

In car to car tests comparing bullet cars with square (Figure 6) and curved fronts (Figure 7), the square fronted car was seen to load the chest more and the curved fronted car concentrated loads more on the pelvis (Table 3).

Mass and Size

Although door intrusion and occupant loading occur very early in a side impact, much of the momentum transfer between the cars occurs much later. Usually, by the time all the important injury parameters have peaked, only between fifteen and thirty percent of the momentum transfer has occurred.

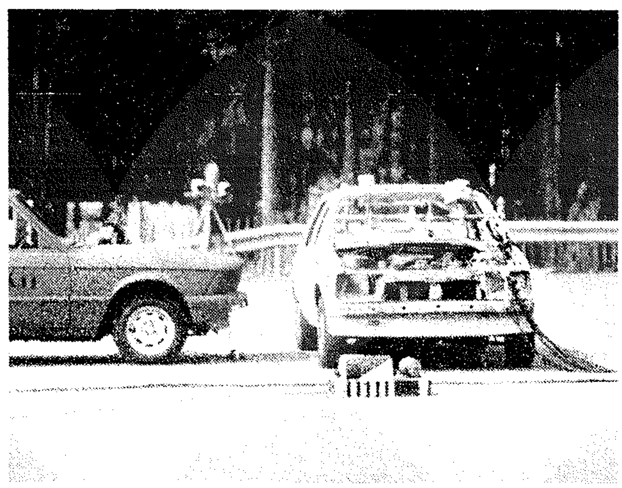

Figure 6. Square Fronted Bullet Car Concentrates Loading on the Chest

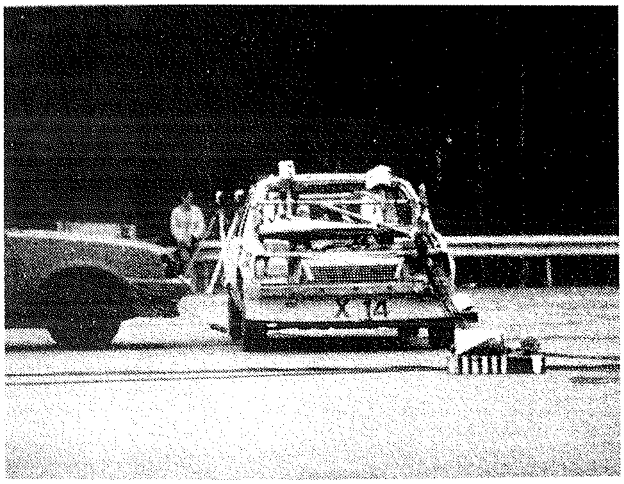

Figure 7. Curved Fronted Bullet Car Concentrates Loading on the Pelvis.

Table 3.
Comparative Side Impact Tests with Square and Curved Fronted Bullet Cars Showing Relative Loading of Chest and Pelvis

Bullet Car	SQUARE	CURVED
Chest Compression (mm)		
Top rib	50	34
Middle rib	52	37
Bottom rib	53	40
Viscous Criterion		
Top rib	1.39	0.77
Middle rib	1.66	0.77
Bottom rib	1.46	0.93
Thoracic Trauma Index		
Top rib	143	127
Middle rib	169	137
Bottom rib	172	133
Pubic Symphysis Force (kN)	6.6	9.8

This low rate of momentum transfer may explain why changing the mass of the impacting vehicle has little effect. This was confirmed from tests with the EEVC barrier where the trolley mass was increased from 950 to 1350 kg (6).

Engine Mass

As for frontal impact, the location of the engine near the front of the car means that there is the potential for it to directly load the target car's side. This does not usually happen in a classic side impact, because there is too little collapse of the bullet car's front. If the engine were closer to the front or the front collapsed more, it may play a more significant part.

DISCUSSION

For this study, the compatibility of cars in frontal and side impacts is being considered. At the same time, consideration is being given to ensure that changes to improve compatibility in such impacts would not compromise protection in impacts with pedestrians, heavy goods vehicles or road side obstacles.

Already, it is becoming clear that some of the commonly held views are open to question and claims for the effect of mass and stiffness may also be related to other factors. Initial findings suggest that the most important factor influencing compatibility is the structural interaction between the cars. With current cars this effect is so dominant that it is difficult to identify how important the other factors such as frontal stiffness, mass, shape and location of engine might be. The effective stiffness of the car both depends on and influences the quality of the interaction. Stiffness also varies through the impact time and the shape of the resulting deceleration pulse needs to be studied to see how it influences occupant injuries. The claim that increased stiffness should be avoided because of compatibility concerns may be overstated. Fronts are already much stiffer than car sides so there may be little direct effect from further stiffening. Mass has its own direct effect on the impact but it is strongly correlated with size, frontal length, passenger compartment size, structural height and the presence and quality of safety features.

In frontal impacts, the extent of overlap and angle of approach will also affect a number of the factors discussed here. In particular, they affect the way the structures interact and their effective stiffness. All of the information about side impact is related to the classic situation which is replicated in the side impact test procedures. The situation will change as the impact configuration and severity vary.

It is still too early to draw many firm conclusions about compatibility. The effect of independently varying the apparently important factors needs to be studied and the potential for making improvements needs to be established. After that, a method of assessing and encouraging good compatibility will need to be sought.

ACKNOWLEDGEMENTS

The work described in this paper forms part of a Vehicle Standards and Engineering Division, Department of Transport funded research programme conducted by the Transport Research Laboratory.

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THE GLIDING ZONE
A new approach to increase passive safety for vehicles

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Paper Number 96-S4-O-06

ABSTRACT

If the amount of energy that has to be converted into deformation work is too high or if the effective surface for the energy exchange is too small, the current concept of passive safety in passenger cars fails. This is already the case within the permissible speed ranges. A transformation into deformation work at lateral areas without breaking into passenger compartment of a vehicle is hardly possible.
Due to legally tolerated speed differences in mixed traffic, it is possible in case of a collision to exceed the amount of energy which can be transformed. Therefore, intrusion into and disrupting of passenger compartments take place in real accidents.
To solve this problem a new safety concept for passenger cars is being proposed. This concept consists of a combination between diversion of impulse by gliding off and transformation of kinetic energy into deformation work.
For this purpose, the vehicle gets a surrounding main member, which provides the essential portion of structural strength. This surrounding member - board frame - is being designed as a frame. Outwards it is closed, has a smooth surface and is very stiff, especially in lateral areas. It is the idea, that in case of an accident there will be no catching and intrusion between vehicles but gliding off at each other → gliding zone. Besides the gliding effect front and rear areas of the vehicle provide also the possibility to transform a certain amount of energy into deformation work, however, without allowing any intrusion or catching.
By combining gliding and crumple zones the speed after collision is being reduced, thus the run-out distance is being shortened. In addition the deceleration of the compartment remains controllable in a biomechanical sense. At higher speed differences it will be true, too. Intrusion or even disrupting of the passenger compartment is being avoided by the combination of gliding and crumple zones.

INTRODUCTION

Passive safety reaches a deadlock. The expenditure of money, manpower and equipment is ever increasing to make even smaller engineering progresses in car safety[1]. Present cornerstones of passive safety are subject of wear and tear.

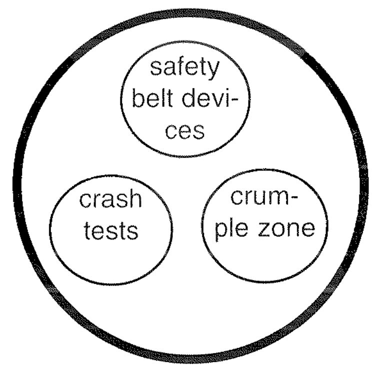

Figure 1. Cornerstones of passive safety

For all efforts there are more than 50,000 persons put to death and 1.5 million injured by traffic accidents in the European Community every year, which results in more than a million sojourns in a hospital and costs round about 70 billions ECU [2].
German traffic accident statistics say that almost every second person during life will be injured by an automobile [3].
These terrifying facts should be a sufficient reason for defining the cornerstones of passive safety in a new way. Due to the adoption of safety belts and the duty to use them there was a halving of the number of killed and injured persons in spite of increasing mobility in the seventies. A jump in the accident statistics to such a degree has not been reached yet despite of the introduction of modern

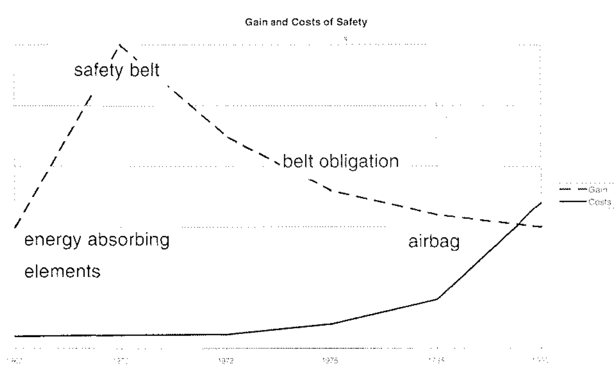

Figure 2. Principle trend of injured persons

elements such as airbags and safety belt tightener.
The thesis on the other hand the more airbags the better should not be our aim. It will be a long way until the day,

when overall electronic devices to prevent accidents will reduce the number of accidents to a minimum. Instead of adopting another European frontal crash test procedure in addition to the existing US one and instead of introducing another European side impact test procedure in addition to the US standards it seems to be more useful not only to polish up one of the former cornerstones of passive safety, but also to allocate financial funds to support innovations.

COMPATIBILITY

It is a fact, that passive safety has largely been understood as a self interested protection system with a serious consequence: mass always wins! The system of the three cornerstones of passive safetywhich apparently seems to be stable, has to be supplemented by a fourth cornerstone: the compatibility. Surely this is not a new demand; for more than two decades safety scientist have requested this. Not only due to compact-cars coming on market, but also due to a lack of recognizability all efforts should be made to decrease the number seriousness of injuries from accidents. Thus the idea of compatibility should be introduced on all vehicles immediately.

In former discussions about compatibility there is a limitation upon level-indices of heavy mass cars, aside from poorly functioning safety bar devices of utility vehicles, therefore upon a discussion of frontal crashes with great overlap.

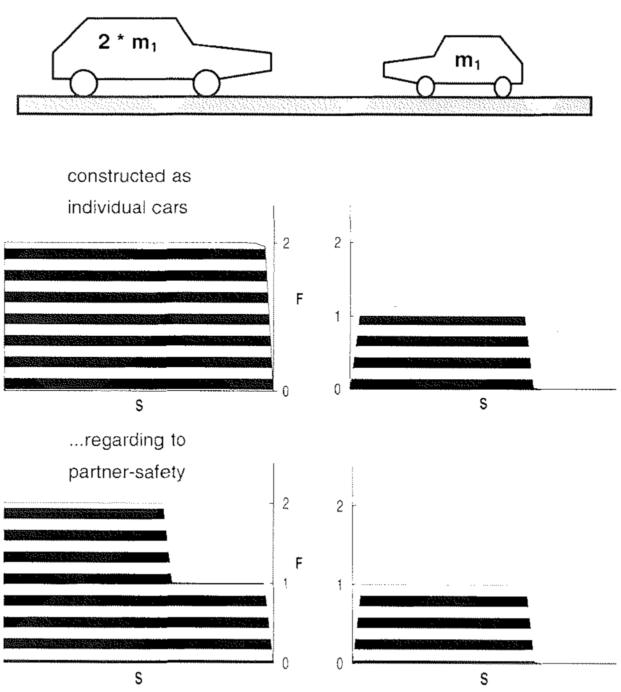

Figure 3. Principle of a level-index

It is also well-known that, with means of level indices, there is no possibility to reduce energy equivalent differences of the legal speed limits. For example in Germany there is a difference of velocity of 200 km/h outside cities, but, under the assumption that distribution of masses is roughly equal, differences of velocity of 110 km/h can be controlled by technical means. Therefore, with adoption of level-indices, the zone of energy absorption on heavy mass

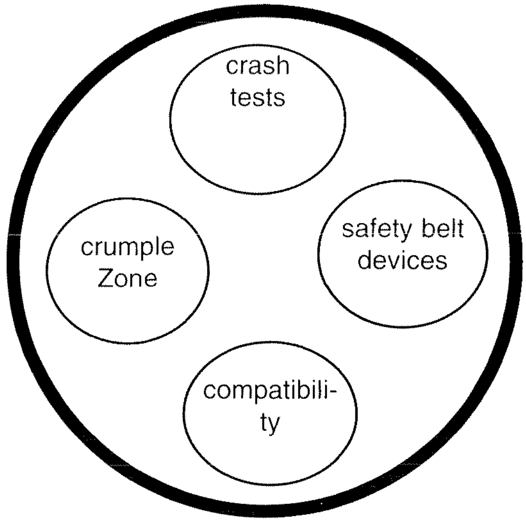

Figure 4. Future cornerstones of passive safety

vehicles cannot be made as long as necessary. Disproportionate cars would be the result. This could be a reason why compatibility has not yet been introduced as a fourth cornerstone for passive safety.

REAL ACCIDENTS

Some examples of real accidents should contribute to illustrate the problematic nature:
First of all there is a gliding collision between a car and a tree. All present elements of safety in this car could not decrease the seriousness of injury of the occupants.

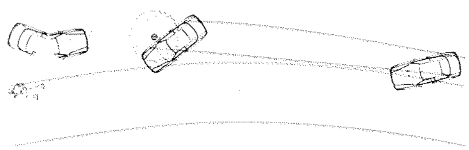

Figure 5a. Car to tree collision

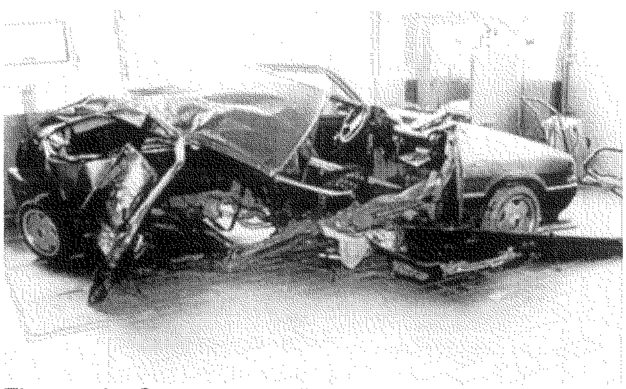

Figure 5b. Car to tree collision

The second accident represents a car to car collision that happened in two-way traffic. In this context it is comparable with the first test. In this case present safety devices such as the crumple zone could not minimize seriousness of injury for the occupants in one of the cars. This car was disrupted into two pieces.

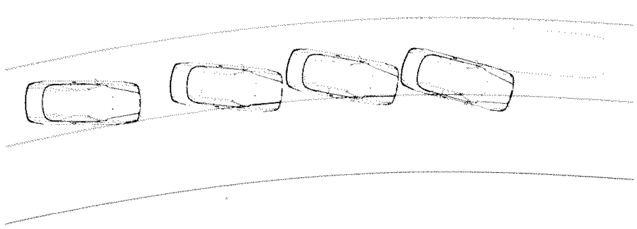

Figure 6a. Car to car collision

Figure 6b. Car to car collision

The next accident shows a typical collision during nighttime between a car and a truck turning into a street. Even in this case most modern safety elements could not prevent car occupant from heaviest injuries. The car ran under the truck.

Figure 7. Car to truck collision

The next accident shows the after collision situation in a motorbike to truck accident. Hardly any comment is necessary.

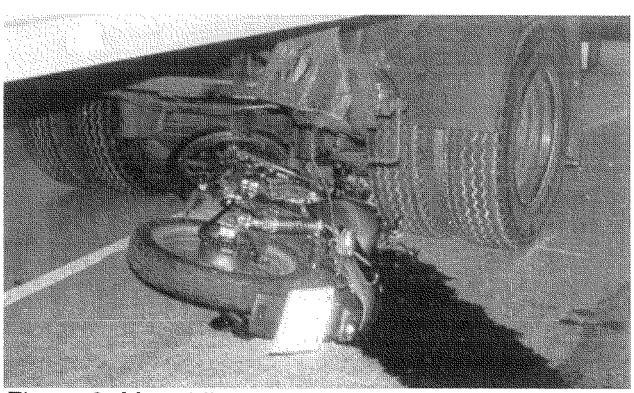

Figure 8. Motorbike to truck collision

The next pictures show a truck to truck collision in two-way traffic with minimal overlap. Even in this case the existing proofing conditions of truck cabs could not minimize seriousness of injury of one of the truck drivers.

Figure 9. Truck to truck collision

The last example shows a two-way traffic accident between a car and a car with trailer. The difference of velocity was about 180 km/h and the overlap was about 2 inches.

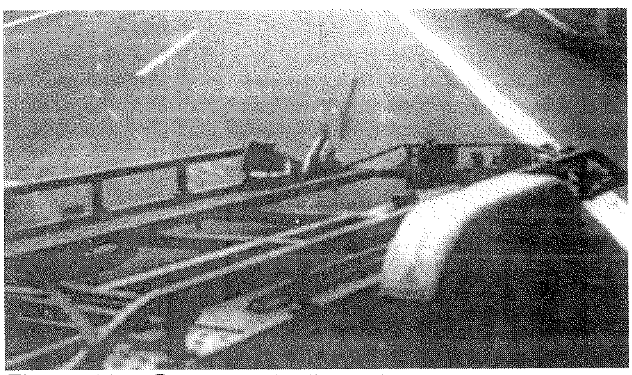

Figure 10a. Car to car/trailer collision

As expected the trailer was nearly undamaged, but the car was split open alongside. All present safety elements could not prevent most serios injuries of the driver.

Figure 10b. Car to car/trailer collision

These few accidents shown before demonstrate the wide spectrum of types of accidents which have to be considered, if there shall be further success in minimizing seriousness of injuries after accidents. It is safe to assume that compatibility is not only coordination between light and heavy mass cars, but also has to include trucks, trailers and motorbikes.

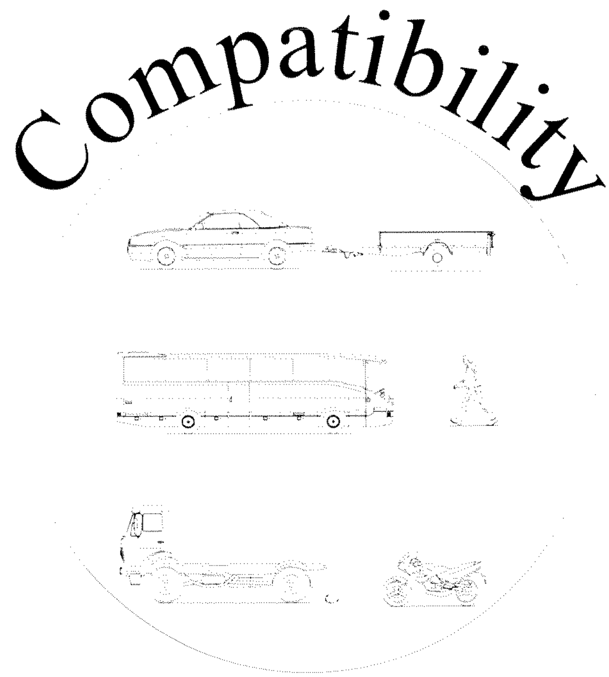

Figure 11. Compatibility

It is obvious that there is no possibility to control differences of velocity such as for example 180 km/h between a motorbike and a truck in frontal collisions even with level indices elaborated at highest accuracy.

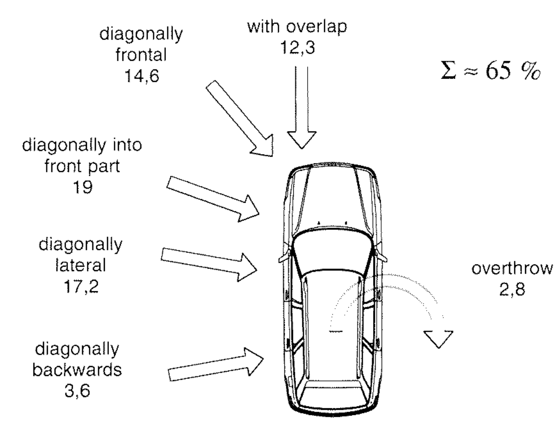

Figure 12. Frequency for collision impact direction angles

It is well-known that only 35 % of all car to car collisions take place with large overlaps and under an angle of 180° respectively 90°. Nearly 65 % of accidents take place with a small overlap and under a clear collision angle.
To get furtheron a positive sharp bend in the balance sheet of accidents it is necessary to point the interest to these collisions with a small overlap and an acute angle of collision.

THE GLIDING ZONE

In this context the question is posed whether it can be efficient to transform kinetic energy directly at the point of collision. Definitely this question can be answered when including the available electronic systems: The more of the present energy will be transformed in the post crash phase the higher the speed difference which can be controlled. Partially transferring energy transformation into post crash phase can only be realized by gliding off. This however can only take place, if there is a gliding zone established at each car.

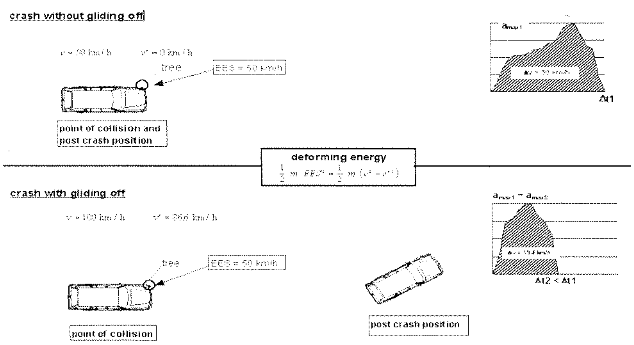

Figure 13. Distribution of energy transformation

Energy absorption and gliding zone together result in compatibility between different vehicles. With those two elements collisions in high speed range become controllable.

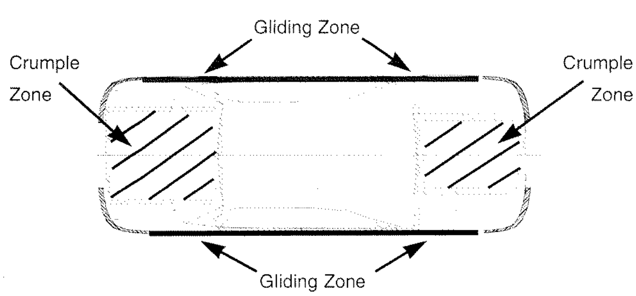

Figure 14. Crumple and gliding zone

AGGRESSIVE BEHAVIOR

Especially when concerning right-angle side impacts into cars the idea of reducing aggressive behavior briefly should be discussed. A typical accident under this topic is the side impact of a motorbike into a door of a vehicle. This type of collision is accentuated by the motorbike being pressed like a wedge into the compartment of the vehicle.

Figure 15.Aggressive behavior of the motorbike

As shown in figure 16 it is possible to ensure that the front wheel will be diagonally placed, during the front wheel fork is pressed back, by simple constructive means at the motorbike. An additional distortion element on the same-level as the headlamp can reduce the depth of intruding into the compartment.

Figure 16. Reducing the aggressive behavior by simple constructive means

REALIZATION OF A GLIDING ZONE

The question is, how a gliding zone can be realized. At the first glance the problems in the previous real accidents can be located at the trailer. There are two zones at the trailer, which create a problem.

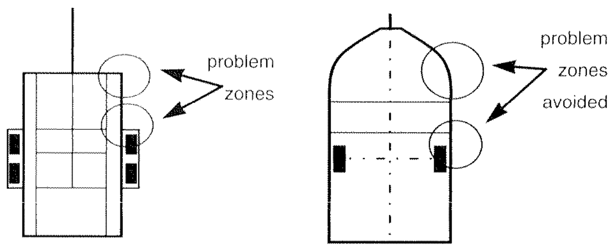

Figure 17. Trailer without and with gliding zone

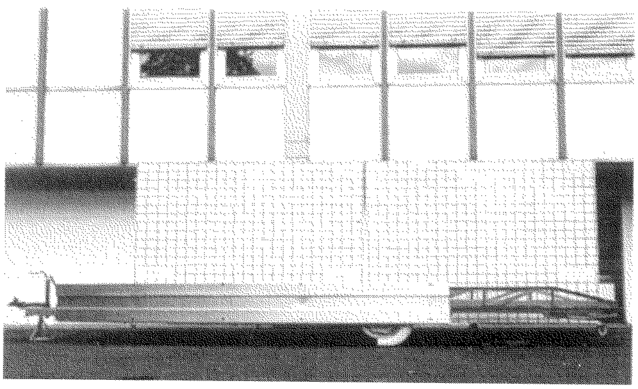

Figure 18. Realized gliding zone at a trailer

There is the possibility to deactivate these zones of problems. As a general principle the idea could come up to create a new design. It would be practicable to place the main frame elements to the outside of the trailer. The following figures illustrate a comparison test.

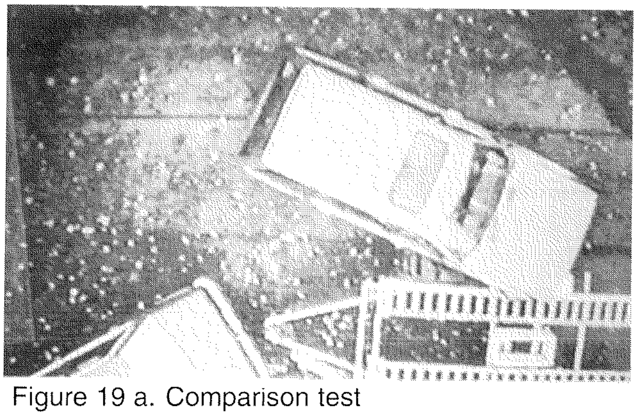

Figure 19 a. Comparison test

It is evident that this design principle for example makes this real accident controllable even under high speed ranges. Such a trailer design would clearly reduce the seriousness of injuries at collision between trailers and bikers.

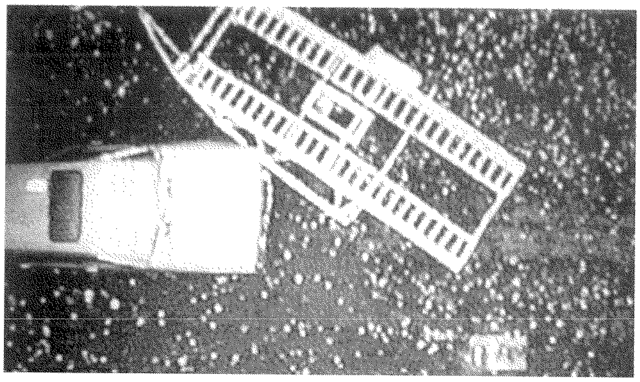

Figure 19 b. Comparison test

If these conclusions are transferred to the range of constructing utility vehicles and a truck will be equipped with comparable boards, then a heavy accident nearly becomes a petty damage as shown in the test series under this topic.

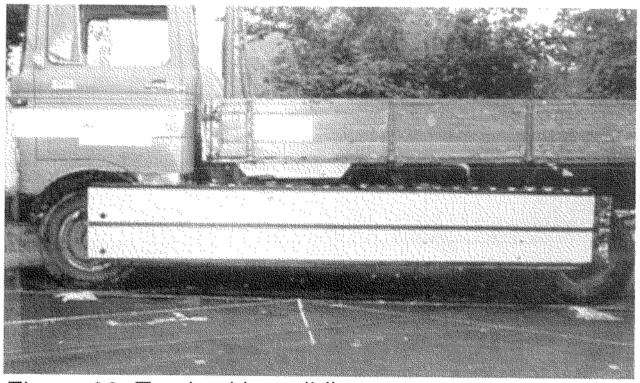

Figure 20. Truck with a gliding zone

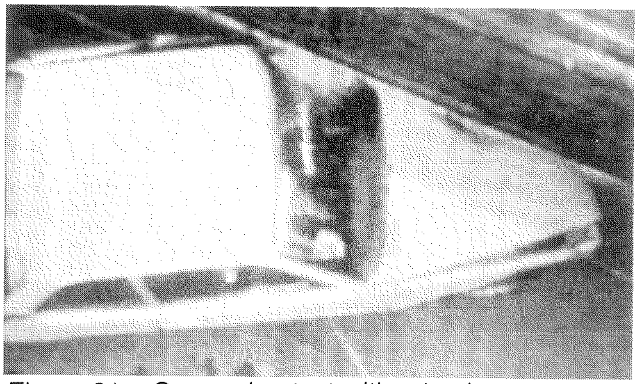

Figure 21a. Comparing test with a truck

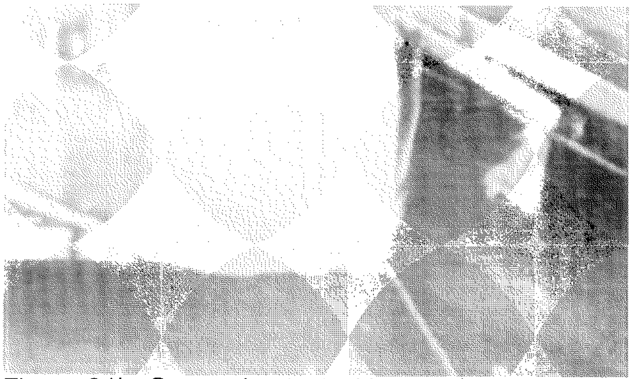

Figure 21b. Comparing test with a truck

This example illustrates that it is worth to include utility vehicles in passive safety. To avoid any loss of payload it is a good deal to reflect on utility vehicles from a design point of view and bearing in mind the statement of a developing engineer from Renault: «In most cases the best way to find a functioning new design is to forget everything that exists.»
One of the possibilities is to place carrying elements outside, that means to equip the utility vehicle with a board frame instead of a ladder type frame to serve as main frame element. First studies about a board frame concept have shown that not only problems of passive safety can be solved.

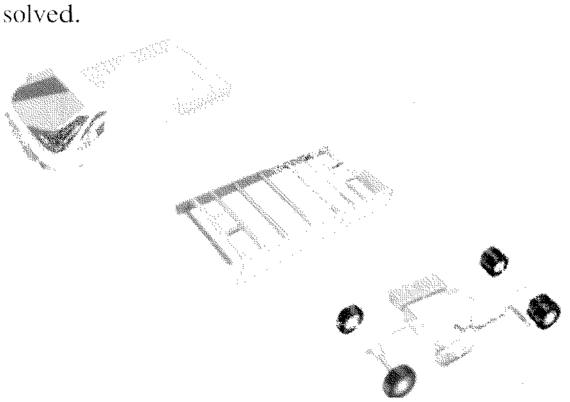

Figure 22. Board frame construction principle

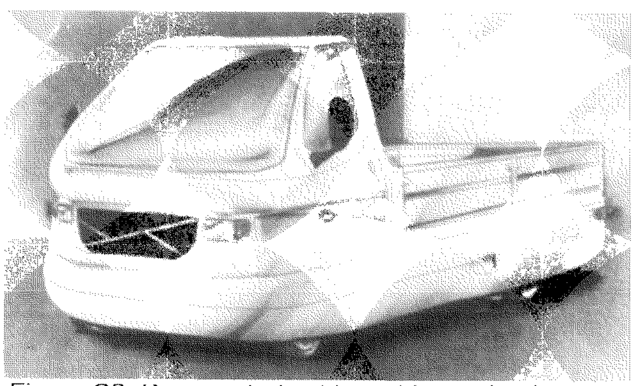

Figure 23. Design study of board frame truck

Simultaneously there is a drastical decrease of the noise level and the problems of splashing water because of the capsular design for the drive train and the tires. Besides, this principle of construction allows a decrease of round about 10 % in weight. Automatically there will be a reduction of aggressive behavior versus pedestrians and bikers from the frame arrangement positioned to the outside. Changing wheels and steering without a plain loss of track width can be arranged.
Basically a main frame positioned outside and designed as gliding zone could be used on passenger cars. However the appearance of the car primarily is being by designers rather than engineers. Designers are used to open wheel arches as styling element. In future we certainly have to reflect on this. It is also necessary to reflect on covered wheels at a sight of passive safety and later on for decreasing traffic noise level.
The first tests with the gliding zone placed outside have shown promising results colliding a 500 kg and a 1000 kg car at an 45° angle.

Figure 24. Passenger car with gliding zone

But even with open wheel arches the gliding effect can be realized as defined by safety partnership. Today the zone for energy absorbing does not surround the area to the right and left of the side frame. Here the cars can catch

Figure 25. Real accident car to tree

each other in a collisions with small overlap. Solving this problem can be demonstrated by a real accident with 20 % overlap against a tree.

In this special case the car has been ripped up. As shown in [4] this problem could be solved in this concrete accident by placing a diagonal bar between front wheel and A-pillar. But doing so would not fulfil the idea of safety partnership a collision intruding into the side of another car under an 45° angle . If the car is being equipped with gliding zone edges even a high speed accident with small overlap can de controlled by attachments as ABS and EPS.

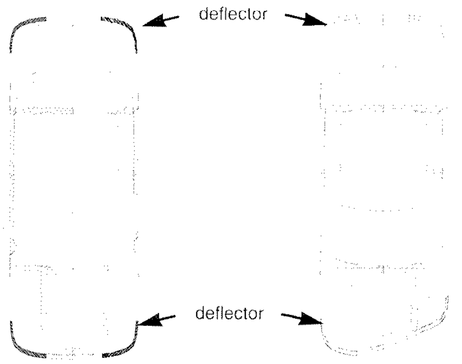

Figure 26a. Passenger car with deflector

These gliding zone edges are called deflectors. Even the first crash tests with deflectors in car to car collisions and car to static barrier crashs have shown good results. Deflectors clearly allow to decrease the seriousness of collisions between heavy and light mass cars under the view of compatibility.

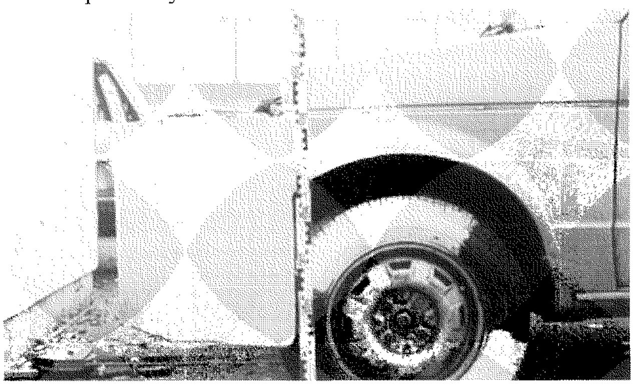

Figure 26b. Passenger car with deflector

A 1200 kg barrier with exchangeable box type deformation zones could be used for test purposes for example to approve a gliding zone. Under certain angles and speeds this standard test device would be crashed against the vehicle to be tested, regardless if it is a car or a truck. In this test the biomechanical load limits for the occupants may not be exceeded, respectively uncontrollable intrusions may not appear.

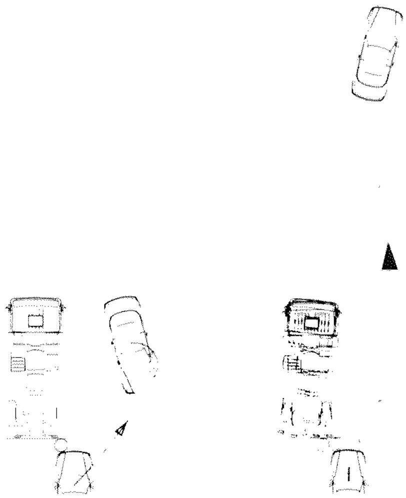

Figure 27a. Progress of collision without and with deflector

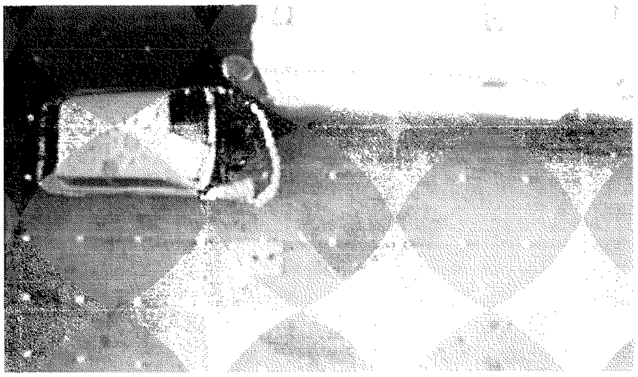

Figure 27b. Progress of collision with deflector

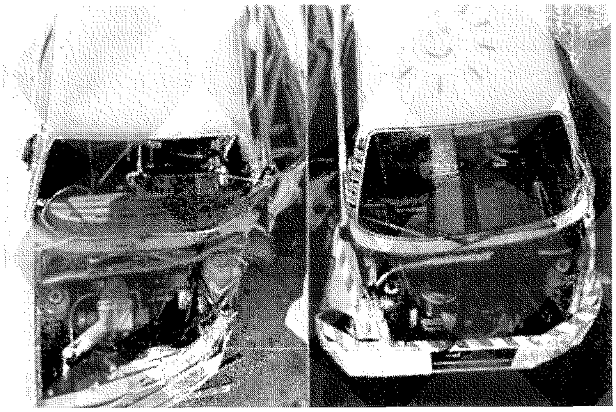

Figure 27c. With and without deflectors

SUMMARY AND CONCLUSIONS

In my presentation, I have tried to demonstrate ideas to minimize the rate of injuries and fatalities in traffic accidents. With the terms

* gliding zone
* board frame and
* deflector

approaches are being shown to help reducing the number of traffic victims. Also on a long-term basis and even with the introduction of electronic traffic guidance and control systems traffic accidents cannot be avoided. Therefore, it is necessary to redefine the meaning of passive safety with the objective to reduce the number of victims.

It is very seldom that all types of vehicles, such as passenger cars, utility vehicles, trailers and motorbikes, are being manufactured by the same company. This makes it desirable that not only car manufacturers, seeing first of all their own products, but as in former times also more neutral institutions such as technical universities develop these new concepts.

Finally I would like to mention a sentence said by Professor Appel from the Technical University of Berlin [5]:

«To introduce the principle of overall safety it is urgently necessary to have consent, cooperation and bills by law.»

Another reference of this subject:
The Board Frame, K.-H- Schimmelpfennig, Münster Germany, Paper No. 96-S11-W-16, 1996

[1] Zeidler: »Erfahrungen aus 25 Jahren Unfallforschung bei MB», Verlag Information Ambs, D-77968 Kippenheim Germany, Heft 9, 1995

[2] Breen: «Reduzierung von Kraftfahrzeugunfällen mit Körperverletzung - die Rolle des Kraftfahrzeugdesigns», European Transport Safety Council, Briefing, 1995

[3] Rumpe: «Sicherheit im Kraftfahrzeug», TÜV Rheinland, Presseinformation, 1993

[4] Zeidler: «Die Analyse von Straßenverkehrsunfällen mit verletzten PKW-Insassen unter besonderer Berücksichtigung von versetzten Frontalkollisionen mit Abgleiten der Fahrzeuge», Verlag Information Ambs, D-77968 Kippenheim Germany, 1982

[5] Appel: «Die Quadratur des Kreises beim Zukunfts-PKW», Automobiltechnische Zeitschrift 96(1994)6

IN-DEPTH ANALYSIS OF OFFSET FRONTAL CRASH TESTS IN VIEW OF EXTERNAL AGGRESSIVITY CRITERIA

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Paper Number 96-S4-O-07

ABSTRACT

Starting from the observation of the results of offset crash tests against a deformable element, the authors suggest it is possible to determine frontal aggressivity criteria taking into account the effect of the car mass and stiffness. Some hypothesis are considered like the stopping time of the vehicle and the energy lost by the vehicle after a pre-determined forward mouvement. It seems these parameters can give a good qualitative idea of the vehicle aggressivities but these hypothesis are still to be confirmed and developped by other test results analysis in order to determine a more quantitative approach. A condition to determine a quantitative criteria based on the deformable element is to avoid bottoming-out during the crash. In fact, it was observed that, in the tests used for this analysis, the barrier very often bottomed-out and this occured more easily and earlier with the heaviest vehicles. In consequence, it is also recommended to adapt the barrier characteristics before going further in the development of aggressivity parameters in offset deformable crash tests.

INTRODUCTION

In the passive safety research field, the passenger car protection has traditionally been studied in particular accident configurations. The main configurations on which researchers worked in a recent past are the car-to-car side and frontal impacts. The researches resulted the more often in the definition of a well-defined laboratory crash test procedure and the evaluation of the protection level offered by a particular vehicle is in term of internal safety. In other words, the main results of a crash test are the static measurements of deformations and intrusions of the subject vehicle and the biomechanical criteria obtained from measurements on dummies in this vehicle. Concerning vehicle-to-vehicle accidents, the only configuration in which the antagonist vehicle characteristics are considered is the car-to-truck impact because in that case, the incompatibility in terms of mass, geometry and stiffness is evident and is the main cause of injuries. For pedestrian impacts also, the characteristics of the impacting car is considered as it has a direct effect on the impact severity. However, the notion of compatibility is rather new as far as car-to-car accidents are concerned. It is clear a research program on compatibility must be mainly based on car-to-car impacts with different cars. Nevertheless, as in the offset frontal crash test procedure defined by the European Experimental Vehicle Committee (EEVC), the obstacle is deformable in order to simulate the other impacting vehicle, we thought it is a interesting first step to introduce in a laboratory crash test procedure the notion of aggressivity.

LITTERATURE REVIEW

The topic of car compatibility has been studied as early as the seventies (Ventre, 1974 - Kossar and al, 1974 - Seiffert, 1974) when the concept of car aggressivity emerged. Analysis of collisions between cars underlined three causes of aggressivity : the mass, the stiffness and the architecture. For 20 years, there was not much concern on compatibility but performance of restraint systems has been greatly improved (Tarrière and al, 1994).

Recently, accident investigations have shown that certain car models known to be safe in term of internal protection on one hand may be on the other hand more dangerous for the occupants of the antagonist cars (Thomas and al, 1994). Moreover, present developments in frontal and side impact research are taking place without much interaction : for example, ensuring better protection in frontal impact through higher stiffness of the front structure may decrease benefits in side impact (Zobel, 1987).

Concerning the mass, two laws are resulting from important early studies; when a crash occurs, other factors being equal, these two laws are: the lighter is a vehicle, the lower is the risk for other road users and the heavier is a vehicle, the lower are the risks for its occupants (Evans, 1994). A study from Thomas and al, 1990, gives the same results: the interior fatality rate decreases when the mass of the cars increases, a consequence of this being the higher exterior fatality rate in the adverse vehicle. In fact, the dependence of compatibility to deltaV has therefore to be given particular attention (Niederer and al; 1995) : the velocity change for an occupant depends mainly on the mass ratio of the cars involved (Thomas and al, 1990).

In a small and low-mass vehicle (LMV), there is unsufficient space available in the frontal direction for an extended crush zone. A smaller car seems to have a possibility to be safer by design, given that it can be constructed in a good way (Koch and al, 1991). Niederer and al, 1995, propose to design the vehicle with a stiff structure extending along its entire periphery and indicate that this rigid-belt body is a valid mean for providing adequate safety to low-mass vehicle occupants. Moreover, the test results as well as the results of the simulations show that a LMV does not represent an excessive compatibility problem for other car occupants in spite of its stiffness. Tarrière and al, 1994 also conclude that the smaller cars must be stiff enough to ensure a complete crush of that deformation area of the opposite car before his body collapses.

J.Y. Foret-Bruno and al, 1994, drew conclusions concerning intrusion and severity of restrained occupant injuries: rigidification of the front structures to reduce intrusion and the potential risk for the femurs will tend to increase the risk of injury to other body areas due to higher decelerations. V Koch and al, 1991, fund that the stiff car seems to have a higher injury risk when colliding with another stiff car, but also have a slightly better position when colliding with a soft car, while when soft cars collide with each other, the total outcome is far better than other combinations.

Tarrière and al, 1994, consider that, in the present situation, progress could only be achieved by means of a dedicated additional test which remains to be worked out; according to these authors it is of the utmost necessity that a regulatory proposal be prepared urgently to accompany any change in design rules, because the trend to build stiffer cars to achieve good ratings under more stringent test conditions may result in further increases of incompatibility.

In conclusion of this litterature review, is appears many ideas were expressed and it must be underlined these ideas are sometimes contradictory. We think two of them are to be restrained at the moment ; first, it is important to take care of interaction between different crash configurations (car-to-car side and frontal impacts, car-to pedestrian impacts,..) and secondly to be sure that new regulatory proposals do not give to car manufacturers design rules opposed to a better compatibility.

FIRST SUBJECTIVE APPROACH OF CRASH TESTS

During the past three years, a lot of offset crash test against a deformable element were performed in different laboratories. INRETS performed some tests like that in the framework of EEVC Working Group 11 research programme partly sponsored by EC DGVII. From this experience, we have already expressed some ideas on the importance to introduce compatibility requirements in such a test procedure (J.A. Bloch, 1994). One of the conclusions was that the improvement of internal protection must not increase the frontal aggresivity of the car and particularly its frontal stiffness. It was also suggested to take into account in the test requirements, the way the deformable element is deformed. In fact , it is clear that during the different offset crash tests results we analysed, the deformable element was deformed in very differents ways. Some were cruched to a relatively uniform shape whereas others were deformed in a more confused way. The same differences are observed on the front part of the vehicles depending of their frontal stiffnesses. On figure 1 is a deformable element deformed by a relatively soft vehicle whereas on figure 2 the deformable element was hit by a vehicle with very stiff front parts. The question is : can we consider a vehicle more aggressive if we observe on its front part undeformed rigid elements and/or if it deformes the barrier in a non uniform shape?

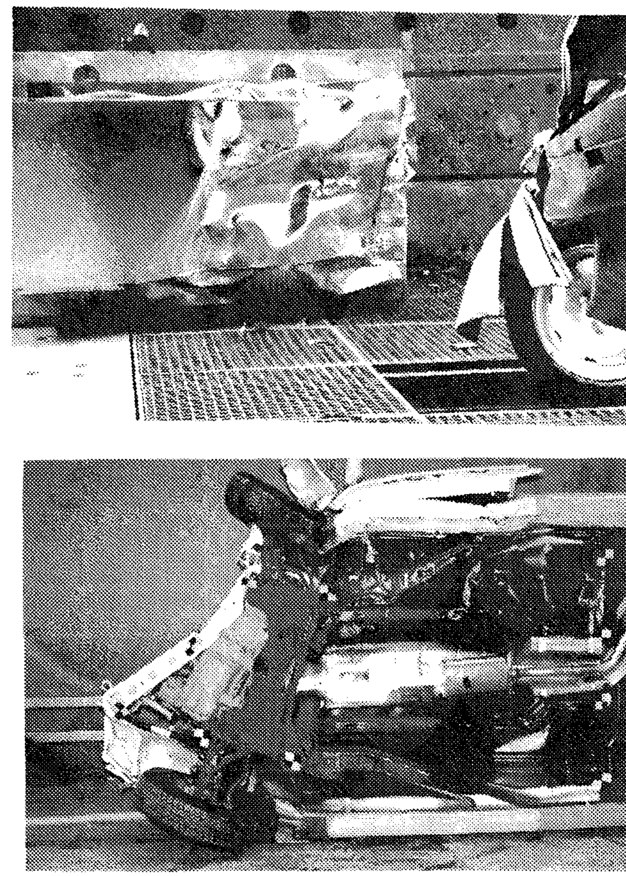

Figure 1. Deformed honeycomb barrier and the corresponding soft front part of the car after offset crash tests.

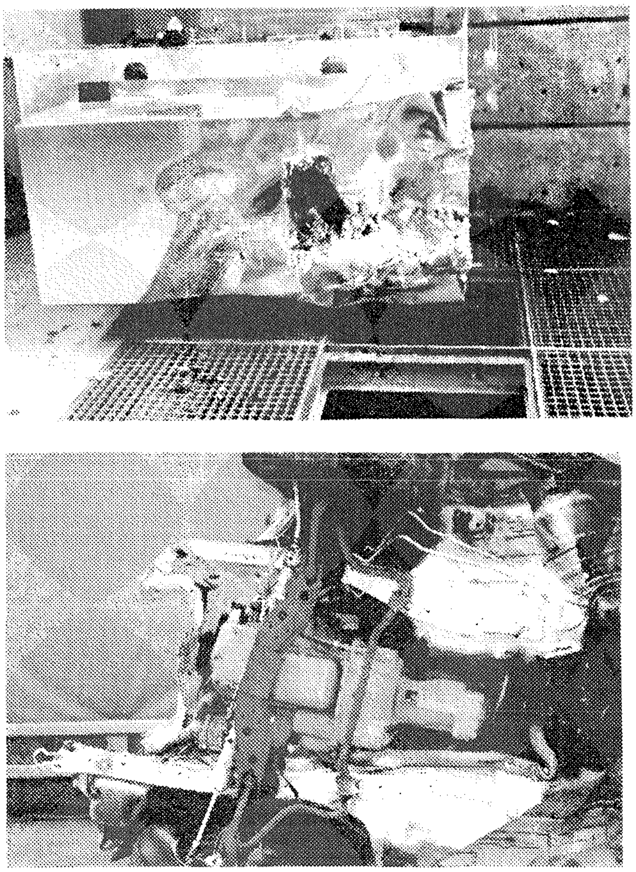

Figure 2. Deformed honeycomb barrier and the corresponding rigid front part of the car after offset crash tests.

FIRST ATTEMPT TO ASSESS FRONTAL AGGRESSIVITY FROM CRASH TESTS

From some offset crash tests against a deformable element (56 km/h, 40 % overlap), we tried to define a way to assess the frontal aggressivity of cars or at least some aspects of this aggressivity.

As we mentioned before, this aggressivity depends on multiple parameters. In these parameters, the mass is the most evident and also the most easy to quantify. In contrary, the geometrical incompatibility between cars (height, width, ground clearance,bonnet length,..) can be analysed only from car-to-car tests in different configurations.

From offset deformable crash tests, we think it is possible to estimate the effect of the car front stiffness combined with the mass against an other vehicle. We suggested in the previous paragraph, the way the car deformes the barrier could be a mean to qualify or to quantify the agressivity but we are far from the definition of a criteria based on this idea. The first approach we developped is to try to compare the global behavior of different cars. We started from the following hypothesis: the more the car is heavy, the longer is the time before it stops and, for two car with equivalent masses, the stiffer one may stop earlier than the softer.

On figure 3 is the distribution of cars stopping times versus the car masses for 8 vehicles. These times are obtained from the integration of the driver side B pillar. From this diagram, the first idea seems to be confirmed as generaly, the heavier is the car the longer is the time to stop it, but it is clear of lot of other data should be added to complete this diagram in order to be more confident in this hypothesis. However, as the and the stiffness are combined, we can go further to suggest a way to assess rigidity. A first approach is to consider that for two vehicles with similar masses, the vehicle which has a shorter stopping time is stiffer than the other one. For example, car D may be stiffer than car C. Of course, this is only a suggestion for an aggressivity assessment method and it must be validated and better defined from a lot of supplementary test results.

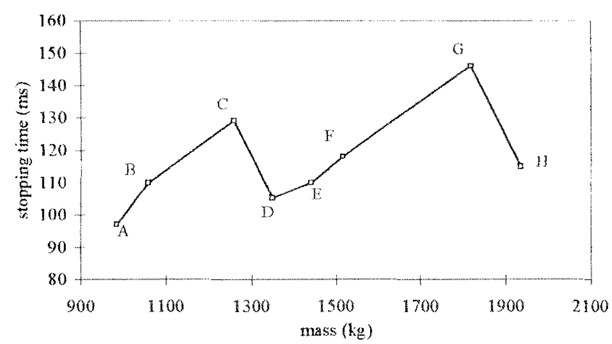

Figure 3. Stopping times versus car masses in offset deformable crash tests.

Another approach is to consider the energy loss during the test by each car at a certain stage of the crash test. On figure 4 is the energy lost by different vehicles with very different masses. The energy loss we considered is calculated from the vehicle speed when its center of gravity moved forward on 540 mm which correspond to the barrier depth. At that time, the deformation is distributed with different repartitions between the car and the barrier. From this figure, it is clear that, in general, the heavier is the vehicle, the lower energy it loses when it crushes the deformable element. In other words, the deformable obstacle which represents the antagonist vehicle is more deformed when the energy lost by a bullet vehicle is lower. As mentionned in the previous paragraph, we can also suggest that for two cars with similar masses, the higher is the vehicle energy loss, the softer is this vehicle. For example, vehicle B may be more aggressive than vehicle A.

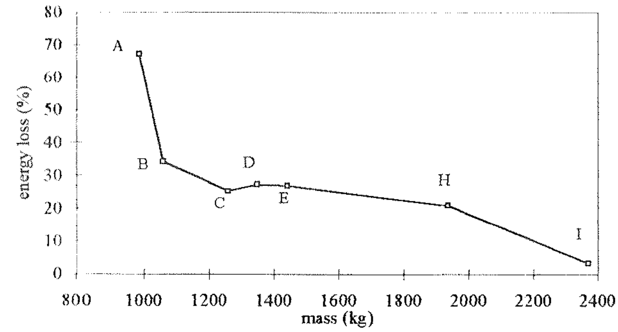

Figure 4. Energy loss in frontal offset deformable crash tests vs. car masses.

ANALYSIS OF THE DEFORMABLE ELEMENT BEHAVIOR

At this point, it must be underlined that the results used for this approach are dependant on the characteristics of the deformable element, particularly on its stiffness (50 psi in our study). As mentionned before, a first view on deformable barriers after crash tests reveals very different behaviors depending on the impacting car. Not only the barriers are deformed more or less uniformly but also, we observed bottoming-out very often at least on the corner and some times in some areas corresponding with rigid parts of the impacting car. It appears consequently the impact is transformed, from an impact against a deformable element to an impact against a rigid one. In fact, in a real car-to-car impact, a similar behavior occurs as the stiffnesses of deforming cars increase during the impact until they stop. The difference we observed is that in the offset test, the obstacle becomes rigid more suddenly and the rigid impact from this time has an effect on the car structure behavior (figure 5).

It can be claimed that this barrier behavior resulting in an angular car structure deformation is not important as biomechanical criteria have their maximum values during the honeycomb crushing. In some crash test we analysed, we noted the times of occurrence of some biomechanical measurement on the driver dummies. On figure 6 is the chronology of these measurements for four different vehicles with different masses. On the same diagram we added the times of occurence of the barrier bottoming-out observed on high-speed films. It appears clearly that for the lighter vehicle, the main biomechanical events on the dummy occur before or just after the barrier bottoming-out. In contrary, the heavier is the vehicle the later these events occur, and consequently, the biomechanical maximum values values are linked to the rigid impact. That means for heavier vehicles, the offset crash test against this honeycomb barrier simulates more a car-to-rigid obstacle than a car-to-car accident. For example, with the heavier car on figure 5 (2370 kg), the bottoming-out occured very early at 46ms whereas the biomechanical values are significant after about 100 ms.

Figure 5. Deformed car after an offset frontal crash test with bottoming-out of the deformable obstacle.

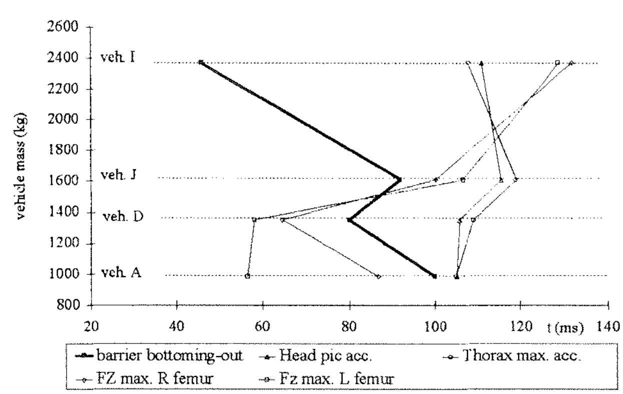

Figure 6. Chronology of crash test events for different vehicle masses.

DISCUSSION

The tests analysed in this paper were offset frontal tests against deformable elements at 56 km/h with 40 % overlap. This procedure was developped by EEVC for assessing internal protection in vehicles and the aim of our work was to see how this test procedure can be used for external safety assessement.

From a first analysis, the authors put forward the frontal aggressivity of cars can be qualitatively observed.

It is suggested, as a first evaluation of the vehicles aggressivity, to use the stopping time of the vehicle and/or the vehicle energy loss in a pre-determined position. A further step is to quantify the aggressivity from these suggested criteria but before to tackle this aspect, it is necessary to confirm the ideas we suggested by the mean of additionnal crash test results and also by the improvement of the deformable barrier design.

In conclusion, in order to enhance road safety, the compatibility between cars in different accident configurations must be improved. Until now, progress has been made for internal protection through new test procedures and requirements in frontal and side impacts. It seems necessary, at this stage, to determine the relevant aggressivity parameters, to identify their potential benefits and then to determine methods for assessing compatibility. Some hypothesis were suggested in this paper but a lot of research work is needed and INRETS will contribute to this work especially through EEVC as this topic is part of its research program.

ACKNOWLEDGEMENTS

The authors gratefully acknowledge the collaboration of E. Fossat (Fiat), J. Green (Rover) and J. Huibers (TNO) for providing complementary test results which were very usefull for this paper.

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INFLUENCE OF CAR WEIGHTS ON DRIVER INJURY SEVERITY AND FATALITIES IN HEAD-ON COLLISIONS

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ABSTRACT

This statistical study looks at injury severity and fatalities for belted drivers involved in collisions between two cars having a known mass. This research, when considered along with the results already presented by other teams, allows an in-depth evaluation of risk as a function of the mass of the impacting cars.

In this study, we have analyzed injury severity and fatalities for the drivers in both involved cars as a function of the cars' respective weights, as well as the overall severity of the collisions.

Moreover, in the second part of our paper, we have applied to our accident sample L. Evans' method of establishing drivers' exposure to risk in two-car collisions through the analysis of pedestrian fatality frequency according to car mass.

The differences between the results observed using the two methods will be discussed.

SAMPLE ANALYSIS

The computerized accident database of the French Gendarmerie (French police force) was used to carry out this study. These files contain information on approximately 3,000,000 roadusers (car occupants, pedestrians, two wheeled vehicles, heavy trucks) who were involved in accidents resulting in injury which occurred outside of cities of over 5000 inhabitants between 1978 and 1995.

We selected 446,498 records concerning car drivers in collisions involving two and only two cars as well as records on single car accidents involving no pedestrians or two wheeled vehicles. In order to be selected, the car's registered weight as approved by the *Département des Mines* (division of the Department of Transportation) had to be known.

(1) see references

In the first part of our study, we exclusively looked at the injury severity and fatalities for drivers involved in head-on collisions which took place outside of intersections. This sample was made up of 41,688 head-on collisions (83,376 drivers).

In the second part, in which we compare the relative risk to drivers in all two car collisions (inside and outside of intersections), we applied the two methods described in the abstract to our sample of 253,836 drivers. A comparison of likelihood of fatality by weight class will also be made for single car accidents involving a rigid obstacle or other obstacles.

As well we will use a second sample from the accident database mentioned above which includes both pedestrians involved in accidents with cars of known mass as well as pedestrians killed.

Table 1 gives a breakdown of the sample sizes for the different accident types involving pedestrians and driver.

Table 1
Breakdown of driver and pedestrian sample size used in this study by accident type and severity of impacts

	involved	fatalities
sub sample drivers	**446498**	**24624**
car-to-car all impact points	**253836**	**6626**
single car accidents against rigid obstacles	**97520**	**11735**
single cars accidents others	**95142**	**6263**
head-on car-to-car collisions(outside intersections)	**83376**	**3090**
sub sample pedestrians	**56481**	**7406**

HEAD-ON CAR-TO-CAR COLLISIONS

The sample of 83,376 drivers involved in head-on collisions includes 20,999 seriously injured and 3,090 fatalities.

Gravity and mortality rate for drivers as a function of their car's mass

We can see, by looking at the 4 mass categories in Figures 1 and 2, that the gravity rate (number of seriously injured and fatalities/number of drivers involved) and mortality rate (number of fatalities/number of drivers involved) for drivers increase significantly when the drivers involved are in the car that weighs less. The mortality rate recorded for cars of less than 850 kg is double that for cars of over 1200 kg and the gravity is multiplied by 1.5.

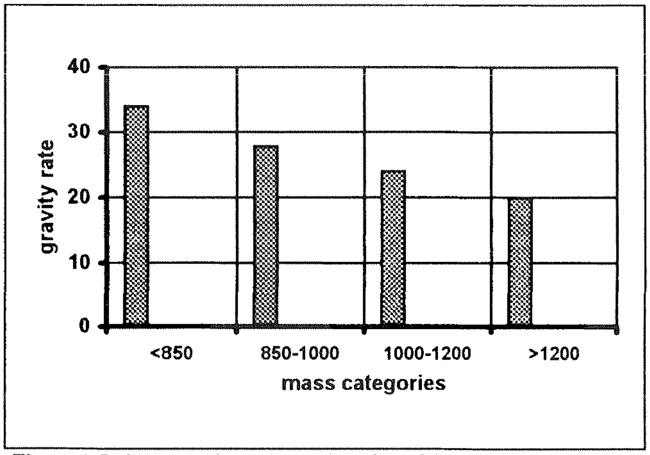

Figure 1. Driver gravity rate as a function of 4 mass categories in head-on car-to-car collisions

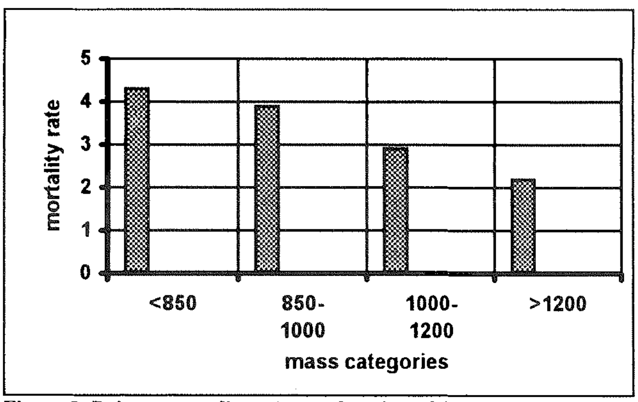

Figure 2. Driver mortality rate as a function of 4 mass categories in head-on car-to-car collisions

Gravity and mortality rate for the driver as a function of his/her car's mass and that of the other impacting car.

Tables 2, 3 and 4 give the number of drivers by gravity and by the cars' masses.

Table 2
Number of Drivers Involved in Cars of Mass M1 in Collisions with Cars of Mass M2

			cars M2				
			m1	m2	m3	m4	total
cars M1	<850kg	m1	13134	11827	4515	3652	33128
	850-1000kg	m2	11827	10728	4034	3150	29739
	1000-1200kg	m3	4515	4034	1732	1183	11464
	>1200 kg	m4	3652	3150	1183	1060	9045
		total	33128	29739	11464	9045	83376

Table 3
Number of Drivers Severely Injured and Fatalities in Cars of Mass M1 in Collisions with Cars of Mass M2

			cars M2				
			m1	m2	m3	m4	total
cars M1	<850kg	m1	3780	4115	1789	1588	11272
	850-1000kg	m2	2495	3180	1376	1231	8282
	1000-1200kg	m3	791	1044	484	418	2737
	>1200kg	m4	525	671	284	318	1798
		total	7591	9010	3933	3555	24089

Table 4
Number of Driver Fatalities in Cars of Mass M1 in Collisions with Cars of Mass M2

			cars M2				
			m1	m2	m3	m4	total
cars M1	<850kg	m1	353	514	276	275	1418
	850-1000kg	m2	243	432	226	249	1150
	1000-1200kg	m3	79	108	72	68	327
	>1200kg	m4	41	65	30	59	195
		total	716	1119	604	651	3090

From these three graphs we can calculate the mortality and gravity rate in order to see the difference in risk to the drivers based on whether they were in a light car in a collision against a car of equal or greater mass.

The results given in Tables 5 and 6, which are graphed in Figures 3 and 4, are compatible with impact physics in terms of the risks involved in accidents between cars of the extreme upper and lower mass range, and represent the results most often published in many articles.

Hence, we can see that mortality for the driver of a car in the lightest range (<850 kg) in a head-on collision with a car in the highest mass range (>1200 kg) is 7 times higher than that of the driver of the bigger car involved (see Figure 4). The gravity rate for that lighter car driver is also approximately 3 times higher.

Table 5
Driver Gravity Rate

	m1	m2	m3	m4
m1	**28.7**	**34.8**	**39.6**	**43.4**
m2	**21.1**	**29.6**	**34.1**	**39.1**
m3	**17.5**	**25.9**	**27.9**	**35.3**
m4	**14.3**	**21.3**	**24.0**	**30.0**

Table 6
Driver Fatality Rate

	m1	m2	m3	m4
m1	**2.69**	**4.35**	**6.11**	**7.53**
m2	**2.05**	**4.03**	**5.60**	**7.90**
m3	**1.75**	**2.68**	**4.16**	**5.75**
m4	**1.12**	**2.06**	**2.54**	**5.57**

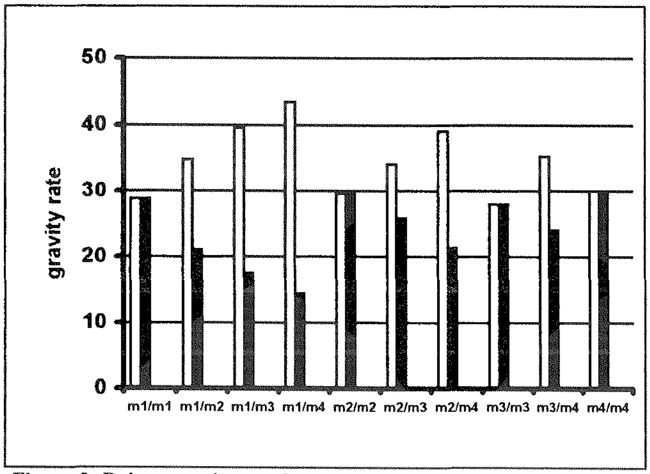

Figure 3. Driver gravity rate by mass category

But in the case of a collision between two cars of equal mass, the average mortality rate for both drivers steadily increases as the mass of the impacting cars increases, becoming twice as high (2.69 for m1 and 5.57 for m4) for accidents between two cars in the upper mass range.

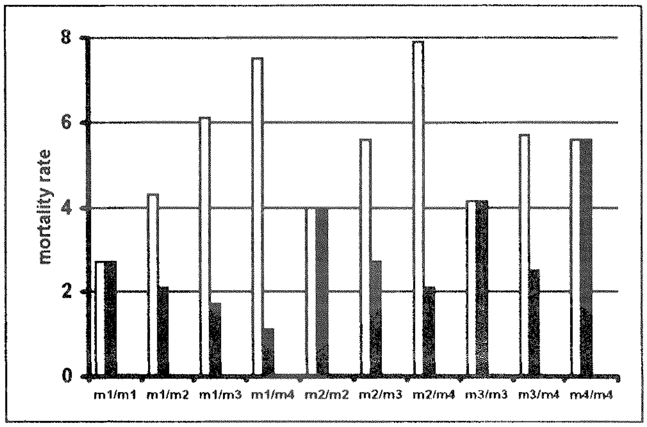

Figure 4. Driver mortality rate by mass category

However, as far as the overall gravity is concerned, the rate is practically stable for all mass categories in head-on collisions involving two cars of equal mass.

Are these results, so clearly shown in Figures 5 and 6, perhaps due to higher speeds in collisions involving two heavy cars or perhaps to other factors not taken into account?

In the last part of our paper we will try to find an explanation for these results, but first we will look at the risks to drivers in other types of accidents and use the method put forward by Evans to calculate exposure to risk.

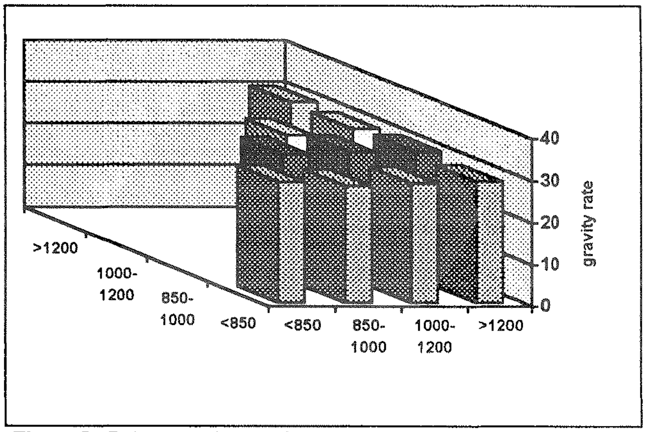

Figure 5. Driver gravity rate by mass category (for both drivers)

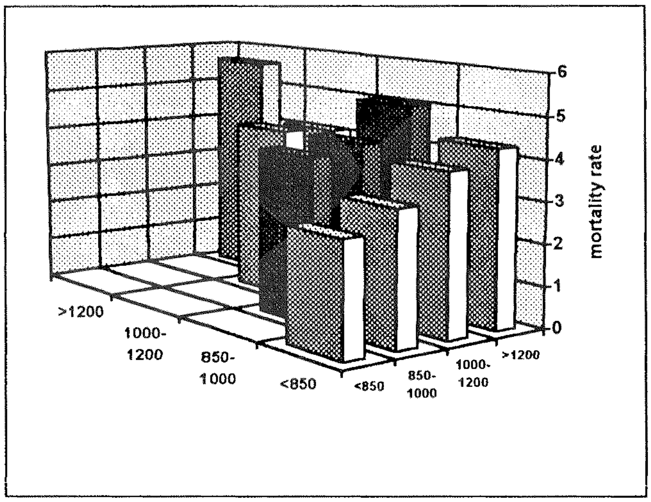

Figure 6. Driver fatality rate by mass category (for both drivers)

CAR-TO-CAR COLLISIONS, ALL IMPACT POINTS

In order to obtain the largest sample possible and to verify the trend observed in head-on collisions, we selected all collisions in which two and only two cars of known mass were involved.

These mass categories are of course different from those used by Evans due to the fact that cars in our country are substantially lighter on the average than American cars. However, our aim here is only to compare the results obtained using the two methods for the same number of mass categories.

As we had a sample of 253,836 drivers involved, of which there were 6,626 fatalities, we sub-divided all the impacting cars into six mass categories, the lightest being less than 750 kg and the heaviest over 1300 kg.

Drivers involved and driver fatalities as a function of the mass categories of the impacting cars

Table 7 and 8 below give the distribution of driver fatalities and drivers involved as a function of the 6 mass categories of the impacting cars. Using the figures from these tables we were able to calculate the mortality rate that is found in Table 9.

Once again, in the case of collisions involving two cars of equal mass, the mortality rate for both drivers increases steadily as the mass of the two impacting cars increases, being 2.6 times higher for the highest mass category than for the lowest.

For collisions involving a car from each of the two extreme mass categories (upper and lower) we confirm obviously that the mortality rate is twenty times higher for the smaller car, which is not surprising given the extreme difference in mass of the impacting cars (m6 being twice as heavy as m1).

Table 7
Number of Drivers Involved in Cars of Mass M1 in Collisions with Cars of Mass M2

			cars M2						
			m1	m2	m3	m4	m5	m6	total
cars M1	<750	m1	7216	9974	11114	7933	4647	2281	43165
	750-850	m2	9974	14942	15703	11211	6285	3130	61245
	850-950	m3	11114	15703	16682	11733	6582	2998	64812
	950-1100	m4	7933	11211	11733	8804	4533	2155	46369
	1100-1300	m5	4647	6285	6582	4533	2632	1189	25868
	>1300	m6	2281	3130	2998	2155	1189	624	12377
		total	43165	61245	64812	46369	25868	12377	253836

Table 8
Number of Driver Fatalities in Cars of Mass M1 in Collisions with Cars of Mass M2

			cars M2						
			m1	m2	m3	m4	m5	m6	total
cars M1	<750	m1	101	220	388	335	266	142	1452
	750-850	m2	112	308	418	438	310	168	1754
	850-950	m3	111	271	439	422	334	205	1782
	950-1100	m4	70	191	245	244	168	117	1035
	1100-1300	m5	22	77	121	89	86	52	447
	>1300	m6	7	23	40	30	33	23	156
		total	423	1090	1651	1558	1197	707	6626

Table 9
Driver Fatality Rate

	m1	m2	m3	m4	m5	m6
m1	**1.4**	2.21	3.49	4.22	5.72	6.23
m2	1.12	**2.06**	2.66	3.91	4.93	5.37
m3	1.00	1.73	**2.63**	3.60	5.07	6.84
m4	0.88	1.70	2.09	**2.77**	3.71	5.43
m5	0.47	1.23	1.84	1.96	**3.27**	4.37
m6	0.31	0.73	1.33	1.39	2.78	**3.69**

Directly calculating driver fatality rates by dividing the number of driver fatalities by the number of drivers involved for each mass category is usually the most reliable means of establishing these rates as long as no other biasing factors influence results.

In order to use a different means of evaluating risk, we have made this study more complete by using Evans' method.

Overview of Evans' method

In his paper, Evans only gives the distribution of fatalities as a function of mass category in impacting cars and doesn't seem to take into account the total numbers involved in the same accident type.

In order to calculate the relative fatality likelihood for drivers, he uses, as a reference, the percentage of pedestrians killed by cars in the same mass categories as those involved in two-car collisions (all impact types together).

The following formula is used to calculate relative fatality likelihood for drivers (using as an example the relative likelihood of driver fatality in the case of a collision between a car in the m1 mass category and a car in the m4 category):

$$\text{risk} = \frac{\text{number of driver fatalities in collisions m1/m4}}{\text{\% of pedestrian fatalities by car m1} \times \text{\% of pedestrian fatalities by car m4}}$$

Results

We applied this method using the same police database and by breaking down the pedestrians involved and killed by mass category of the impacting car (using the 6 mass categories defined earlier).

Table 10 gives the distribution by mass category.

Table 10
Number of Pedestrians Involved and Fatalities by Mass of Involved Car and Fatality Rate

		involved	%	fatalities	%	fatality rate
<750	m1	**9479**	**16.78**	**974**	**13.15**	**10.28**
750-850	m2	**15372**	**27.22**	**1908**	**25.76**	**12.41**
850-950	m3	**14390**	**25.48**	**1871**	**25.26**	**13.00**
950-1100	m4	**9826**	**17.4**	**1425**	**19.24**	**14.50**
1100-1300	m5	**5219**	**9.24**	**823**	**11.11**	**15.77**
>1300	m6	**2195**	**3.89**	**405**	**5.47**	**18.45**
	total	**56481**	**100**	**7406**	**100**	

This table shows that the pedestrian mortality rate increases very regularly as a function of the mass of the impacting vehicle. We then looked at the distribution of pedestrians involved as well as pedestrian fatalites in order to calculate the relative risk to drivers for a mass category in order to evaluate whether results differ depending on the calculation method used. The three ways of calculating were:

-risk obtained by calculating the driver mortality rate = number of fatalities/number of drivers involved

- risk calculated using the number of pedestrian fatalities as a reference

- risk calculated using the number of pedestrians involved as a reference

Tables 11, 12 and 13 give the results obtained by using each one of these methods and taking 1 as the base value of the relative risk to drivers in cars of mass m6 impacted by cars of mass m1.

Table 11
Driver Fatality Rate

	m1	m2	m3	m4	m5	m6
m1	**4.56**	7.19	11.38	13.76	18.65	20.28
m2	3.66	**6.72**	8.67	12.73	16.07	17.49
m3	3.25	5.62	**8.57**	11.72	16.53	22.28
m4	2.88	5.55	6.80	**9.03**	12.08	17.69
m5	1.54	3.99	5.99	6.40	**10.65**	14.25
m6	1.00	2.39	4.35	4.54	9.04	**12.01**

Table 12
Relative Likelihood of Driver Fatality using Pedestrian Fatalities (Evans Method)

	m1	m2	m3	m4	m5	m6
m1	**6.00**	6.67	12.00	13.61	18.71	20.29
m2	3.40	**4.77**	6.60	9.08	11.13	12.25
m3	3.43	4.28	**7.07**	8.92	12.23	15.25
m4	2.84	3.96	5.18	**6.77**	8.08	11.43
m5	1.55	2.77	4.43	4.28	**7.16**	8.79
m6	1.00	1.68	2.98	2.93	5.58	**7.90**

Table 13
Relative Likelihood of Driver Fatality using Pedestrians Involved (Evans Method)

	m1	m2	m3	m4	m5	m6
m1	**3.35**	4.49	8.47	10.70	16.00	20.29
m2	2.29	**3.88**	5.62	8.63	11.50	14.80
m3	2.42	3.64	**6.31**	8.88	13.23	19.29
m4	2.24	3.76	5.15	**7.52**	9.75	16.12
m5	1.32	2.86	4.79	5.16	**9.40**	13.50
m6	1.00	2.03	3.76	4.13	8.56	**14.18**

It can be seen that, if we look at collisions involving cars of equal mass, the results are quite different, in particular when the pedestrian fatalities are used as a reference to determine the relative likelihood of driver fatality.

With this reference, the likelihood to drivers increases very little when car mass increases, contrary to the results obtained with the other two calculation methods. However, in the method using pedestrians involved, the trend to show increased likelihood of driver fatality is too marked in comparison to the trend obtained with the method calculating mortality rate.

This difference in results is more clearly shown in Figure 7 which gives the regression line for likelihood of driver fatality in collisions involving cars of equal mass according to the three methods described above.

The highest correlations are found between the methods using mortality rates and pedestrians involved (R^2 = 0.98 and 0.92 respectively). By using pedestrian fatalities, the correlation coefficient is only 0.64.

The difference in the results obtained by using the different categories of pedestrians (involved and fatalities) as a reference are of course correlated with the mortality rate by impacting car mass as observed in Table 10 in which the mortality rates are multiplied by 1.8 when comparing accidents involving cars in the lower mass range to accidents involving cars in the higher mass range.

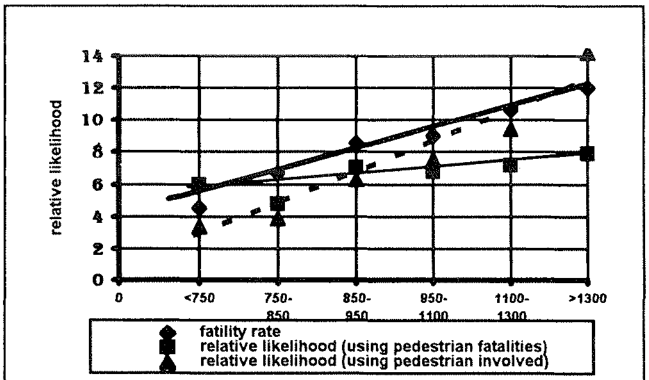

Figure7: Relative likelihood of a driver fatality when cars in the same category crash into each other according to the 3 methods

Let us now look at the case of single car accidents in order to see if the same sort of increase in likelihood of driver fatality occurs with an increase in car mass.

SINGLE CAR ACCIDENTS

Table 14 shows the distribution of single car accidents as a function of two accident categories:

- impact with rigid obstacle (pole, tree, wall,...)
- car roll-over or impact with other obstacles (ditch, guard rail, ...)

Under each accident category there is a large sample and we can once again observe that the mortality rate increases steadily as the involved car's mass increases; the likelihood of driver fatality being multiplied by a factor of 1.7 and 2 respectively for the two accident configurations.

These results are also different from those obtained by Partyka (2), in which a decrease in the likelihood of driver fatality is shown for drivers of cars in the heaviest mass range involved in single car accidents, especially in the case of roll-overs. However the calculations given in his paper are based on the number of fatalities per 100,000 licensed vehicles.

Table 14
Single Car Accidents

	against rigid obstacles			other obstacles and rollovers		
	drivers involved	driver fatalities	fatality rate	drivers involved	driver fatalities	fatality rate
<750	15975	1534	**9.6**	15584	808	**5.18**
750-850	28027	3181	**11.35**	28531	1647	**5.77**
850-950	25886	3187	**12.31**	24478	1669	**6.82**
950-1100	16563	2217	**13.39**	15852	1160	**7.32**
1100-1300	7981	1102	**13.81**	7423	646	**8.7**
>1300	3088	514	**16.65**	3274	333	**10.17**
total	97520	11735		95142	6263	

COMPARISON OF THE DIFFERENCES IN MORTALITY RATE AS A FUNCTION OF MASS FOR THE DIFFERENT ACCIDENT CATEGORIES

Figure 8 shows the mortality rates as a function of the six mass categories and the different accident configuration categories for drivers and impacted pedestrians. That is to say for:

- impacts between cars of equal mass (Table 9)
- cars impacting a rigid obstacle (Table 14)
- car roll-overs or cars impacting other types of obstacles (Table 14)
- pedestrians impacted by cars (Table 10)

The trend is the same in all four categories: a higher risk is observed when cars in the heaviest categories hit each other or are in single car accidents. As for pedestrians, it is on the average less dangerous for them to be hit by cars in the lighter ranges.

The correlation coefficients (R^2) of the linear regression lines that can be defined for these four accident configurations are between 0.95 and 0.97.

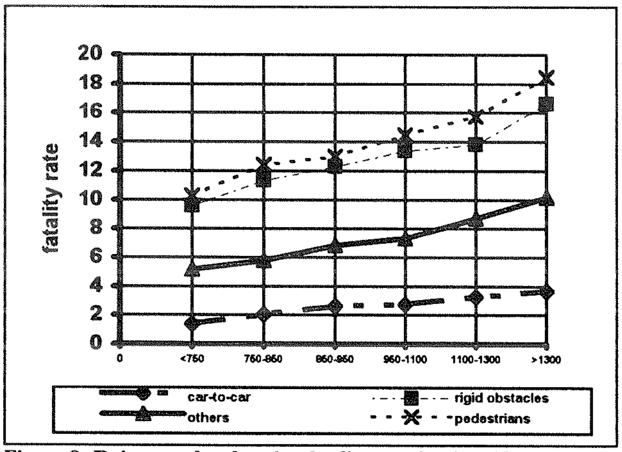

Figure 8: Driver and pedestrian fatality rate for 4 accident configurations by mass category

DISCUSSION

Likelihood of driver fatality

By comparing the regression lines for the likelihood of fatality obtained using Evans' method and the mortality rate calculation method (see Figure 7), we can see that there is a difference in results for cars in the heaviest mass category (8 and 12 respectively, equal to a factor of 1.5) while for lighter cars both methods' results were around 6.

If Evans' regression line is the most reliable, then it would lead us to believe that a bias factor influenced our results and led to more fatalities in collisions involving two cars in the heavy mass range.

But, based on the similarity in the curves in Figure 8, can we assume that a possible bias factor had enough influence to multiply by 1.5 the fatality rate for collisions involving two heavy cars or single car collisions involving the heaviest mass range?

To reach a mortality rate nearer to that obtained by Evans' would mean assuming that 50% of the drivers of heavy cars involved in accidents did not suffer bodily injury and were therefore not recorded by the police.

But is this probable when the accident involves two cars, which means the presence of occupants who are not at fault and who tend to want to receive damages and therefore obtain a declaration of injury?

It would seem that such hypotheses would be unlikely to be accepted by on-site road crash investigators regularly working in the field. Moreover, the risk observed for pedestrians contradicts this hypothesis as we will see in the next part of our paper.

Pedestrian fatality risk

From Figure 8 we can see that the mortality rate for pedestrians is multiplied by 1.8 if the impacting car is in the highest mass category (twice as heavy as the lightest car).

And yet it would seem evident that impacted pedestrians do not declare themselves to have suffered no injury 1.8 times more often when it is a heavier car that hit them.

It would also be difficult to assume that, given the same impact speeds, heavier cars present a much more dangerous shaped body for pedestrians!!

It may therefore be a question of impact speed that would explain these differences between lighter and heavier cars.

<u>It might be possible to assume that, on the average, at the moment of impact with the pedestrian, a heavy car is travelling at a higher speed than a lighter car .</u>

But then why would this higher speed for heavy cars impacting pedestrians not also be reflected in impacts involving a rigid obstacle or in collisions with another vehicle of its mass category?

In France, the horsepower of a car is highly correlated to its mass as we can see in Figure 9. This could lead us to hypothesize that drivers finding themselves in heavier cars, which are usually quieter, more comfortable and more powerful, have a tendancy to drive slightly faster on the average, thereby resulting in impacts or collisions of greater severity.

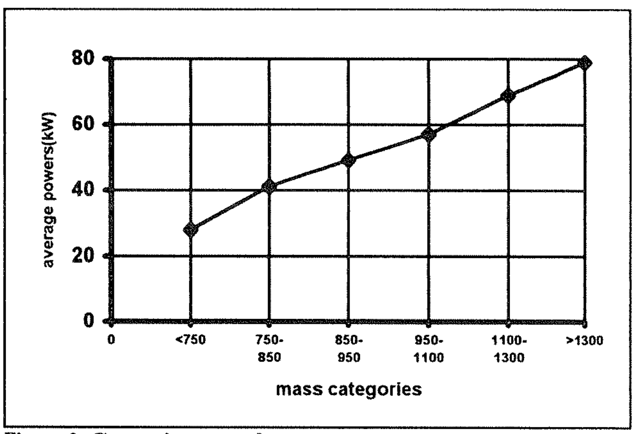

Figure 9: Car engine power by mass category

The effect of higher speed on fatality rates

In this part we have tried to determine what variation in speed would almost double the fatality rate by using our LAB database.

For 2,374 front belted occupants involved in frontal impacts, we estimated the delta velocity (Δ V) from the energy absorbed by the cars. If the fatality rate by class of Δ V and the distribution of belted occupants involved in the same class are known, then it is possible to calculate a new fatality rate from a distribution of occupants involved for whom the Δ V is slightly higher (5 km/h). See Tables 15 and 16.

The fatality rate for these belted occupants in our LAB database is close to that of our accident sample, 5.3 and 5.5 respectively. Thereby making it possible to compare these two results.

For all belted occupants involved, a 5 km/h increase in impact speed results in a mortality rate 1.74 times higher.

Table 15
Distribution of Front Belted Occupants Involved and Number of Fatalities by Category of delta V

delta V	involved	number of fatalities	mortality rate
<15km/h	77	0	0
16-25	562	0	0
26-35	664	1	0.1
36-45	513	12	2.3
46-55	295	31	10.5
56-65	213	50	23.5
66-75	42	24	58
>75	8	7	90
TOTAL	2374	**125**	**5.3**

Table 16
New Distribution of Front Belted Occupants Involved and Number of Fatalities by Category of delta V for a higher speed (5 km/h)

delta V	involved	mortality rate	**new number of fatalities**
<15km/h	39	0	**0**
16-25	319	0	**0**
26-35	613	0.1	**1**
36-45	589	2.3	**14**
46-55	403	10.5	**42**
56-65	255	23.5	**60**
66-75	127	58	**74**
>75	29	90	**26**
TOTAL	2374	**9.1**	**217**
	mortality rate x 1.74		

Using the same procedure, the calculations for an added delta V of 10km/h result in the driver's fatality rate being multiplied by 2.5.

So we can see that, given a delta V upon impact that is only 5 to 10 km/h higher for all people involved in accidents, the fatality rate for belted occupants in frontal impacts is multiplied by a factor of 1.7 to 2.5 which is close to what was found for the different configurations described earlier when comparing the heaviest cars to the lightest.

It is therefore possible that the explanation for the results obtained in our study lies in a slightly higher speed of 5 to 10 km/h at impact for heavier cars given their higher engine power.

Comparison of the situation in France and the U.S.A.

The application to our sample of Evans' method, using the number of pedestrian fatalities as a reference in calculating the relative likelihood of driver fatality, therefore greatly underestimates the actual trend for our sample while using the number of pedestrians involved rather than pedestrian fatalities overestimates the relative likelihood of driver fatality.

If in the USA differences in the distribution of pedestrian fatalities and pedestrians involved by car mass category exist as they do in France, then the results published by Evans (1) should be different using pedestrians involved as the reference.

We can also therefore assume, based on the results obtained in France, that the fatality likelihood in the US would not be 30% lower for collisions between two heavy cars in comparison to collisions between two light cars (see Figure 10: regression line defined using Table 5 in Evans' publication and taking the 500-900 kg range and the 1300-1500 kg range, regression with $R^2=0,69$) but rather more or less the same likelihood of fatality.

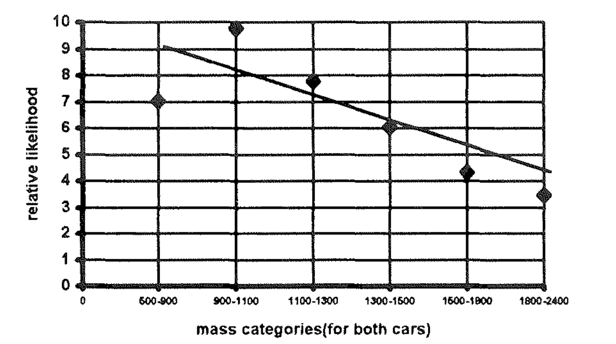

Figure 10: Relative likelihood of a driver fatality when cars in the same category crash into each other, all impact points (from Table 5 in Evans' publication)

It would be interesting to carry out an analysis on U.S. data using all pedestrians involved if such an analysis is possible from the different accident files in the Department of Transportation's database (FARS and NASS).

CONCLUSIONS

In head-on car-to-car collisions it is obviously better to be in the heavier of the two cars in order to have the lowest likelihood of injury or fatality.

However, when two cars of equal mass collide, the higher they are in the mass range, the greater the driver fatality rate.

This observation, which also applies when looking at all car-to-car collisions or at single car accidents, is quite simply due to higher average impact speeds when heavier cars are involved.

This slightly higher speed is also observable when pedestrians are hit by heavier cars.

We can estimate the average speed at impact to be between 5 to 10 km/h higher for heavier cars based on our Accidentology Laboratory database.

These extra km/h are enough to result in the increased risk to drivers.

Applying Evans' method of calculating relative likelihood of driver fatality, using only pedestrian fatalities as a reference, does not make it possible, at least in the case of our accident sample, to approximate very closely the real driver fatality rate obtained by the ratio « number of driver fatalities / number of involved drivers ».

One question remains: does the French situation is similar to other ones?

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DEVELOPMENT AND TESTING OF ENERGY ABSORBING REAR UNDERRUN BARRIERS FOR HEAVY VEHICLES

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Paper No 96-S4-O-10

ABSTRACT

Rear underrun crashes involving heavy vehicles with rear overhangs represent the most extreme examples of the incompatibility between heavy and light vehicles, particularly passenger cars. This paper describes the design, development and prototype testing of a practical, effective energy-absorbing rear underrun barrier system. This builds on the extensive work previously undertaken demonstrating the effectiveness of well designed lightweight -but rigid- rear underrun barriers. The energy absorbing unit consists of two light weight steel tubes, containing the energy absorbing glass fibre reinforced composite tube. The full system exhibits very good force-deformation characteristics, with minimum energy absorption in excess of 40KJ. Testing has included static and dynamic loading, including centred and offset crash testing of the prototype unit to compare the injury outcome with that of a full frontal barrier test of the same vehicle model.

INTRODUCTION

Rear underrun crashes are a particularly severe crash type: because the floor structure of most heavy vehicles is above bonnet height, cars can run under this structure (e.g. the tray of a rigid truck) with the tray penetrating through the car's windscreen pillars and into the passenger compartment. The usual occupant protection features built in cars such as seatbelts, airbags, crush zones are bypassed and ineffective in this crash type. An effective means of preventing underrun lies in adding a frame structure to the rear of the truck, which is of sufficient structural strength and geometry to engage the front structure of the car and prevent underrun. Most heavy vehicles do have some sort of barrier already, but these are typically poorly designed and quite ineffective. Rear underrun crashes in Australia account for some 15 or so people killed every year, and some hundreds injured. Over the decades many hundreds have been killed and seriously injured through this crash type.

Since 1991 a series of projects carried out at Monash University have led to the development of design criteria for rear underrun barriers for all heavy vehicles over 3.5 tonne, and included prototype development and crash testing. The resultant recommended design criteria in terms of barrier strength exceed that of current international standards. The reader is referred to references 2, 3, 4 for details of this work.

Following this major development work on rigid rear underrun barriers for the State's road safety agency VicRoads, the Federal Office of Road Safety funded an extension to this work to design, develop, and test an energy absorbing rear underrun barrier (see Fig. 1). This type of barrier may then result in a less severe impact for the car occupants, and would also extend the effectiveness of the barrier to a higher impact speed range. The project involved the design and testing of a suitable energy absorbing module; the construction and static testing of a prototype energy absorbing underrun barrier; and crash testing of the prototype energy absorbing rear underrun barrier, including use of Hybrid 3 dummies.

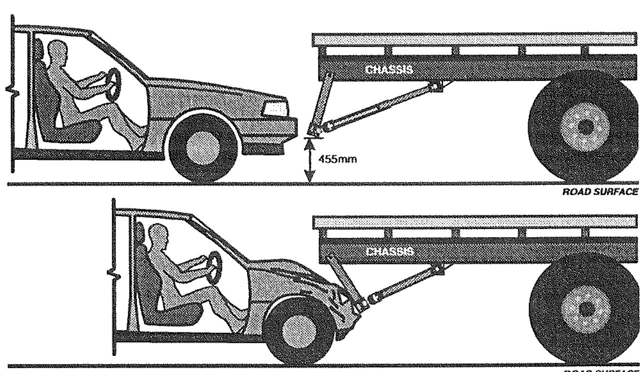

Figure 1. Illustration of energy absorbing rear underrun barrier system on rear of truck, before and after impact.

DEVELOPMENT OF ENERGY ABSORBING MODULE UNIT

The dilemma for the design of energy absorption systems has traditionally been the apparent difficulty in making the system effective for the range of vehicle masses from light to heavy cars, and for a range of impact speeds (Murray, 1988). However from a calculation of the equivalent barrier impact speed for a range of energy absorption values, it is clear that a system that absorbs 40 to 60 KJ would have a significant benefit for cars ranging in mass from 800kg to 1800kg. The following equation (ref. 1 & 5) allows calculation of the equivalent barrier speed (V_B) for different levels of energy absorption (G):

$$V_B = \sqrt{\left(\frac{m_1 m_2 V_1^2}{2(m_1 + m_2)} - G\right)\frac{2}{m_1}} \quad (1.)$$

- where m_1 = car mass; m_2 = truck mass, and V_1 = impact velocity.

For example, an 800kg car impacting a 40 tonne truck with a 60KJ barrier, at 50kph, is equivalent to a 21kph rigid barrier impact. For a 1500kg car the equivalent speed is 36kph - a considerable reduction in impact severity in both cases. Alternatively for an equivalent barrier speed of 50kph, the respective vehicles could impact at 67kph and 61 kph (for further details of this type of analysis and a review of energy absorption systems and literature relating to heavy vehicles, see refs. 5-9).

The other key consideration is the difference in force level required for light and heavy vehicles and hence the threshold needed to activate the energy absorption mechanism. From a consideration of all these factors it was concluded that the system be designed for a 40kJ to 60KJ energy absorption capacity (shared amongst 4 units), at 400mm deformation. The proposed 'ideal' force-deflection curve for each unit is given in Figure 2.

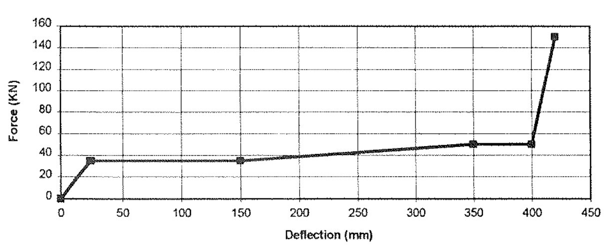

Figure 2. Idealised force deflection curve for the energy absorption module.

This is based on an energy absorption capacity of 16kJ; an initiation force of 35kN per unit, and a maximum of 140kN for the system. Based on these considerations it was found that a fibreglass tube was ideal in that it most closely followed the desired performance characteristics, was light weight and low cost. A series of tests were conducted on a range of fibreglass tubes. The final selected tube is a 38mm square by 3.2mm wall thickness

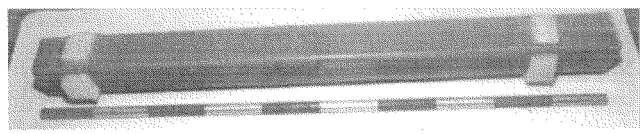

Figure 3. View of 500mm by 32mm fibreglass square tube with spacers.

FRP tube (fibreglass reinforced epoxy), 500mm in length (Figure 3). The tube is a standard commercially available product and closely matches the original force deflection, and energy absorption parameters proposed for this unit. The unit is contained within a 65mm square thin walled (1.6mm) steel tube (Fig. 4), with a smaller 50mm square tube (1.6mm) reacting against the FRP unit, and acting as the 'piston'. The steel tubes are standard and commercially available. The full energy absorbing tube-in-tube system is illustrated in Figure 6.

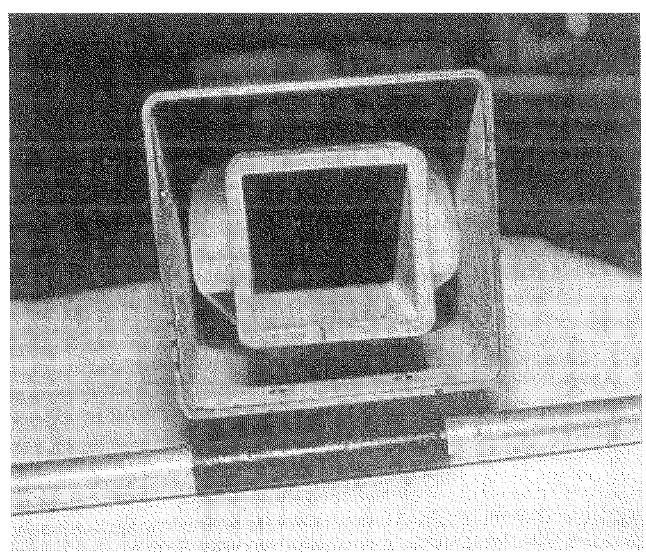

Figure 4. The 38mm square FRP tube within the 65 mm steel box section.

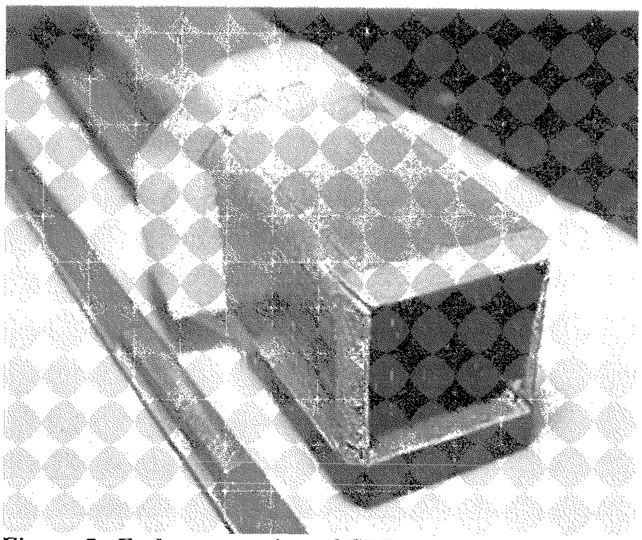

Figure 5. End preparation of FRP unit within the 65mm steel box showing 60deg. chamfer and corner cuts.

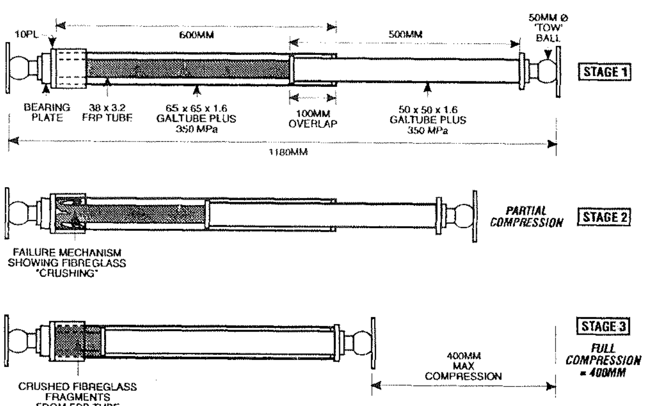

Figure 6. Schematic of the energy absorbing tube-in-tube system.

To ensure proper initiation of the failure mode in the FRP tube, the tube is chamfered at 60 degrees at one end, with 10mm long cuts made along each corner (Fig. 5) to help reduce the initial failure load to the desired level. The failure mode for the tube in unconfined compression is shown in Figure 7. Figure 8 illustrates lack of development of a good failure mechanism due to over-confinement within a round steel tube of too small a diameter; Figure 9 illustrates a well developed failure mechanism within the suitably sized 65mm square steel tube.

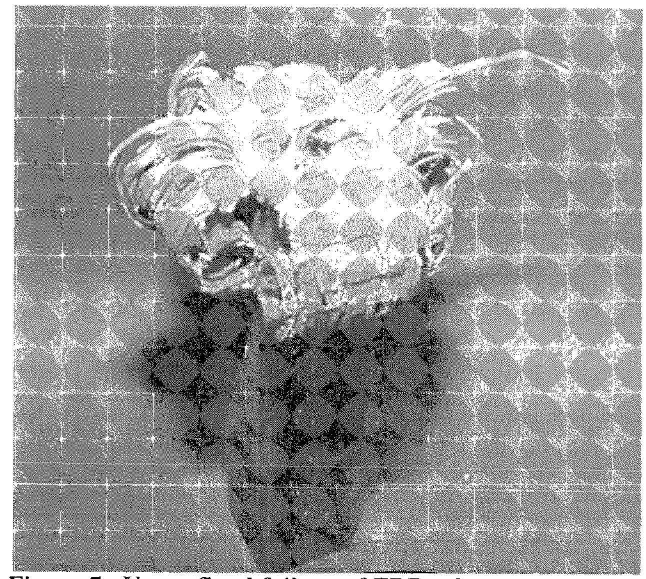

Figure 7. Unconfined failure of FRP tube.

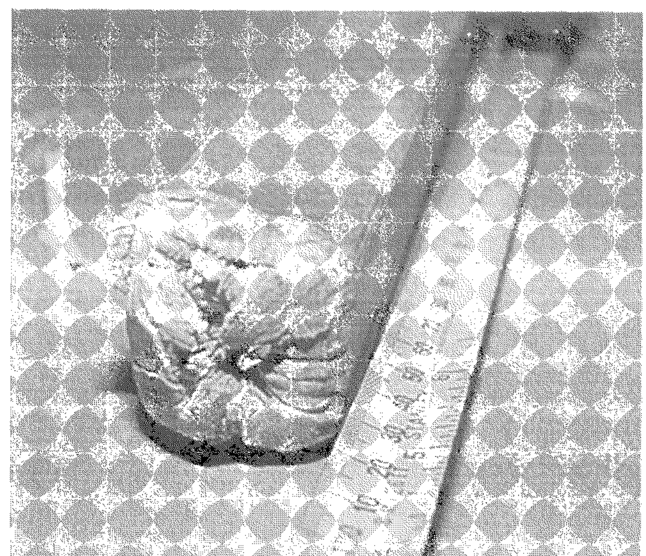

Figure 8. Overly confined 'failure' of tube.

Figure 9. Failure mechanism of the FRP tube in static compression test within 65mm tube.

The force - deformation characteristic of the tube-in-tube system is shown in Figure 10, for a static compression test. This shows very good characteristics, and is in good agreement with the idealised curve in Figure 2. Energy absorption for the 400mm crush was 14kJ.

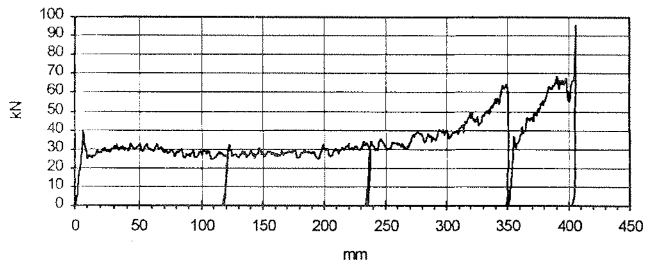

Figure 10. Force deformation curve for static compression of the system (see Fig 6).

To verify the dynamic characteristics of the system, two drop tests were conducted. A mass of 1100kg, at a height of 1.3m was dropped on the unit. The system performed well, and similar to the static test - with the measured force deformation characteristics shown in Figure 11. The total energy absorbed was 19.3kJ, of which the FRP tube absorbed 14.3kJ at a crush of 400mm; the steel tube absorbed the remainder, with an additional crush of 25mm.

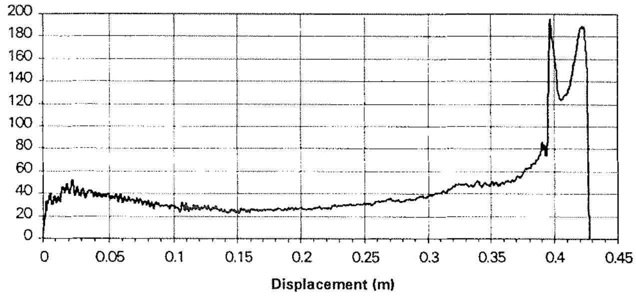

Figure 11. Load-deflection curve from the drop test on the tube-in-tube system.

Overall crush length for the FRP tube is 400mm, at an average force of between 30-40kN. The steel tube system has an axial load capacity in excess of 180kN, at which stage local buckling of the tube walls occur. The local buckling (a desirable characteristic at these higher load levels) is in part initiated by the bulging of the bottom 100mm of the tube walls due to the compression of the disintegrated FRP tube particles. However at these higher loads additional energy can then be absorbed by the 'controlled crushing' of the steel tubes. The overall load capacity of the unit is in keeping with the recommendations made by Rechnitzer, Scott and Murray (1993).

STATIC TESTING OF THE FULL PROTOTYPE SYSTEM

To test the full system a frame was constructed to simulate the rear structure of a truck. The underrun barrier system consisted of 4 energy absorbing units, with cross beam and hangers. Static testing was carried out using two jacks loading simultaneously, to simulate both centred and offset impacts (Figure 12). Following initial testing, design modifications were made which included the use of ball joints at each end of the strut, to isolate the struts from bending moments and to ensure good rotation in all directions. The system deformed satisfactorily with energy absorption of around 50kJ for the offset test and 60kJ for the centre static test.

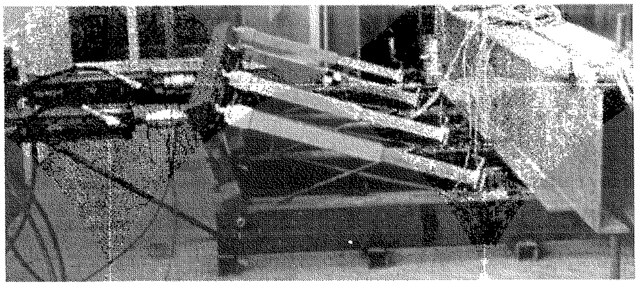

Figure 12. Prototype energy absorbing rear underrun barrier unit undergoing static load tests.

CRASH TESTING OF THE PROTOTYPE SYSTEM

Three crash tests were carried out on the prototype system at 48kph. The prototype system was transported to the test barrier and fixed to the face of the barrier (simulating a very high mass truck).

The first test involved a large family sedan (1800kg), donated by Ford Australia, in a centred impact (similar to that in Fig 14). In this test the system performed very well with underrun prevented and the two centre energy absorbing modules fully compressed (400mm) and the two outer modules 270mm. A comparison of the acceleration pulse (Fig. 13), measured at the base of the B pillar of the car, for the normal rigid concrete barrier and the energy absorbing system shows a marked decrease in severity. The peak acceleration reduced from 50G to 25G with the pulse duration increased from 70ms to 150ms, with both results indicating a significant reduction in crash severity and important benefits in terms of occupant protection. It is noted that as the barrier height of 450mm was above the car's front longitudinals, the impact forces were concentrated on the car's engine via the underrun barrier's crossbeam. These forces were then transferred to the car's firewall and floorpan.

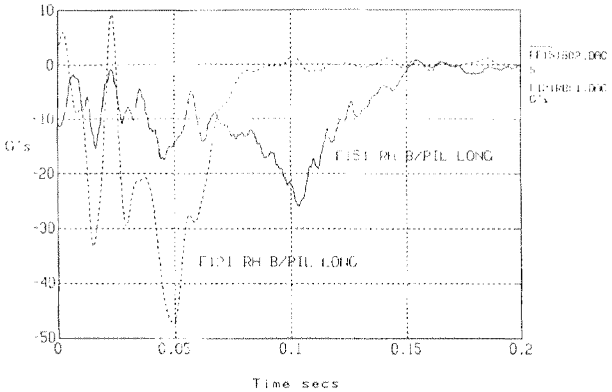

Figure 13. Crash pulse for 48kph impact (1800kg sedan). Solid line is for impact with the energy absorbing rear underrun system; the dashed line is a standard rigid barrier test.

The next two tests were carried out at Sydney's Crashlab facility under contract to FORS, and included both driver and front seat passenger Hybrid 3 dummies. Figure 14 shows the centred impact, with Figures 15 and 16 showing the crash pulse at the base of the vehicle's left and right B pillars. Once again the results show relatively low peak accelerations (20G for the centred test, 16G for offset) and an extended duration of up to 220ms for the offset test.

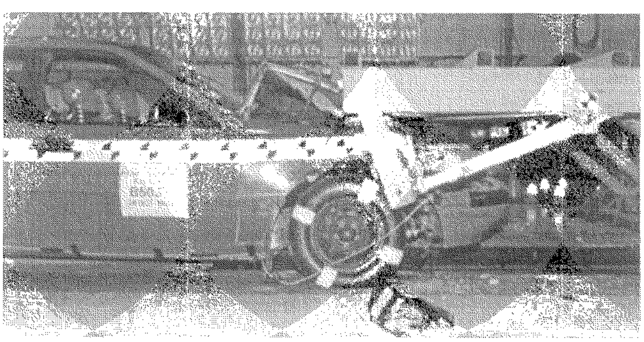

Figure 14. Photograph of the car in the centred 48kph impact with the energy absorbing barrier.

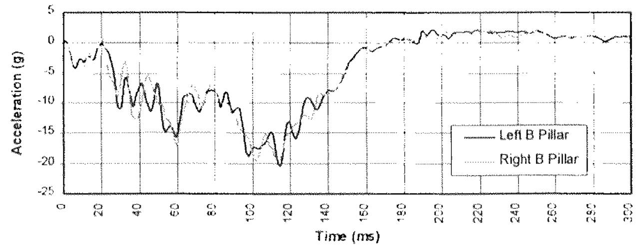

Figure 15. Acceleration pulse at base of B pillar, 48kph centred test with prototype energy absorbing underrun barrier system (m=1700kg)

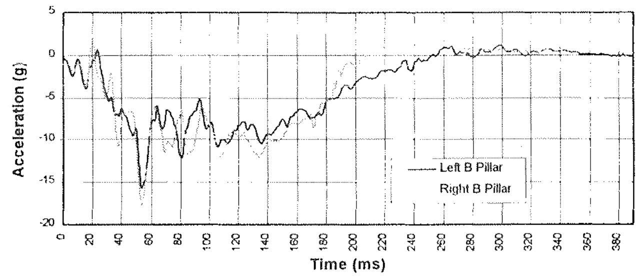

Figure 16. Acceleration pulse for base of B pillar, 48kph offset test with prototype, energy absorbing underrun barrier system (m=1700kg)

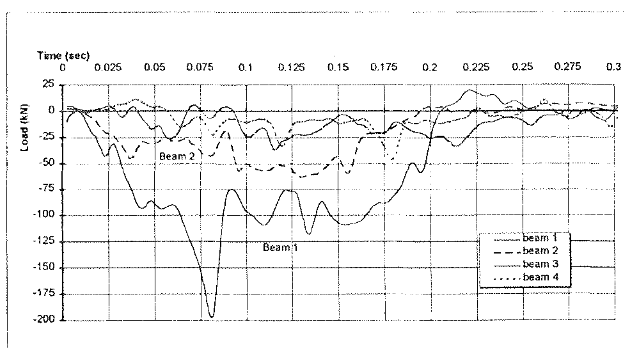

Figure 17. Horizontal load vs time for energy absorbing underrun barrier, 48kph, 50% offset test (m=1700kg). The load is for each of the four strut units, with the forces calculated from strain gauges on the barrier framing. Beam 1 is at the offset side. Figures 16 and 17 are from the same test.

In terms of the performance of the energy absorbing modules, some problems were identified from these crash tests. These included the need to increase the load capacity of the steel tubes, as the peak force measured was 200kN (refer Figure 17) in the offset tests (particularly onerous loading due to the engine striking the corner of the barrier directly) and compression failure and buckling of the unit. In addition the ball joints require longer 'necks' to ensure adequate rotation prior to locking up.

These issues are considered to be easily overcome, with the system demonstrating very good performance in terms of energy absorption and reduction in crash severity. Significantly the measured injury criteria for the Hybrid 3 dummies showed low values and lower than normally measured for this vehicle type, as shown in Table 1, below.

Table 1.
Hybrid 3 results for the centred and offset 48kph crash tests into the energy absorbing rear underrun barriers system compared with test results for the same model to ADR 69 full frontal rigid barrier test

	ADR 69 Test result		Centred		50% Offset	
Injury Criterion	Driver Dummy	Passeng. Dummy	Driver Dummy	Passeng. Dummy	Driver Dummy	Passeng. Dummy
Head Injury (HIC)	848	699	566	271	229	89
Max. femur compressive load (kN)	3.3	2.1	0.6	1.2	1.2	1.1
Chest compression (mm)	36.6	39.1	43	38	32	30

CONCLUSION

An effective energy absorbing unit has been developed consisting of two telescoping light weight steel tubes, containing the energy absorbing *glass fibre reinforced composite* tube. The system operates by crush of the composite tube within the larger diameter steel tube. The full system exhibits very good force-deformation characteristics, with energy absorption in excess of 50kJ. A particular appeal of the use of the fibreglass system is its versatility and ability to readily adapt the system by use of different FRP tubes or graduated short tubes placed in series within the steel tube.

Testing has included static and dynamic loading of a variety of FRP tubes, and finally offset and full frontal impact crash testing of the prototype rear underrun barrier unit to compare the injury outcome with that of rigid barriers. The resultant vehicle crash pulse shows significant lowering of peak accelerations and a nearly doubling of crash duration compared to the equivalent rigid barrier tests, resulting in a beneficial lowering of injury criteria measured on the Hybrid 3 Dummies.

This work in conjunction with the earlier work on fixed rear underrun barriers indicates that barriers should have a load capacity for offset and centred impacts in excess of 250kN, with individual struts units having capacity of at least 200kN peak load. The energy absorption capacity, to be effective should be in the order of 40-60kJ.

FUTURE WORK

A further crash test is scheduled to examine the performance threshold of the barrier. This will be a centred impact test to be conducted at 75 km/h.

The design brief deliberately required a barrier ground clearance of about 450 mm to allow for sufficient departure angles particularly for rigid trucks with long rear overhangs. It is hoped to discuss the trialing of some of these barriers with heavy transport fleets.

FORS expects to do some preliminary investigations on how the barrier design could be adapted to front and side under-run barriers.

ACKNOWLEDGMENTS

The authors gratefully acknowledge the considerable support of the Federal Office of Road Safety (the project sponsor); the staff of Monash University Department of Civil Engineering structural Laboratories - viz, Max Graeme, Roger Doulis, Greg Stroot, Graeme Rundle, Don McCarthy and Peter Dunbar & Dr R. Grzebieta for their considerable work in the construction and testing of the prototype units; Ford Australia and RTA Crashlab for their work in the crash tests; and Jared Rechnitzer for producing the sketches. The idea of using fibreglass composite tubes as an energy absorbing system was suggested by Professor Noel Murray, and is gratefully acknowledged. The views expressed in this report are those of the authors.

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INTEGRATION OF BULL-BARS AS IMPACT ATTENUATION DEVICES WITH AIR BAGS

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ABSTRACT

Bull-bars are often added to many Australian vehicles to minimise damage and maximise the possibility of vehicle driveability after a frontal impact. The increased use of SRS air bags in vehicles has meant that the problem of possible premature air bag deployment during a sub-critical impact speed, must be addressed. Finite element modelling (FEM) was used to evaluate the crash "transparency" of several bull-bars fitted to Australian passenger vehicles. Some calibration of the models using static load tests was also carried out. The FEM was used to analyse the action of impact loads using force-time functions developed on the basis of typical vehicle response histories obtained from crash testing of vehicles. The results from the FEM indicated that the bull bar had very little effect on the crash characteristics of the vehicle. Failure in the bull-bar system typically occurred in the order of mounting plates, attachment bolts and finally the bull bar itself. This meant that very little energy was absorbed before the bar met the front of the vehicle. The bull-bar system could be designed to obtain a predetermined failure mechanism which would control the impact attenuation properties of the bull-bar system.

INTRODUCTION

The use of bull-bars in Australia is widespread because of a perceived need to minimise damage caused by roaming stock, wildlife and roadside flora and to maintain vehicle driveability after a frontal impact. A bull-bar typically is an assemblage of tubular members in the form of a structural frame, mounted to the front of a vehicle, connecting to the frontal chassis members, or equivalent. The bull-bar can absorb energy during a collision, thereby reducing the damage to the vehicle (Figure 1). The bull-bar is usually mounted to the vehicle's chassis by a mounting system consisting of brackets, plates and bolts. The entire system is usually called a "frontal protection system" (FPS). The increased use of SRS air bags in vehicles has meant that the problem of possible premature air bag deployment, during a sub-critical impact speed, must be addressed. Concerns have been expressed by various organisations, that the installation of a FPS to a vehicle fitted with a SRS air bag would modify the stiffness properties of the vehicle and the deployment characteristics of the air bag. These concerns must be addressed as they have great consequences for both manufacturers and users of FPS.

Figure 1. Typical bull-bar used on Australian passenger vehicles.

If a FPS is to fulfil its role then it must be integrated into the overall occupant protection design of the vehicle, including SRS air bags. The FPS must be designed such that it does not prematurely activate an air bag in the case of what would normally be a sub-critical impact.

The FPS also must not add significant stiffness to the vehicle during a high speed impact, increasing danger to the vehicle occupants. Thus a FPS will ideally act as a fitting which will protect the vehicle at low speed impacts (without triggering air bags) and also provide an extra crush zone in high speed impacts.

An extensive investigation was undertaken in the School of Civil Engineering at Queensland University of Technology to identify the impact deformation characteristics of FPS and provide information to manufacturers and users. Several bull-bars, together with their mounting systems, have been evaluated both analytically and experimentally in order to understand their response under impact loads, to determine the load paths, and location and sequence of failure(s) in the FPS.

OBJECTIVES

As indicated above, the main objective of the study was to characterise the load deformation behaviour of FPS under simulated impacts. This would then allow conclusions to be drawn concerning the possibility of premature deployment of SRS air bags. The following points summarise the project's objectives.

- to understand the impact response of bull-bar systems,
- to determine the percentage of energy absorbed by the bull-bar prior to its impact with the car and,
- to recommend means of preventing premature deployment of air bag, if necessary.

SIMULATED IMPACT ANALYSIS

During an impact between a vehicle and a solid barrier the mutually imposed impact force varies with time. If the re-bound velocity of the vehicle is assumed to be zero, the impulse, given by the integral of the time varying force, will be equal to the initial momentum of the vehicle. In this study, a simplified approach was used, where the time varying impact force, was assumed to be triangular in shape with an area equal to the initial momentum of the vehicle. The peak value of the impact force will depend on the duration of impact, and several triangular impact forces were developed and applied to the bull-bar system as force-time functions. This paper presents the details of modelling and analysing a FPS subjected to a single force time function. The FPS used in this study was a T5 type, manufactured by TJM Products Pty Ltd in Queensland.

Modelling

The finite element method (FEM) was used to analyse the bull-bar and mounting system. FEM is a well known and established powerful technique for modelling and analysing structural systems. The system to be analysed is broken up into a number of smaller portions called elements, these elements being connected at points called nodes. Within each element a simple general solution to the displacements, and hence to the stresses, is assumed. These simple solutions are then summed for all the elements to give a general solution for the system. This usually leads to a very large number of simultaneous equations that have to be solved, and thus this method is ideal for use on a modern computer. For the present study a commercially available FEM package was employed.

The bull-bar presented in this study was modelled using beam elements, which were able to simulate bending, axial and torsional forces while the mounting plates were modelled using plate elements, which simulate bending and in-plane plate forces. The FEM model of the bull-bar component created and used in the study is shown in Figure 2. In order to gain a complete understanding of the response of the FPS, finite element analyses were carried out on (a) the bull-bar and (b) the mounting system. The analytical investigation was complemented by static testing of the FPS.

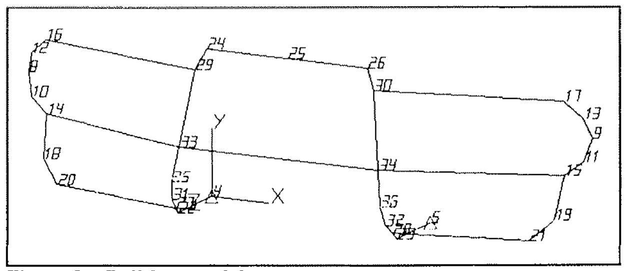

Figure 2. Bull-bar model.

Force-Time Functions

The analysis was carried out by applying time varying loads to the bull-bar at the appropriate points of contact during a full frontal impact. A schematic representation of a typical force-time function used in the study is shown in Figure 3. As indicated earlier, this and other triangular force-time functions, were developed by equating the area of the function to the initial momentum of the vehicle during the crash. For the case reported herein, the initial velocity of the car was set at 25 km/h (6.94 m/s) and the mass of the car was taken as 1614 kg, which is typical of an Australian passenger sedan vehicle.

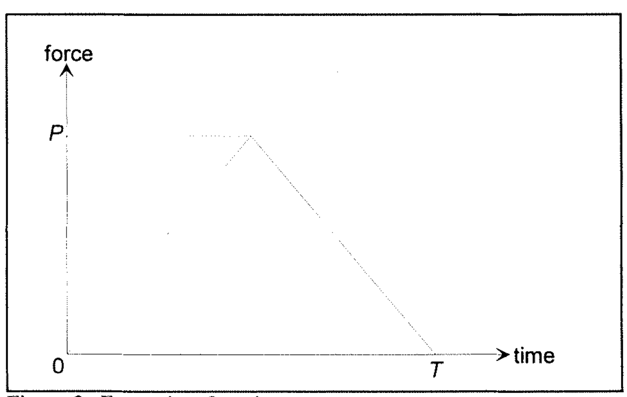

Figure 3. Force-time function.

Assuming the re-bound velocity of the car is zero, the impulse suffered by the car is obtained as the change in momentum, as in equation (1) where M is the mass of the car and V is the velocity just prior to the impact (Beer and Johnston, 1984).

$$I = MV \tag{1.}$$

The time of impact during collisions usually lie approximately between 30 and 40 milliseconds (ms), as obtained from the extensive response records of a previous investigation (Ishac 1993). In the above example, the time of impact has been assumed to be 30 ms.

Considering the principle of impulse and momentum, the area under the force-time function must equal the impulse I. For a triangular force - time function, the area A is given by equation (2) where T is the impact time and P is the maximum impact force. By equating the impulse I with the force-time function area A, the maximum impact force P can be calculated by use of equations (1) and (2). P is given by equation (3).

$$A = \frac{1}{2} TP \tag{2.}$$

$$\begin{aligned} P &= \frac{2MV}{T} \\ &= \frac{2 \times 1614 \times 6.94}{0.03} \\ &= 747 \text{ kN} \end{aligned} \tag{3.}$$

This impulse force was then used in the dynamic FEM analysis, as a force-time function.

Results of Bull-Bar FEM Analysis

The analysis was conducted by assuming the bull-bar component to be rigidly attached to the vehicle. Since the stiffness and the mass of the car is very much larger than those of the bull-bar, this procedure is acceptable until the bull-bar comes into contact with the front of the vehicle. Once the bull-bar contacts the vehicle there is a rapid increase in stiffness and the model no longer applies. The clearance between the front of the car being modelled and the bull-bar was approximately 27 mm. From the displacement time history of the bull-bar, it was evident that the bull-bar impacted with the front of the vehicle after t = 3 ms. The impact force F_o at this time is given by proportion as shown in equation (4).

$$\begin{aligned} F_o &= \frac{2Pt}{T} \\ &= \frac{2 \times 747 \times 0.003}{0.03} \\ &= 149 \text{ kN} \end{aligned} \tag{4.}$$

From the displacement time history of the bull-bar, the energy absorbed by the bull-bar prior to its impact with the vehicle is given by the work done on the bull-bar by the applied force moving through a displacement of 27 mm.

$$\begin{aligned} \text{workdone} &= \frac{1}{2} d_o F_o \\ &= \frac{1}{2} \times 0.027 \times 149 \\ &= 2.02 \text{ kNm} \end{aligned} \tag{5.}$$

The total kinetic energy of the car before impact E is given by

$$\begin{aligned} E &= \frac{1}{2} MV^2 \\ &= \frac{1}{2} \times 1614 \times 6.94^2 \\ &= 38.87 \text{ kNm} \end{aligned} \tag{6.}$$

The percentage of energy absorbed by the bull-bar is therefore about 5.2 %.

Examination of the distribution of stresses throughout the bull-bar revealed that the yield strength of the material had been reached, at the lowest points in the two vertical members, just prior to the bull-bar impacting with the vehicle.

A second analysis was then conducted to examine what effect yielding of the material within the bull-bar had upon its overall response. This was achieved by performing a non-linear static analysis, where the prior and post-yield properties of the material were entered into the model. The loading was then increased in gradual increments until the final loading was reached. An examination of the deflection verses force graph (Figure 4) from the non linear analysis reveals the changes in overall stiffness.

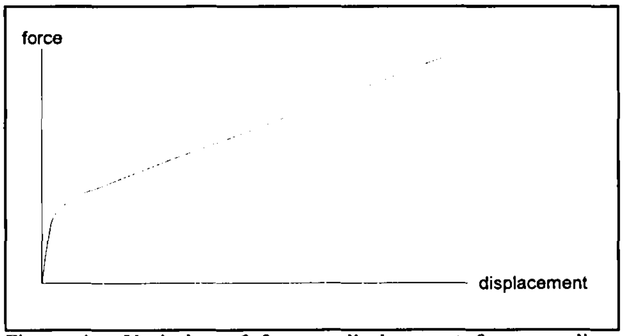

Figure 4. Variation of force - displacement from non-linear analysis.

The non-linear analysis revealed that after yielding of the material within the bull-bar component (of the FPS) the stiffness dropped to around 5 % of the original stiffness. From this it can be concluded that after initial yield the bull-bar would have a minimal effect on further crushing resistance of the vehicle.

A static test of the bull-bar conducted in the laboratory confirmed the predictions of the non-linear analysis.

Results of Mounting System FEM Analysis

In the second part of this study, an FEM analysis similar to the one conducted on the bull-bar was applied specifically to the mounting system. The analysis showed that for this particular FPS, initial failure would occur in the mounting system before in the bull-bar. Calculations similar to those used in the previous section showed that the total percentage of energy absorbed was approximately 2 %, which is significantly lower than the 5 % calculated for the case when the bull-bar component failed. This prediction of the mounting system failing first was confirmed by a static test of the full FPS conducted in the laboratory.

DISCUSSION

The above discussion highlights an important and interesting finding which can be used to evaluate the performance of a FPS. A simple static test on the FPS can be used to obtain a first approximation of performance. If the failure occurs within the mounts and if the energy absorbed is of order of 2 - 3 %, then that FPS can be considered to be satisfactory. If the sequence of failure is different, then to comply with crash requirement such as demanded by SRS air bags the mounting system (plates and bolts) can be redesigned for failure to occur first and at a predetermined energy level.

Further, very sophisticated FEM modelling (Vehicle Safety Research Group, 1994), verified the simplistic approach used in the study reported here. That FEM work confirmed the load path, failure pattern and level of energy absorbed by the FPS in this study.

CONCLUSIONS

During the initial (elastic) stages of the impact, the FPS studied did not add significant stiffness to the crush stiffness of the vehicle. The most likely effect of existence of the FPS would be to modify impact pulses, and deceleration profiles, marginally in the early stages of impact. There is no evidence that the elastic stiffness of the aluminium bull bar should significantly change the overall shape of the deceleration profile during these early stages. The elastic deflection and any minor plastic deformation of the bull bar prior to vehicle impact will provide an initial, low energy absorbent (about 5%) mechanism.

The interaction of the bull bar and frontal area is a complex mechanism. Analysis of crush data from NCAP is based on an elastic/plastic assumption which uses an initial elastic energy requirement prior to commencement of crushing. Crushing is then approximated by plastic deformation, which is responsible for absorbing the great proportion of impact energy. The deceleration curve (of the occupant) for the impact typically lags behind the force/crush diagram and reaches the first significant level after about 20 to 30 ms into the impact. It is important to define when the FPS stops contributing stiffness to the front of the vehicle, in order to predict the effect of the FPS on deceleration. The FPS studied in this report yielded within the bull bar and at the mounting bolts, in the first few milliseconds prior to impact with the front of the vehicle. It appears that little stiffness would then be contributed to the crush zone after the bull bar impacted the front of the vehicle.

A beneficial aspect of this FPS is that it possibly provides an extended, low stiffness, impact zone which will increase the deceleration time. The bar may also assist with reducing intrusion into occupant space. It should also be noted that other variables such as impact angle, vehicle load, and mass and stiffness of the body impacted could change the deceleration profile of the vehicle occupants and SRS air bag sensor. Perhaps more so than the T-5 FPS studied in this report.

The study carried out demonstrated that the bull bar and its mounting system must be considered as a single integrated system. or FPS. It was shown that the FPS investigated in the study acted as a soft extension of the vehicle which did not affect the deployment characteristics of the vehicle being modelled.

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BULLBAR DESIGN FOR AIRBAG EQUIPPED VEHICLES.

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Paper Number 96-S4-0-12

ABSTRACT

This paper addresses the design philosophies required to design a functional and robust bullbar for an airbag equipped vehicle. Areas to be addressed in this paper include; potential issues with after market bullbars, design of the bullbar to vehicle substructure interface, testing and validating original equipment (OE) bullbars and real world functional performance.

INTRODUCTION

The issue of original equipment bullbar fitment to air bag equipped passenger vehicles is a relatively new design requirement for vehicle manufacturers around the world. Traditionally vehicle manufacturers have ignored the subject of bullbars and left the design, manufacture and fitment of bullbars to the after market specialists. With the advent of the airbag becoming standard fitment to most vehicles, vehicle manufacturers have had to re address whether allowing the after market to design and fit bullbars systems, in isolation from the design of the vehicle airbag triggering system, is the most robust design approach for their vehicles. This is particularly true for markets such as Australia, where the traditional bullbar fitment rate in some country areas is estimated to be as high as 60%. Vehicle manufacturers cannot afford to ignore the potential serious implications of having an after market bullbar change the crash characteristics of their vehicle, which potentially could effect the performance of their active restraint systems.

Vehicle manufacturers around the world are now addressing the subject of integrating original equipment bullbars into the design of their next generation vehicles.

BULLBAR DESIGN ELEMENTS

For a vehicle manufacturer to incorporate a bullbar as part of its product line up, it must address many design elements, many of which may or may not be have been addressed by the after market manufacturer.

The possible deleterious effects a bullbar may have on a variety of vehicle sub systems need to be quantified. For example, does the bullbar cause any problems with sub systems such as air bag triggering calibrations, crash pulse signature, vehicle lighting, engine cooling, suspension elements, vehicle substructure fatigue life, vehicle serviceability, wind noise, damage to vehicle systems during low speed collisions and vehicle aesthetics to name a few.

Functional bullbar performance issues need to be addressed. Robust mechanical design parameters such as beam strength, load reaction paths, minimised centre of gravity from structure connection points and fatigue life of attachment and associated vehicle structure, are all important elements in the design of any bullbar to ensure it has design robustness.

Pedestrian friendliness is a very important element of any bullbar design. Issues such as elimination of all sharp and protruding edges, minimisation of the potential to hook or graze pedestrians needs to be addressed. Another important element in a pedestrian friendly design bullbar is the "plan view" shape of the bar. The bullbar needs to incorporate "plan view" curvature, to assist with imposing "lateral shedding force components" onto any contacted element. Pedestrian kinematics need to be reviewed to ensure that as a minimum, the kinematics do not deteriorate with the fitment of the bar assembly compared to a standard vehicle front end.

Demonstrated compliance with occupant protection standards must be addressed to ensure that a bullbar system does not deteriorate or impede the vehicle's passive and active restraint systems. Computer simulations are not a robust form of validating a design, due to the complex dynamic interactions that can occur with the vehicle's load bearing members during dynamic collisions. It is quite possible that an incorrectly designed bullbar could not only upset the calibration of a vehicle's airbag sensor system, but also provide a "structural lock up" between load bearing members of the vehicle structure and hence change the crash pulse and resultant passenger and driver "ride down" kinematics. The implication on injury criteria such as HIC and chest deceleration can only be robustly validated by way of a full vehicle crash test.

All of the above elements were addressed in the design of the "Smart bar" for the Australian Ford Falcon. The remainder of this paper will concentrate on the issues of interfacing the bullbar system with the vehicle sub structure, pedestrian impact modeling and functional testing.

DESIGN OF A BULLBAR TO VEHICLE STRUCTURE INTERFACE.

The crash detection system employed for the EF Ford Falcon is an electromechanical system similar to that used on most current US airbag systems. The system incorporates 2 crash sensor points, one on the radiator upper cross member and one on the vehicle's dash panel. A safing sensor is also included, which is located in the airbag diagnostic module. To initiate an airbag fire, one of the vehicle's two crash sensors must close together with closure of the vehicles safing sensor.

Detail of the crash sensor assembly is shown in Figure 1. The crash sensor contains a metal ball, guide, magnet and two contacts. The ball is held away from the switch contacts by a magnetic bias. During an impact, the ball break's away from the magnetic bias, travels down the tube and closes the contacts at the other end. The travel of the ball in the guide is air damped by designing a very small gap between the ball and the guide.

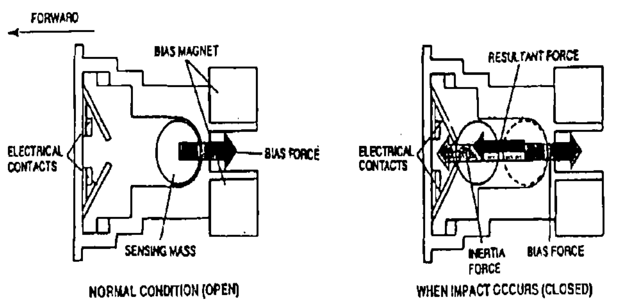

Figure 1. Falcon crash sensor operation schematic.

The level of deceleration required to overcome the magnetic bias on the ball is a calibration parameter for the sensor. The sensor calibration is essentially determined by quantifying the vehicle's "crash signature" at two given impact velocities, a low velocity (12~16km/hr) no fire threshold and a high velocity (+25 km/hr) must fire, threshold.

The low velocity, no fire calibration, of the crash sensors can be severely altered with the fitment of a non approved aftermarket bullbar system. The crash signature of the vehicle at low speed, is determined by the "dynamic stiffness" of the front end elements. The elements of the front end system that can effect the "dynamic stiffness" include, bumper skin, bumper beam, bumper mounting brackets and potentially the first 100mm of the vehicle's side rail structure.

It is the low speed (less than 13 km/hr) type impacts that are of major concern with fitment of after market bullbars to an airbag equipped vehicle. After market bullbars, typically directly couple, the vehicle's substructure to the leading edge mechanical elements of the bullbar, this has the effect of "short circuiting" the compliant elements of the standard bumper system. Short-circuiting the compliant elements, results in a stiffer front end and hence a change in the vehicle's de-acceleration signature. Since the airbag sensors for any vehicle are calibrated for the intended vehicle's crash signature, changing the front end structure characteristics, by short circuiting the compliant bumper system with an after market bullbar, will result in a different crash signature and hence unpredicted performance of the airbag triggering system. Such unpredictable performance during low speed impacts has a high probability of causing unintended airbag misfires

During development of the "smart bar" for the Australian Ford Falcon, considerable development work was undertaken to quantify the performance objectives for a bullbar system at low speeds. A series of low speed (13 km/hr) 90 degree barrier tests were undertaken on a standard Falcon bumper system, a simulated after market directly coupled bullbar system and the dynamic compliant Ford "smartbar". Figures 2, 3 &4 summarise the results obtained during the development program.

Figure 2 is a plot of the longitudinal velocity of the Falcon front upper cross member crash sensor, relative to time, for a 13 km/hr 90 degree barrier impact. It can be seen from Figure 2, that the crash signature for the simulated directly coupled after market bullbar system, differs considerably from the crash signature of a standard Falcon front bumper system. The "short-circuiting" effect of the after market bullbar is quite obvious, when the time taken to cross the zero velocity axis is compared. For a standard Falcon it takes approximately 60+ms to cross the zero velocity axis, whereas for the simulated after market bullbar system it takes approximately 40ms.

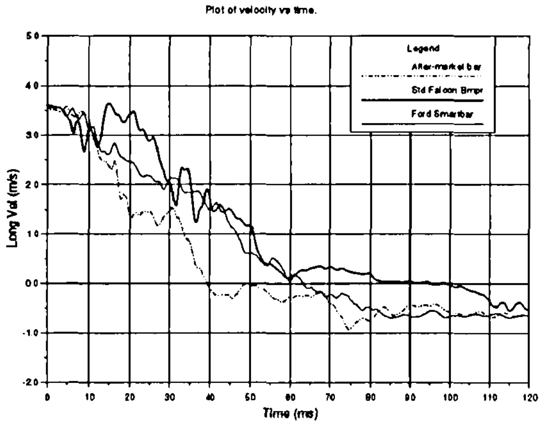

Figure 2. Falcon front crash sensor longitudinal velocity relative to time. (13 km/hr 90 degree barrier)

The longitudinal displacements plots for the front crash sensor Figure 3, confirm that the stiffness of the front end with the after market bullbar system is considerably greater than the standard front bumper system. The standard front bumper system has a peak displacement of 137mm, whereas the simulated after market bullbar system has a peak displacement of 80mm.

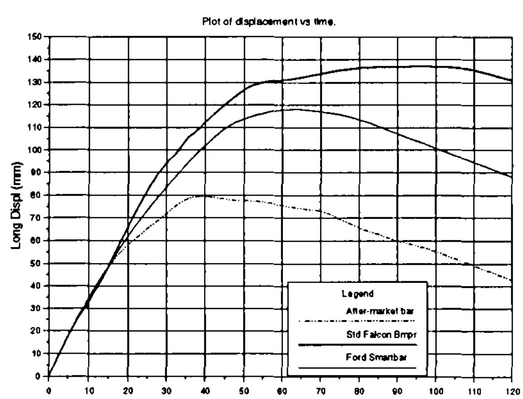

Figure 3. Falcon front sensor crash sensor longitudinal displacement relative to time. (13 km/hr 90 degree barrier).

Examination of the front sensor for closure for the standard front bumper system versus the after market bullbar system can be seen in Figure 4. Clearly for the simulated after market bullbar system, there would have been an airbag deployment at the low velocity no fire threshold.

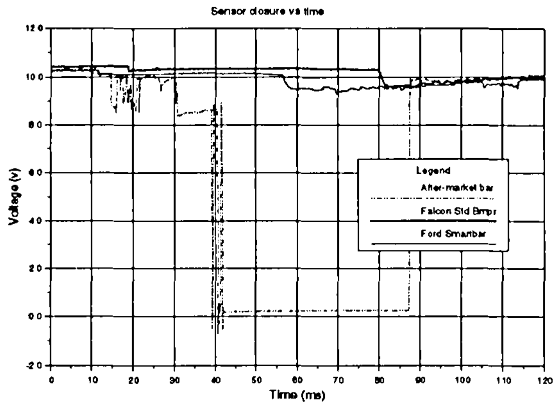

Figure 4. Falcon front crash sensor closure versus time. (13 km/hr 90 degree barrier)

In conclusion our development tests confirmed that the fitment of an after market bullbar had the potential to seriously modify the crash signature of a standard bumper system during low speed impacts and cause inadvertent airbag fires.

It had therefore become clear, that any OE (original equipment) bullbar needed to have a dynamic compliance characteristic that closely resembled the standard front bumper system if we were to avoid the inadvertent air bag fire problems experienced with the traditional "short-circuit" after market bullbar systems. The design team then set about to design an interface system between the mechanical elements of the bullbar system and the Falcon's siderails. The approach adopted was to design and package a Poly Gel Mitigator (PGM) system between the bullbar and siderails. Figures 2, 3 and 4 clearly show how successful a PGM interface system can be if carefully designed and developed. Figures 2 and 3 illustrates that the "smartbar" has a crash signature that closely simulates the Falcon's front bumper system, with a time to cross the zero velocity axis of approximately 60ms and a maximum longitudinal displacement of 118mm. Both of these parameters compare favorably to the standard bumper system unlike the simulated "short circuited" after market system. Sensor closure monitoring for the "smartbar" system as shown in Figure 4 clearly shows its design robustness in that there were no airbag firing closures with the PGM interface.

A sectional view of the Falcon "smartbar" is shown in Figure 5. As can be seen from Figure 5 the design consists of four principle elements, the front bumper beam, the PGM, the PGM interface bracketry and the vehicle's siderail structure. It is fair to say that it would be near on impossible to package a PGM/bullbar system into the design of front bumper system, if it were not tackled from the early design feasibility stages. Packaging a bullbar system that complies with all current and future projected design rules, that has no sharp or protruding edges, that integrates aesthetically, that travels upto 40mm during low speed impacts, that has a round beam section and has "plan view " curvature can present many varied and quite often new design problems. Such problems could not be tackled after the vehicle had been designed.

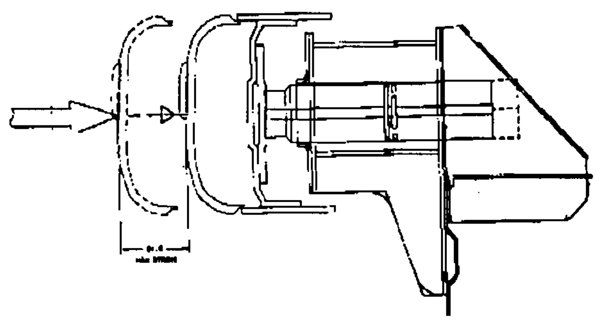

Figure 5. Sectional view of Falcon "Smartbar".

MADYMO PEDESTRIAN IMPACT SIMULATION.

A major element of concern for an OE bullbar manufacturer is the potential effect the bullbar geometry may have on pedestrian injury criteria. Fitment of bullbars has the potential to increase lower leg injury levels due to the solid metal beam at bumper height instead of the plastic deformable beam structures of most standard front ends. The area that an OE manufacturer can influence, is the geometry design of the upper bullbar structure. A bullbar manufacturer needs to demonstrate due diligence in the design of the upper bullbar structure. Such due diligence can be demonstrated by undertaking a MADYMO pedestrian impact simulation of the standard vehicle front end versus one with the bullbar attached.

As part of the Falcon "smartbar" development, a series of due diligence MADYMO simulations were undertaken. A MADYMO model was created which included the following elements;
-a pedestrian model based upon the Part 572 dummy
-a vehicle model built to match the front end geometry of the Falcon. The contact surfaces modelld included the grill, spoiler, bumper, hood, front wheels, roof and windshield.
-an 18 ellipsoid bullbar model.

A Taguchi parameter trade off analysis was undertaken on 30 model runs to determine the influence that bullbar geometry, braking and speed had on a variety of parameters. Table 1, is a Taguchi mean value parameter trade off summary, for two selected parameters, namely HIC and Chest g's. The values have been normalised to 100% due to the fact that insufficient modeling details had been used in the model to make any absolute numbers meaningful.

From Table 1, the following conclusions can be drawn;
-in relation to the geometry parameter (which is effectively the difference between a standard front end versus one with the Falcon "smartbar" fitted) the HIC and chest values are very similar. Such similarities in numbers confirm that the design requirement to have similar pedestrian kinematics for the "smartbar" versus the standard front end had been achieved.
-the impact velocity had the expected outcome of, the higher the velocity the higher the injuries.
-it appeared that car pitching during braking had a deleterious effect on injury numbers.

Table 1. Taguchi parameter trade off table of mean values

TAGUCHI ANALYSIS
(Relative Deviation Compared With 100%)

Parameter Setting		HIC	Chest 3ms
Geometry	Standard front end	100	100
	Falcon 'smart bar"	90	79
Speed (km/h)	5	0	5
	10	1	36
	20	10	49
	30	47	66
	40	100	100
Braking	No braking	79	79
	Braking and pitch	100	100

FUNCTIONAL TESTING PROGRAM.

A robust development and test program to validate a design must be undertaken before any product is released onto the market. For the Falcon "smartbar", the following tests were undertaken during the extensive test and development stage of its design;

-traditional rough road durability program
-48 km/hr ADR69 frontal barrier test
-air bag due diligence, deployment tests
-PGM impact tests, low, medium and high speeds
-Pendulum impact tests
-Insurance damageability tests
-Finite element analysis
-Lighting compliance tests
-Water fording
-Day/night driver evaluations
-Wind noise evaluations
-Accelerated stone pecking
-Corrosion resistance testing
-Car wash cycle
-Engine cooling tests
-Assembly feasibility tests
-Functional performance tests

The major reasons why customers fit bullbars to their vehicles is to provide a degree of protection to their vehicle and themselves, in the advent of a collision with a stray road animal. It is fair to say that the customer is relatively uninterested with the subtle design problems faced by vehicle designers in interfacing a bullbar to an airbag equipped vehicle. The customer just wants to be confident that the bullbar fitted to his/her vehicle will perform as expected, during a collision with stray road animals. The following paragraphs describe in detail the processes used to functionally validate the Falcon "smartbar" from a customer's perspective.

From researching the technical literature on wildlife impacts, it became clear that very little work had been undertaken on development of wildlife impact tests. After consultation with our fleet customers as to what where the major animals of concern on the Australian roads, we set about to develop a test program based around the 75 Kg Australian male red kangaroo. After consultation with the Melbourne zoo on species information, we then set about to develop a "kangaroo dummy" that could be used for our impact testing. Figure 6 defines the approximate characteristics for our "kangaroo dummy". The main elements we endeavored to incorporate in our design were the correct centre of gravity, approximate girth dimensions at the hips and a representative head and chest mass distribution.

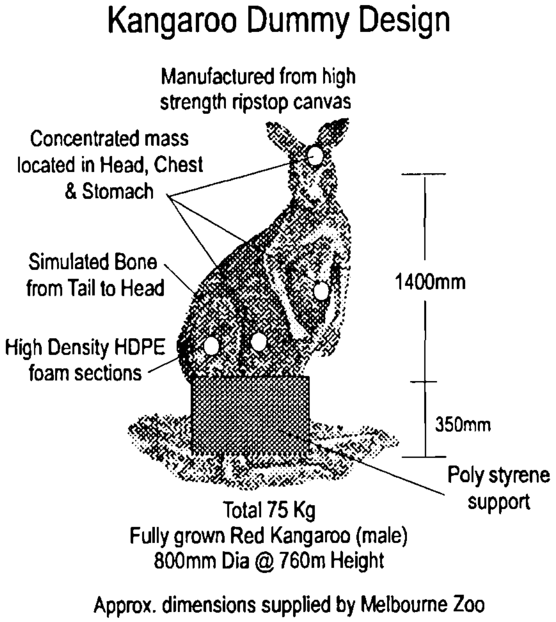

Figure 6. Australian male red kangaroo impact dummy specifications

The impact test procedure consisted of impacting the kangaroo at 100km/hr through the centre line of the vehicle. This procedure was conducted for a standard vehicle and one fitted with a Falcon "smartbar". Figures 7 and 8 show the post kangaroo impact damage for a standard Falcon front end and a Falcon fitted with a "smartbar" respectively. The "smart bar" performed as a customer would expect, with the vehicle being completely driveable, the only damage being minor hood and bumper deformations, with no structural damage. The standard vehicle on the other hand, was undriveable with major structural damage to the upper crossmembers, siderails, lamps, cooling system and bumper components.

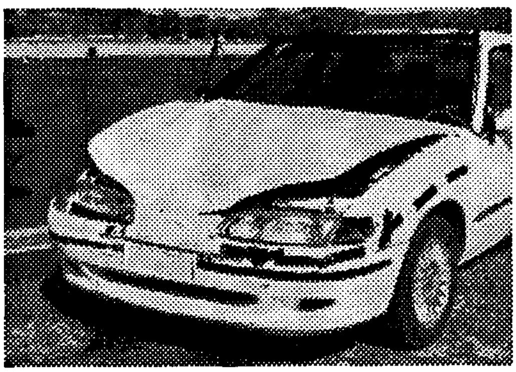

Figure 7. Post 75Kg 100km/hr kangaroo impact damage for standard front end.

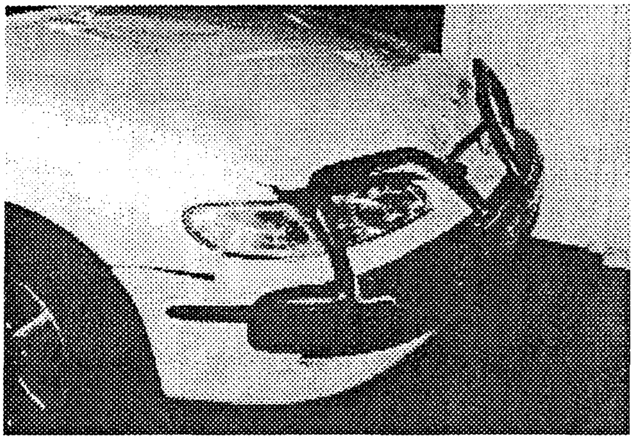

Figure 8. Post 75Kg 100km/hr kangaroo impact damage for "smartbar" installation.

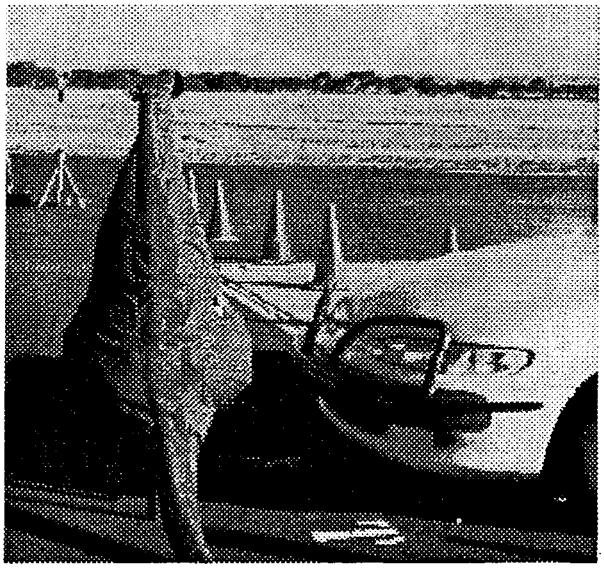

Figure 8 Kangaroo dummy/impact vehicle relationship.

CONCLUSION

This paper has endeavored to illustrate the macro design elements necessary to interface a bullbar system with airbag equipped vehicles. To avoid airbag misfires during low speed impacts, the bullbar to vehicle structure interface, must simulate the dynamic stiffness characteristics of the standard front end to ensure that the crash sensors are not "tricked" into firing the airbags.

It is possible, to design a bullbar system for airbag equipped vehicles, however it is the authors belief that it is not possible to demonstrate compliance or "design robustness" without an extensive test validation phase, using real vehicles and real hardware. To pursue a route of validation based upon computer simulation without hardware testing and extensive reference to the manufacturer's knowledge of their complex, interactive systems, would in the author's opinion be less than diligent. This paper has attempted to quantify the necessary robust testing that is required to demonstrate "design robustness" in bullbar design.

ACKNOWLEDGMENTS

The author would like to take this opportunity to express his appreciation to his team within Ford Australia test operations, who without their help, this paper could not have been generated. Special thanks go to Kent Fuller who conducted all the baseline testing, David Stonehouse who generated all the computer images and Ming Loo who conducted the fundamental data analysis.

SAFETY CONCEPTS FOR VERY SMALL VEHICLES, EXAMPLE : OPEL MAXX

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ABSTRACT

The safety performance of very small cars is a big challenge in modern vehicle design. Basic physics requires entirely new concepts for such small vehicles in order to meet the safety standards of normal-sized cars and to achieve crash compatibility. This paper outlines the safety concept for very small cars using the example of the Opel MAXX. The structural design and the restraint system are discussed, together with computer simulations of the crash behaviour.

INTRODUCTION

The increasing number of vehicles in use worldwide suggests the need for new design concepts for cars. This is particularly obvious in downtown city areas where space is limited. In addition, the increasing awareness of ecological considerations means that fuel economy is of ever greater importance. Such requirements automatically lead to small, light-weight cars. At the same time, customer expectations of comfort and safety are also increasing. Therefore, as usual, the design of the vehicle must be a compromise between different and contrasting development goals and boundary conditions, which may be even more restrictive for very small vehicles.

The physical limits of passive safety for lightweight cars has already been studied by Richter & co (ref. /1/). It is chiefly the mass ratio which is responsible for the higher injury risk for the occupants of such vehicles. In addition, the short front end of smaller vehicles results in a reduced deformation zone. Therefore the energy absorption, which should be as high as possible, depends on a more sophisticated structural design. Finally, the restraint system should be optimized to reduce the high deccelerations, which are directly related to the injury risk for the occupants.

The present paper discusses the safety aspects of very small vehicles. Here, the Opel MAXX concept car has been taken as the case study for the evaluation of the new technologies. The design is based on modular components constructed almost entirely from extruded aluminium profiles. This manufacturing technique is also used for the main body structure. The front and rear ends of the structure are unusually short. This ensures a wide passenger compartment and a minimum overall length for the vehicle. For the two-door version, the total length is of approximately three meters. The engine is mounted transversely in the front end. In fig. 1, the sizes of the Opel CORSA and the Opel MAXX are compared.

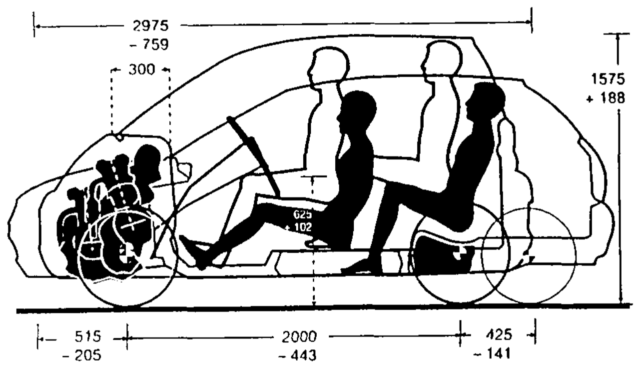

Figure 1: Geometrical comparison between Opel CORSA and Opel MAXX (values in mm)

The paper contains five chapters. After this introduction, safety concepts for small cars are discussed. This is followed by guidelines for the structural design of such cars, using the Opel MAXX as an example. The next chapter contains some remarks about an optimized restraint system. The paper is concluded by a short summary.

SAFETY CONCEPTS FOR SMALL CARS

There are two main reasons why occupants of small cars have a higher injury risk then those of big cars. The first is physical and concerns the mass ratio of a small car compared to larger cars. The physical conditions may be reviewed by considering the laws of impulse and conservation of energy. Two cars are represented by the masses m_1 and m_2, respectively.

m_i	-	masses of the cars
v_{ib}	-	velocities before impact
v_{ia}	-	velocities after impact
v_s	-	velocity of the center of gravity
$(i = 1, 2)$		

For simplicity the one dimensional case is considered, the friction is neglected and the direction of motion is along a line joining the centers of gravity of the two cars. The velocity v_s of the whole system remains the same before and after the impact. A crash against a rigid wall is assumed by setting $m_1 = m_2$.

$$v_i = v_s + v_i^s \quad ; \quad (i = 1, 2)$$

$$v_1^s = \frac{m_2(v_1 - v_2)}{m_1 + m_2} \quad , \quad v_2^s = \frac{m_1(v_2 - v_1)}{m_1 + m_2}$$

$$v_s = \frac{m_1 v_1 + m_2 v_2}{m_1 + m_2} \quad \Rightarrow \quad \frac{m_1 v_1^s + m_2 v_2^s}{m_1 + m_2} \equiv 0$$

$$\Rightarrow \quad \frac{m_2}{m_1} = -\frac{v_1^s}{v_2^s}$$

In the reference system, the mass ratio is inversely proportional to the ratio of the velocities. This is the main reason why the vehicle with the smaller mass is subjected to a higher decceleration level than the car with the bigger mass, which may led to unacceptable high levels of accelerations for the passengers in the smaller car. In general a vehicle with a small mass is subjected to a higher acceleration when colliding with a bigger vehicle.

The second reason is geometrical. The requirement to build such cars as small as possible means that the crush zones (i.e. front , rear and side) are reduced to a minimum. An optimal design of these areas with respect to the crash behaviour should result in a high absorption of the kinetic energy into plastic deformation. A plasticity index η_p may be defined by the ratio of the velocities before and after the crash. The energy absorption may be measured by an energy absorption index η_w, which is related to η_p. The purely plastic case is indicated by a value of 1 for both, a value of 0 represents the purely elastic case.

η_p	-	plasticity index
η_w	-	energy absorption index

$$\eta_p = 1 - \left|\frac{v_{1a}^s}{v_{1b}^s}\right| = 1 - \left|\frac{v_{2a}^s}{v_{2b}^s}\right| = 1 - \left|\frac{v_{1a}^s + v_{2a}^s}{v_{1b}^s + v_{2b}^s}\right|$$

Using η_w, the amount of absorbed (plastic) energy ΔW may be calculated.

$$\eta_w = \eta_p\,(2 - \eta_p)$$

$$\Rightarrow \quad \Delta W = \eta_w \left(\frac{m_1}{2} v_{1b}^{s2} + \frac{m_2}{2} v_{2b}^{s2}\right)$$

$$= \eta_w \frac{m_1}{2} v_{1b}^{s2} \left(1 + \frac{m_1}{m_2}\right) = \eta_w \frac{m_2}{2} v_{2b}^{s2} \left(1 + \frac{m_2}{m_1}\right)$$

It may be stated that, even with a short crushzone such as for a small vehicle, the energy absorption should be as high as possible. This results in the lowest possible decceleration values due to the mass ratio. The remaining occupant loads have to be minimized by a proper design of the restraint system.

The design goals for a very small vehicle may therefore be defined as follow:

- passenger compartment optimized with respect to minimal deformation
- crushzone optimized with respect to high energy absorption
- restraint system optimized with respect to low occupant loads

Of course several test conditions should be considered to ensure the satisfactory behaviour of the concept. This is mainly important within the optimization process. In the present study, the investigation of these tests is performed by computer simulations based on the Opel MAXX. Fig. 2 shows the undeformed finite element model.

STRUCTURAL DESIGN AND CRASH BEHAVIOUR

The Opel MAXX is a front wheel drive car, as is usual for small vehicles. For the engine in this study a three cylinder version is considered. The gearbox and differential transmission are mounted compactly one above the other.

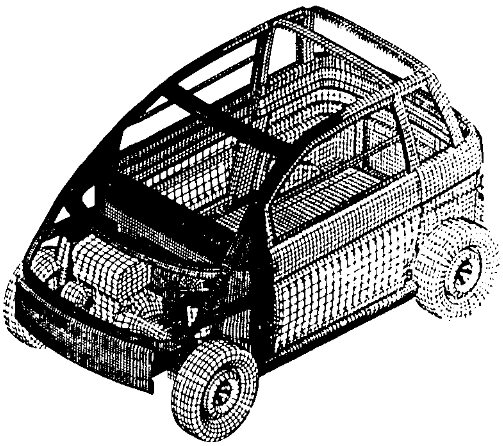

Figure 2: Finite element model of the Opel MAXX

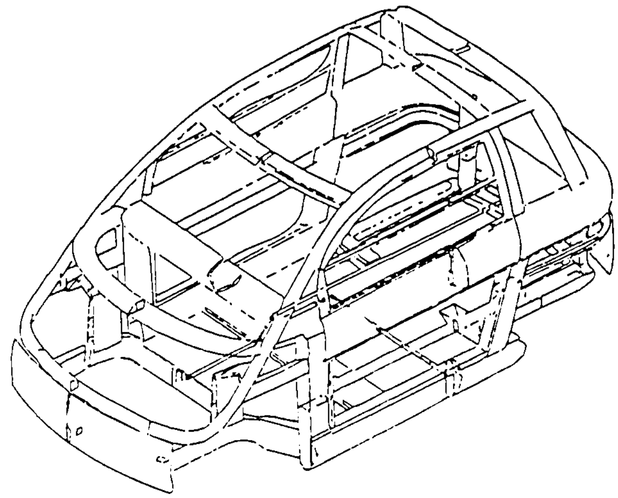

Figure 3: Body structure of the Opel MAXX

The main body of the passenger compartment is made from extruded aluminium profiles. Using such a manufacturing technique, components of different cross section may be built easily. Profiles can be provided open or closed to optimize the crash behaviour by minimizing the weight and maximizing the stiffness of the structure. The different parts of the body are connected by the MIG welding technique. No casting is used within the short crush zones because of the rigid behaviour of such components, which is contrary to an optimal crush performance. The front end contains hydro-formed aluminium components and also bending-deformed extruded aluminium profiles. The concept ensures a high absorption of the kinetic energy in case of a crash even if the crush zone of the front end is only 400 mm long, as with the Opel MAXX.

Defined load paths conduct the forces into the stiff cell structure. The longitudinal frames and the brace wheel house are made from extruded aluminium profiles and contain an improved buckling behaviour. The required lower load path is created by welding the frame structure to the rocker. To get an upper load path the brace wheel house is attached to the lower A-pillar. From these points, the forces are conducted to the circumferencial aluminium profile, which can be seen from the outer side of the body. Using closed extruded aluminium profiles, a high stiffness for the frame body and a low mass of the passenger compartment can be achieved.

The bottom structure contains a two-wall profile, which consists of two horizontal plates connected by vertical stripes. The dash panel is made from a sandwich aluminium structure. The upper part is reinforced by a strong C-beam which is also the steering cross-member. The bumper cross-member is high and ensures that in an offset crash the unloaded frame structure absorbs part of the plastic energy.

In the following, the frontal crash behaviour is discussed, because it is the most critical. To investigate the crash performance of the concept, four loading cases are considered. A computer simulation was performed for each.

- 100 % overlap, 50 kph against a rigid barrier
- 50 % offset, 55 kph against a 15° rigid barrier (AMS configuration, see ref. /6,7/)
- 50 kph against the offset deformable barrier, 40 % overlap
- 50 kph offset car-to-car (with Opel OMEGA, compatibility study), 50 % overlap

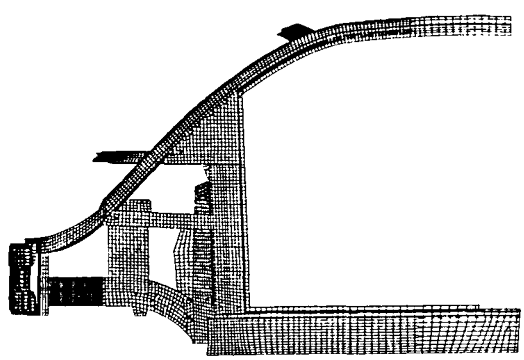

Figure 4: Undeformed frame structure of the Opel MAXX

In fig. 5, the deformed structure of the vehicle is shown for the simulated rigid barrier test (100 % overlap, 50 kph). The passenger compartment remains undeformed. The deformation of the front end structure is shown in fig. 6. Compared with the undeformed shape of fig. 4, it is obvious that only the main frame structure in the front deforms. From fig. 7, the deformation of the dash panel can be seen. It is low for this loadcase. The crush length for this test is about 385 mm.

In fig. 8, the deformed structure of the vehicle is shown for the simulated AMS offset rigid barrier test.

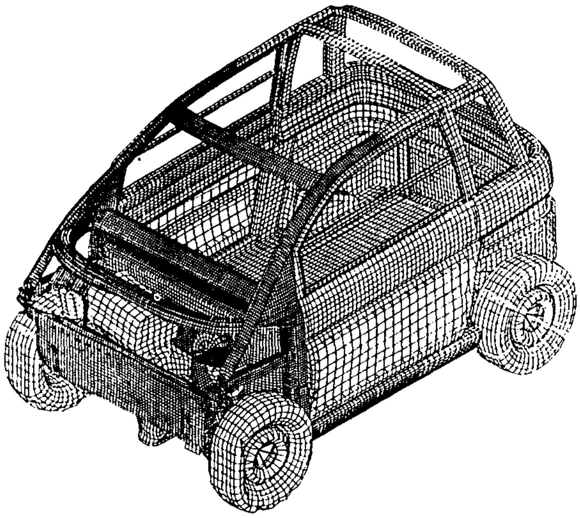

Figure 5: Deformation of the Opel MAXX v. rigid barrier

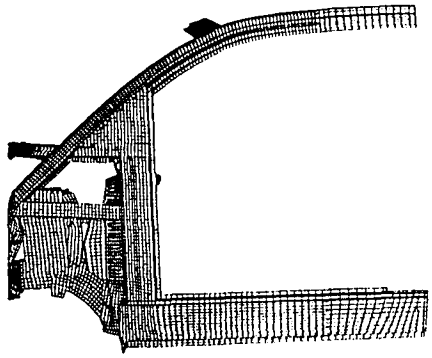

Figure 6: Deformation of the frame structure, Opel MAXX v. rigid barrier

Only small deformations of the passenger compartment occur. The deformation of the front end structure is shown in fig. 9. Although there is a longer deformation of the front end than in the 100 % barrier loadcase, the overall behaviour of the structure is very acceptable. Fig. 10 shows the low deformation of the dash panel.

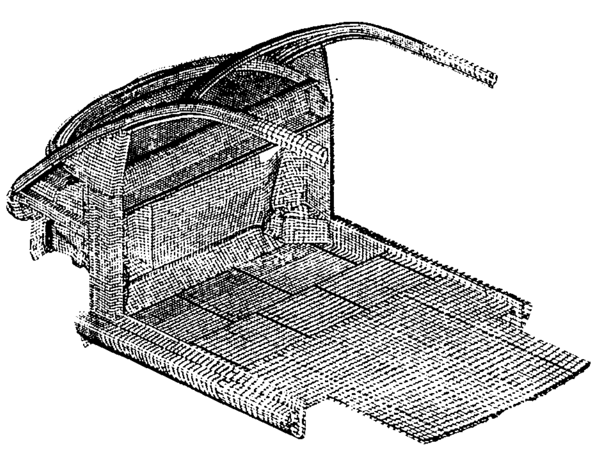

Figure 7: Deformation of the dash panel, Opel MAXX v. rigid barrier

The crush length of the front end is about 470 mm.

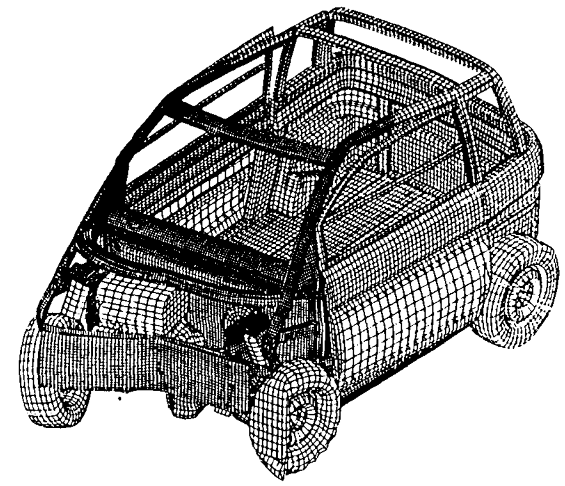

Figure 8: Deformation of the Opel MAXX, AMS Configuration

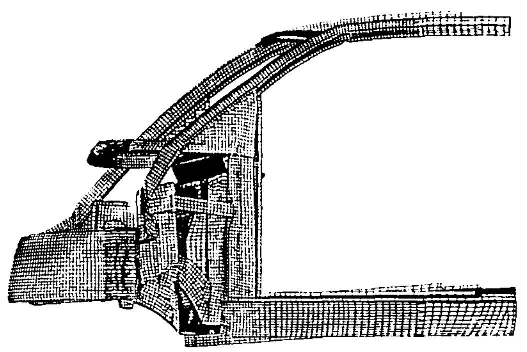

Figure 9: Deformation of the frame structure, Opel MAXX, AMS Configuration

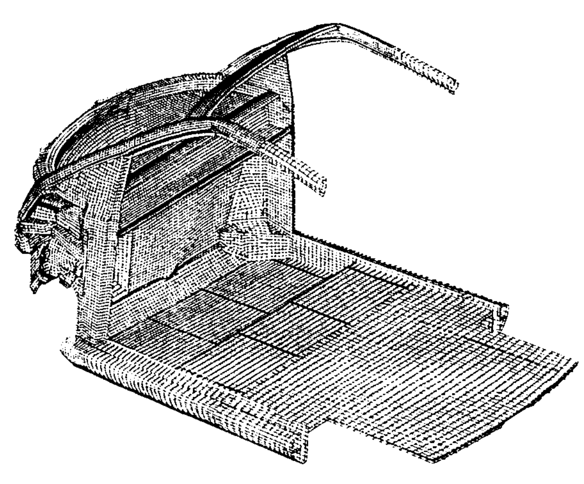

Figure 10: Deformation of the dash panel, Opel MAXX, AMS Configuration

In fig. 11, the deformed structure of the vehicle is shown for the simulated offset deformable barrier test.

In this configuration, the lower load path is important. As may be seen from fig. 12, the structure behaves well, even a little better than for the AMS configuration. The dash panel deformation for this loadcase, shown in fig. 13, is low as is required. The total deformation for this configuration is about 640 mm (barrier plus vehicle).

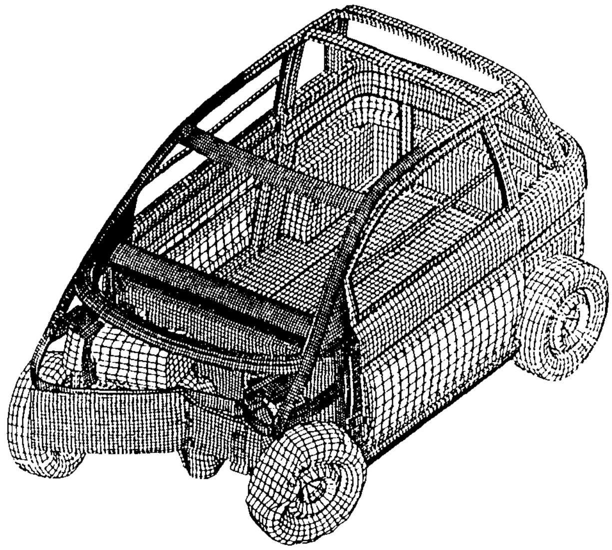

Figure 11: Deformation of the Opel MAXX v. offset deformable barrier

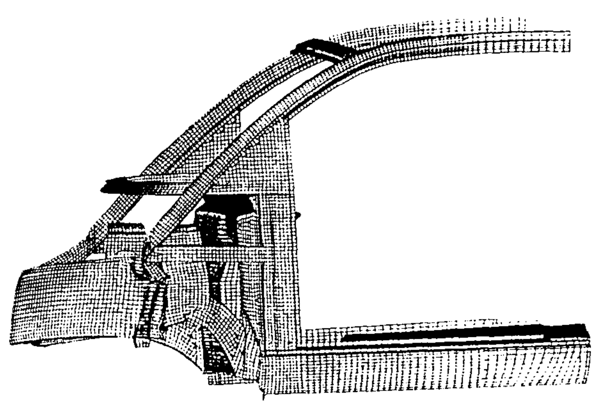

Figure 12: Deformation of the frame structure, Opel MAXX v. offset deformable barrier

In fig. 14, the initial configuration of the car-to-car simulation is shown. In this loadcase the compatibility of the Opel MAXX concept is investigated. The maximum deformation of both cars is shown in fig. 15. Although there is a mass ratio of 1/2.5, the behaviour of the small Opel MAXX car is quite acceptable. The deformation of the frame structure is shown in fig. 16. This loadcase also has low dash panel intrusions, see fig. 17. In this test configuration, the crush length for both vehicles is about 1150 mm.

In the following table, the plasticity index η_p and the energy absorption index η_w are shown for the Opel

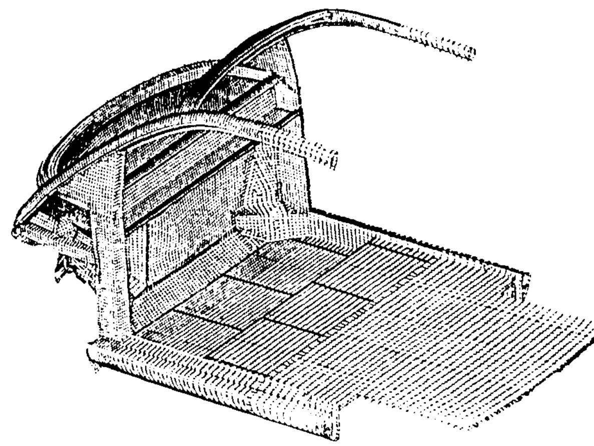

Figure 13: Deformation of the dash panel, Opel MAXX v. offset deformable barrier

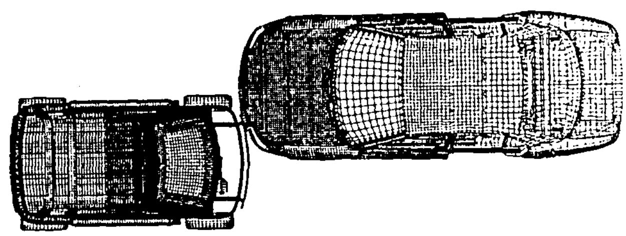

Figure 14: Opel MAXX v. Opel OMEGA

MAXX and the different loading conditions. The highest energy absorption is achieved for the car-to-car configuration. This shows the compatibility of the concept, which is most important for real accidents.

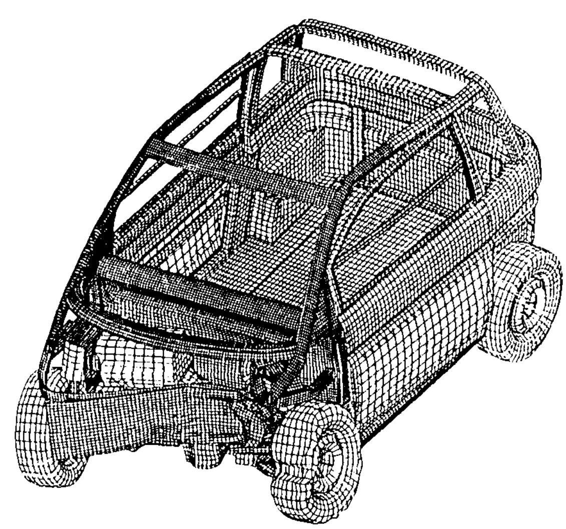

Figure 15: Deformation of the Opel MAXX v. Opel OMEGA

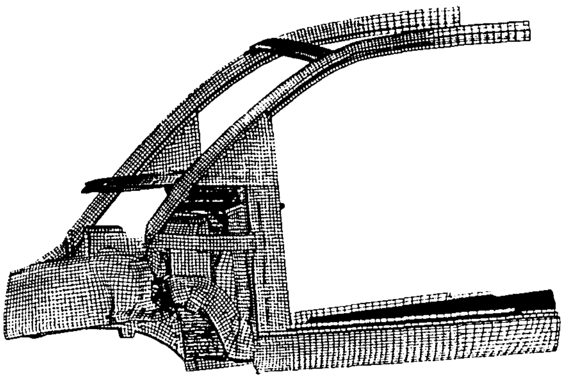

Figure 16: Deformation of the frame structure, Opel MAXX v. Opel OMEGA

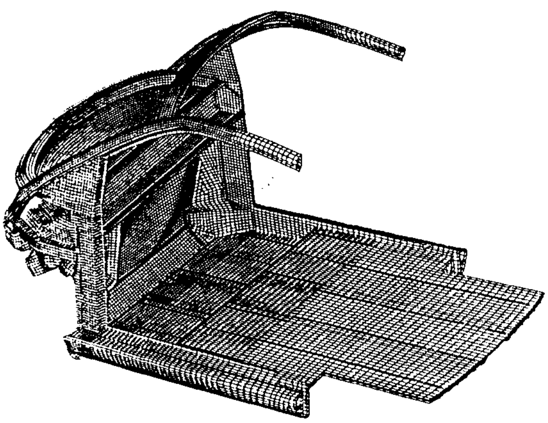

Figure 17: Deformation of the dash panel, Opel MAXX v. Opel OMEGA

Opel MAXX - plasticity and energy absorption		
Configuration	η_p	η_w
rigid barrier	0.81	0.96
ams barrier	0.49	0.74
offset def. bar.	0.64	0.87
car-to-car	0.96	0.99

The values of the Opel MAXX may be compared with those of the Opel CORSA, which are given in the following table. In general, the energy absorption is higher for the Opel CORSA. Obviously one reason is the larger front end (see fig. 1) of the Opel CORSA, which results in a bigger crush zone.

Opel CORSA - plasticity and energy absorption		
Configuration	η_p	η_w
rigid barrier	0.89	0.99
ams barrier	0.90	0.99
offset def. bar.	0.93	0.99
car-to-car	0.90	0.99

The results show that small vehicles such as the Opel MAXX can fullfil the safety standards of today. However, the behaviour of the simulations must of course be verified by the equivalent physical tests.

The investigation of the side impact behaviour is also important. An optimal performance may be achieved by combining an almost stiff structure with a well-adjusted side airbag. Side impact has not been considered in this study. However, as this paper deals only with the two-door version of the Opel MAXX, the short wheel distance would probably mean that side impact is of less importance for this case.

RESTRAINT SYSTEM AND OCCUPANT SAFETY

In this chapter the optimal design of the restraint system is discussed. Based on theoretical investigations, the system was optimized for a very small vehicle such as the Opel MAXX. As already stated, the structural behaviour must satisfy two requirements: an almost stiff passenger compartment and a crush zone optimized with respect to the plastic energy absorption. These are the basic assumptions prior to the following discussion.

The main function of the restraint system is the reduction of the high decceleration of the structure experienced by the occupants. Considering a highly optimized restraint system, the theoretical minimum for the deformation zone is 160 mm for frontal crash (see ref. /1/). For the Opel MAXX, the advantages of the new structural concept can be combined with the theoretical potential of the restraint system. Therefore the H-Point is much higher than is usual for small passenger cars (see fig. 1). This nonconventional packaging condition ensures a more upright seating position and prevents the occupants from submarining.

The mathematical investigation was performed using a rigid body approach, with the rigid barrier test with 100 % overlap at 50 kph as the chosen loadcase. The restraint system was optimized for the 50th % ile dummy in the driver's seat. Its performance was then checked for the 5th % ile and 95th % ile driver and the 50th % ile passenger. A side view of the mathematical model is shown in fig. 18.

To modify the behaviour of the restraint system, a set of five parameters has been chosen: the load limiter, the length of the belt pretensioner, the starting time of the belt pretensioner, the starting time of the airbag and the size of the vent-holes of the airbag. Several variants have to be analysed to find the optimal combination of the parameters which must reduce the injuries for the driver within a frontal crash. Any set of initial values is suitable to start the optimization process. Here the equivalent parameter set of the Opel CORSA has been chosen. In the following sensitivity analysis, the load limiter has been shown to be the most important with respect to the decceleration of the dummy. This seems reasonable, since this parameter is the main influence

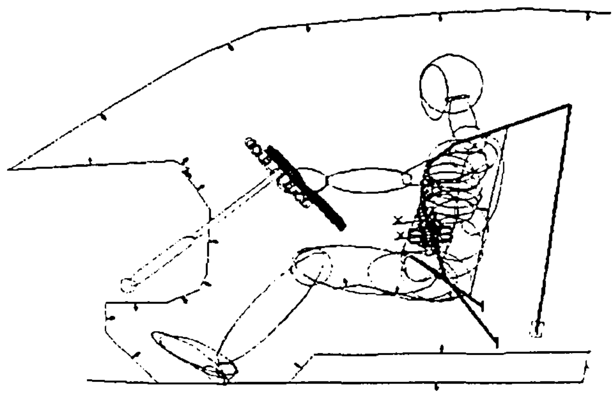

Figure 18: Optimization of the restraint system, driver, mathematical model of the Opel MAXX

on the coupling between dummy and vehicle. This coupling should be weak, so that almost the whole distance between the normal seating position and steering wheel is used to achieve a low decceleration without touching the front structure. The final values of the parameters are shown in the following table. These same values were used for the passenger restraint system.

Optimized values of the restraint system	
Parameter	Value
Load limiter	2500 N
Length of belt pretensioner	100 mm
Start time for belt pretensioner	5 ms
Start time for airbag inflation	10 ms
Size of the vent holes of the airbag	28.6 mm

Fig. 19 shows the restraint system with deployed airbag. The harmonic displacment of the dummy shows that all components of the system perform well together.

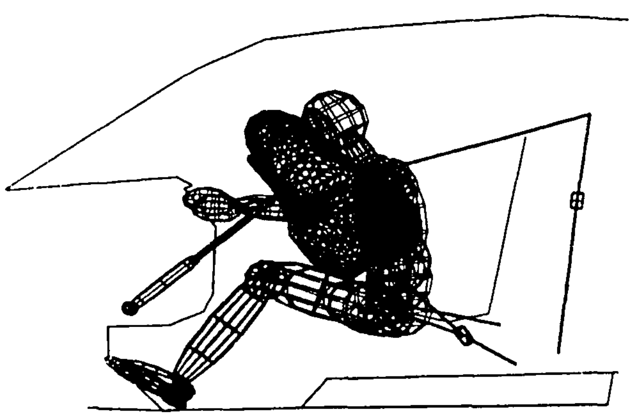

Figure 19: Restraint system in operation, mathematical model of the Opel MAXX

The following table shows the predicted decceleration values for the driver and the passenger. Both are 50th % ile dummies. The values are within accepted limits.

Predicted decceleration values - Opel MAXX		
Dummy (50th % ile)	Driver	Passenger
Head	48 g	45 g
Chest	43 g	35 g
Pelvis	52 g	53 g

The following table shows the values of the 5th % ile and the 95th % ile dummies on the driver's side. Looking at the head decceleration, it is obvious that the system has been optimized for the 50th % ile dummy. Nevertheless, the values are acceptable.

Predicted decceleration values - Opel MAXX		
Dummy (Driver)	5th % ile	95th % ile
Head	65 g	56 g
Chest	45 g	37 g
Pelvis	47 g	46 g

The results show the potential of the restraint system even for very small vehicles. Further investigations should concentrate on the other loadcases, such as offset and car-to-car.

From this study of very small vehicles, there are two obvious facts about an optimized restraint system: firstly, the main task of such a system is not to fix the occupants to the vehicle body, but rather to guide their movement optimally within the rigid passenger compartment; secondly, components such as belt and airbag have to operate well together rather than be designed independently.

Of course, the functionality of the restraint system has to be proven by a physical test. Nevertheless, it should be possible to construct such a system, since all its parameters are technically feasible.

CONCLUSIONS

The physical and geometrical conditions of very small vehicles present difficulties in achieving maximum occupant protection. To overcome these disadvantages, three characteristics are necessary for the car: an almost rigid passenger compartment, crush zones optimized with respect to energy absorption and an optimized combined restraint system. The latter should not fix the occupants to the vehicle, but guide their movement optimally, using all available passenger space in order to achieve as smooth a decceleration as possible. As this study has shown, with these three characteristics even a small vehicle such as the Opel MAXX can provide a high degree of safety.

ACKNOWLEDGEMENT

The authors would like to express their sincere appreciation for the excellent cooperation of R. Hoffmann, T. Rodemann and J. Hülsmann (Easi Engineering GmbH), D. Ulrich and T. Queckbörner (Esi GmbH) and U. Rusche and R. Emrich (Tecosim GmbH).

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CONCEPTS TO REDUCE HEAVY TRUCK AGGRESSIVITY IN TRUCK-TO-CAR COLLISIONS

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ABSTRACT

This paper describes the concept development and testing of a new front end design for heavy trucks. The design objective was to develop a bumper which reduces the aggressivity of the truck in an offset frontal collision with an automobile. Collisions involving heavy trucks and cars are particularly unfavorable to the car due to the large mismatch in mass. The bumper design concept described in this paper has evolved from a program of work which has actively focused on reducing truck aggressivity without adversely affecting the other operational characteristics of the truck. The new bumper design involves an energy absorbing honeycomb block, covered by an impact surface which swivels upon impact, thus deflecting the car away from the path of the truck. Computer simulations were used to develop the design from a basic concept to a form suitable for stationary barrier testing. The prototype bumper was fabricated and crash tested on a stationary barrier. A Ford Taurus was used as a bullet vehicle for this purpose. The test showed that the barrier deflected the vehicle as desired, with minimum intrusion into the passenger compartment. The barrier test of this design has confirmed the potential for further refinement for truck adaptation.

INTRODUCTION

When two vehicles of widely dissimilar masses collide, the lighter vehicle is at a definite disadvantage. One such example is the heavy truck to car collision, which is aggressive not only because of the much higher mass of the heavy truck, but also due to the height as well as the front-end stiffness of the heavy truck being higher. It is estimated that about 3500 light vehicle (i.e., vehicles with less than 10,000 lb. GVW) occupants are killed each year due to collisions with medium and heavy trucks (i.e., trucks with more than 26,000 lb. GVW) (Clarke et al., 1994). Furthermore, about 63,000 are injured in such collisions. Approximately 70 percent of these collisions involve the front of the truck striking some portion of the car. Based on an analysis of the NASS (National Accident Sampling System) data between the years 1981 to 1987, it has been estimated (Clarke et al., 1994) that 55 percent of the serious frontal crashes involving the front of the heavy truck can be addressed by having a suitable front bumper on the truck.

BUMPER DESIGN CONCEPT DEVELOPMENT

Purpose of Bumper

The primary purpose of the heavy truck bumper is to deflect the car away from the path of the truck during an offset frontal collision. By deflecting the car, the gross mismatch in mass is addressed since it is not practical to consider designs that would absorb and dissipate all the kinetic energy involved in such collisions. Also, appropriate compatibility in height and front end stiffness between the colliding heavy truck and the car is desirable so that override (Clarke et al., 1994), snaring, excessive crush, excessive deflection and large post-impact velocity of the car are also avoided. Excessive deflection and large post-impact velocity are undesirable because of the increased potential for secondary impacts of the car.

Design and Development Approach

Computer simulation was used extensively to develop and refine the heavy truck bumper design concept. Crash simulation models of the bumper and a typical mid size car were created. Both lumped parameter and finite element crash simulation methodologies were utilized. Lumped parameter models were used primarily in the initial stages to evaluate overall response trends, followed by coupled lumped parameter and finite element models for more detailed concept development. Finally, complete finite element model simulations were used for design verification. Also, 3 full-scale tests were conducted to validate the design.

Lumped Mass Model of a Midsize Car

A lumped parameter model of a midsize car was created. The Ford Taurus was selected as a representative baseline vehicle for the preliminary crash simulation studies because it had been used in prior crash tests pertaining to heavy truck aggressivity reduction by NHTSA. The topology of the model is illustrated in Figure 1. The masses and moments of inertia of the eight lumped masses in the model were derived from the report by Varadappa (1993). The crush characteristics of the six discrete springs in the model were derived from both Patwa (1991) and from additional finite element crash simulation of the Ford Taurus model discussed in the report by Varadappa (1993). Contact stiffness functions for component stack up were derived from test data in the same report. The immediate application of the model was to use it as a representative vehicle for designing a crash barrier. Therefore a complete validation of the deceleration pulse was not necessary as long as realistic crush characteristics were achieved. Figure 2 shows the crush and velocity of the vehicle compared against corresponding test data and finite element model results from Varadappa (1993).

One of the previous crash tests that had been performed by NHTSA under the heavy truck aggressivity program involved an angled barrier crash of a Ford Taurus at 50 mph (Johnston, 1994). The barrier had been angled both sideways and upwards. This test subjected the front end of the car to lateral and vertical crush forces simultaneously. A vehicle hitting the proposed heavy truck front end design would also be subjected to similar loading conditions. Therefore the MADYMO model was modified by adding lateral stiffness between the lumped masses which define the front end structure so that the trajectory of the simulation response was similar to that observed in the actual test. Figure 3 shows the MADYMO car model and the simulation verification of the vehicle trajectory for the angled barrier.

Bumper Design Concept

A preliminary design concept for a heavy truck front end was synthesized for the primary purpose of deflecting the car. This design involves a high, relatively rigid, front bumper which rotates upon offset impact and offers an angled surface for the impacting vehicle to slide away from the path of the truck. An early illustration of the concept using simulation is shown in Figure 4. It is important that the impact surface remain as flat as possible during the collision in order to promote sliding behavior. It was decided that a physical pivot would not be practical as it would increase the aggressivity of the truck in a direct frontal impact. Instead a design was sought in which the bumper would rotate under an offset impact but still yield sufficiently in frontal impact. Therefore the selected approach was to use a block of energy absorbing material behind the bumper plate shaped in way that would yield the desired kinematics in an offset crash.

It was first necessary to determine the effect of barrier angle on the vehicle. A series of simulations were performed with a rigid barrier set at angles varying from 30deg to 55deg. A 50mph impact speed was used. Figure 5 shows the x and y velocity at the center of gravity of the vehicle. At lower barrier angles the vehicle is subjected to very high effective frontal impact forces. At higher barrier angles the car retains more of its impact velocity and undergoes smaller ΔV. Based on this analysis it was decided that the concept bumper should rotate about 45 deg during impact. At this angle the change in forward velocity is about 30 mph at a 50mph closing speed.

A finite element model of the preliminary bumper concept was created using the LS-INGRID mesh generator for simulation in LS-DYNA. The MADYMO model of the car and the finite element model of the bumper concept were simulated using the coupled simulation method. The geometry of the bumper concept was revised extensively based on the simulation results until a satisfactory design was achieved. Figure 6 shows some of the stages of the evolutionary design process. The ellipsoids in the figure are parts of the MADYMO car model defined for contact interaction with the finite element model. Figure 7 shows the MADYMO car model and the finite element bumper model set up for a coupled simulation.

The main components of the final barrier design that evolved from the iterative refinement process is illustrated in Figure 8. The design consists of three parts which are, a front end impact plate, an Aluminum honeycomb block, and a rear support plate. The front impact surface is 0.75 m high, 2.43 m wide, and is positioned 0.25 m off the ground. It has six ribs welded on to the rear surface. The assembly is fabricated from 4 mm thick steel plate. The type of aluminum honeycomb in the model has a density of 25 Kg. The geometry of the honeycomb block and the rear support plate are designed to achieve the required kinematic behavior of the impact plate. The entire assembly has a mass of approximately 115 kg. In an actual truck application the impact plate may be covered by a bumper fascia. Additional support structure for attaching these components to the truck frame without hindering the crash kinematics would also be in place.

For the preliminary barrier crash tests the impact plate is supported on two beams which project outward from the stationary barrier. The colliding vehicle hits the impact plate, which distributes the impact forces over the entire face of the honeycomb behind it. The honeycomb block absorbs some energy during the process. During the course of the impact the honeycomb crushes in a controlled manner so that the impact plate rotates outwards. The honeycomb helps to distribute the impact force on the rear support plate and henceforth to the truck chassis rails.

The MADYMO lumped mass model was used to develop the bumper concept. A more accurate verification of the design was performed using a finite element vehicle model as shown in Figure 9. Figure 10 shows the deformation of the vehicle and barrier in a 50 mph offset impact. Figure 11 shows how the kinetic energy of the car impacting at 50 mph is transformed during the course of the simulation. The barrier absorbs 21 percent of the energy. Friction and other dissipative effects account for 23 percent. The vehicle structure absorbs 36 percent. Twenty percent of the initial kinetic energy remains in the vehicle as it is deflected away at 23mph.

DESIGN VALIDATION USING FULL-SCALE TESTS

The FEM model of the bumper developed was validated by three full-scale tests. The crash tests are described below:

Test 20 (Test Date:05/25/95)

The first test consisted of a stationary front-end design being impacted by a moving passenger car (Ford Taurus). The front end fabricated for this test is shown in Figures 12 and 13.

This was an offset frontal impact between a stationary simulated truck front-end bumper design described earlier and a 1989 Ford Taurus, mass 1,592 Kg (3,502 lbf) traveling at 88.7 kph (55.1mph). The HIC was 298, and the 3 msec clip peak chest g's was 39.7. This test was a follow-up to Test #18 (Johnston, 1994) to deflect the car and have a better post-impact trajectory.

Upon impact, the dummy translated forward and to the left across the seat. Both knees impacted the instrument panel. The dummy was restrained by the three point belt. The dummy rebounded rearward and to the right. The dummy then came to rest facing forward and slightly leaning towards the right.

The injury readings were low as the ΔV from the impact was 27 mph in the longitudinal direction and 16 mph in the lateral direction. The vehicle was still moving at 31 mph on separating from the barrier. This effectively deflected the vehicle and made the car-truck collision a less severe event. The honeycomb between the front rotating plate and the rear rigid plane crushed and absorbed some of the impact energy. In general, the kinematics of the barrier matched the results of FEM analysis.

There was no override of the car bumper structure. The damage was limited to the front left corner and was 506 mm (20 inches). There was no intrusion of any component into the vehicle's occupant compartment.

Test 21(Test Date : 06/16/95)

After demonstrating the success of the deflection technique, a baseline test was run using a moving heavy truck to impact a stationary car. A 1969 cab-over engine tractor, shown in Fig 15 and 16 was used in the test to determine the effect of the cab structure. The test showed that for cab-over trucks, in certain front offset impacts, the cab can have a major role in causing serious to fatal injuries to the passenger car driver.

This was an offset frontal impact between a stationary 1987 Ford Taurus and the truck frame with a cab loaded to 11,318 kg (24,900 lbf) traveling at 88.7 kph (55.1 mph). The head injury readings to the Hybrid III in the driver's seat were the highest of all the tests conducted. The HIC was 4609, and the 3 msec clip peak g's was 47. The head x-axis accelerometer reached full scale at 99 msec, when the dummy head impacted the truck cab around the headlight housing. Actual HIC is therefore expected to be higher than the recorded value.

The truck was in the standard configuration, with no axle set-back. The left portion of the truck bumper was pushed rearward and upward. The bumper was cracked where it was mounted to the frame. The steering gearbox was also damaged. The truck axle dislocated from the front and shifted rearward, There was extensive damage to the car, with the steering wheel hub intruding about 10 inches into the occupant compartment. Much of this damage was caused by the truck's cab structure impacting the car's A-pillar and windshield.

Upon impact, the dummy translated forward across the driver's seat and both the knees impacted the lower instrument panel. The right side of the dummy's thorax impacted the steering wheel rim, and the dummy head hit the front left corner of the truck cab near the headlight housing. There was significant override of the car structure by the truck. The extent of damage was more than that in previous baseline tests with extended bumpers and more closely represented those seen in real world crashes.

Test 22 (Test date : 10/23/95)

Finally, a test was run with the deflecting front end mounted on a moving truck, impacting a stationary car.

This was an angled frontal impact between a stationary 1988 Ford Taurus and the truck frame with a cab and the modified front (similar to the one used in Test # 20), loaded to 10,099 kg (22,265 lbf) traveling at 89.2 kph (55.4 mph). The head injury readings to the Hybrid III in the driver's seat were the lowest of all the tests conducted. The HIC was 176, and the 3msec clip peak chest g's was 29.6. The other readings from the dummy instrumentation were also very low (no chest deflection, and femur loads of 295 and 561 N).

The test kinematics for the car and its occupant were similar to Test 20. There was little damage to the truck cab or understructure, and the damage to the modified front was similar to that in Test 20. The car front left corner was crushed with little or no intrusion into the occupant compartment.

Upon impact, the dummy translated forward and to the left across the seat. Both knees impacted the lower instrument panel. The dummy was restrained by the three point belt and rebounded rearward and to the right. The extent of damage and injury readings were much less than that in the comparable baseline (Test # 21) as indicated in Table 1. The vehicle and dummy kinematics were similar to the test with static truck front (Test # 20).

CONCLUSIONS

Deflecting a car by using a swiveling truck front appears to reduce the severity of offset truck-car collisions. Further research will examine the possibility of addressing the problem centerline and minor offset collisions. This approach also provides for a lower deflecting front with some energy absorbing capacity, and without a rigid center pivot. Such a truck front structure could also result in fewer tripped truck rollovers involving guard-rails, thereby preventing some collisions with other vehicles in opposing traffic lanes as well as truck occupant fatalities and injuries due to rollovers.

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FIGURES

Injury Measure	*Baseline Bumper*	*Modified Bumper*
HIC	4609	176
Chest G	46.9	29.6
Chest Deflection	28.7 mm	-----
Left Femur Load	464 N	295 N
Right Femur Load	3986 N	561 N

Table 1 Comparison of Injury Numbers from Tests

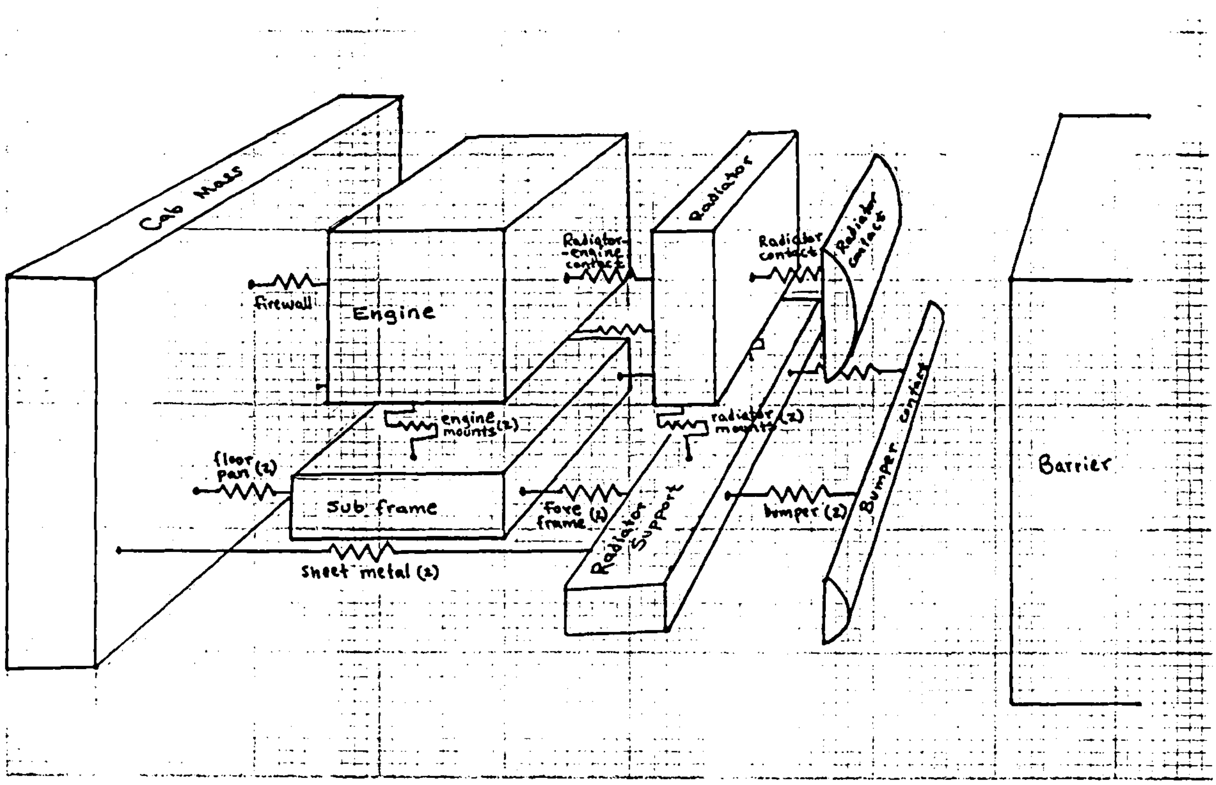


Figure 1 Mid-size Car Lumped Mass Model Topology

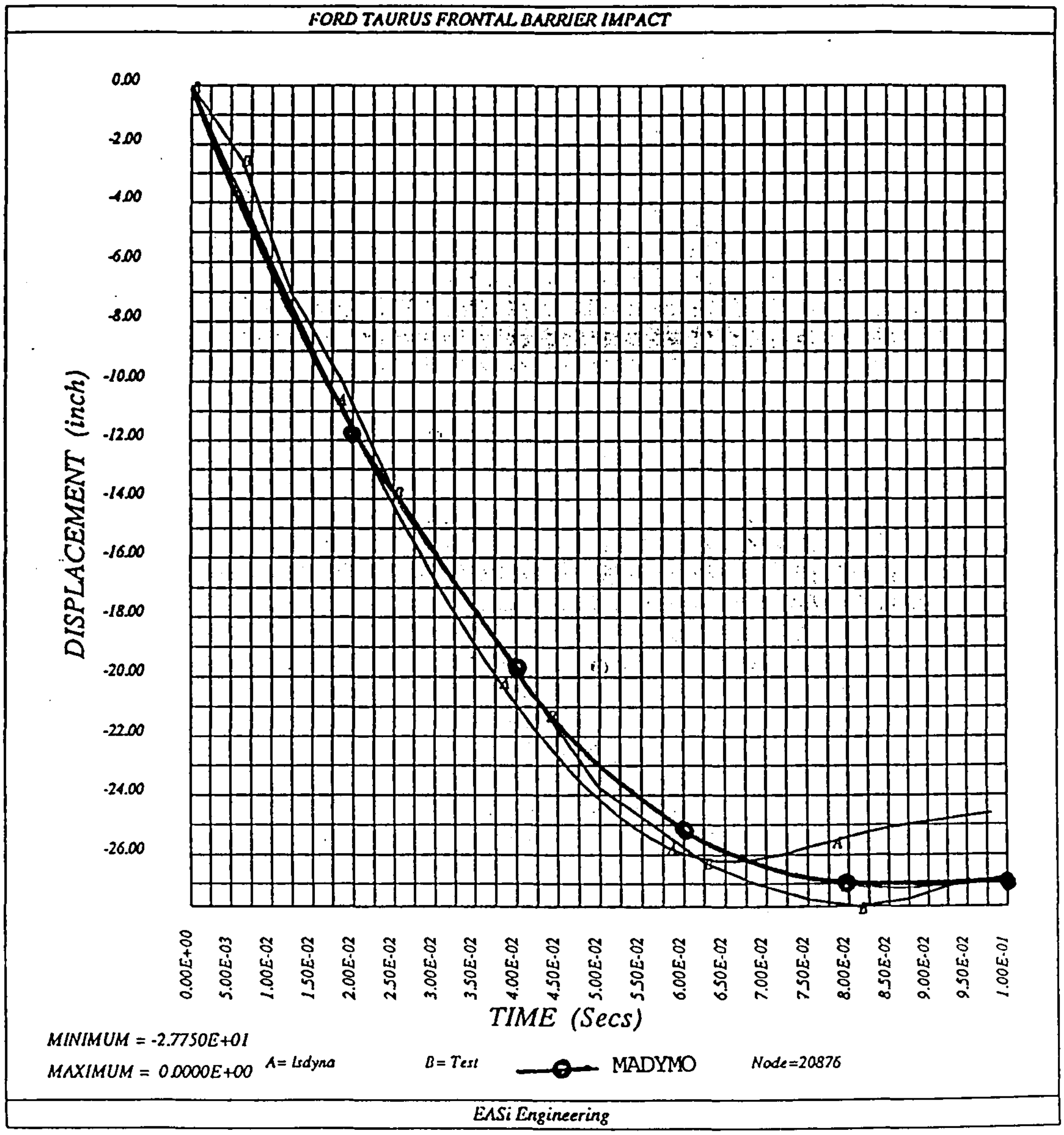


Figure 2 35 mph Frontal Crash Response of Lumped Mass Model

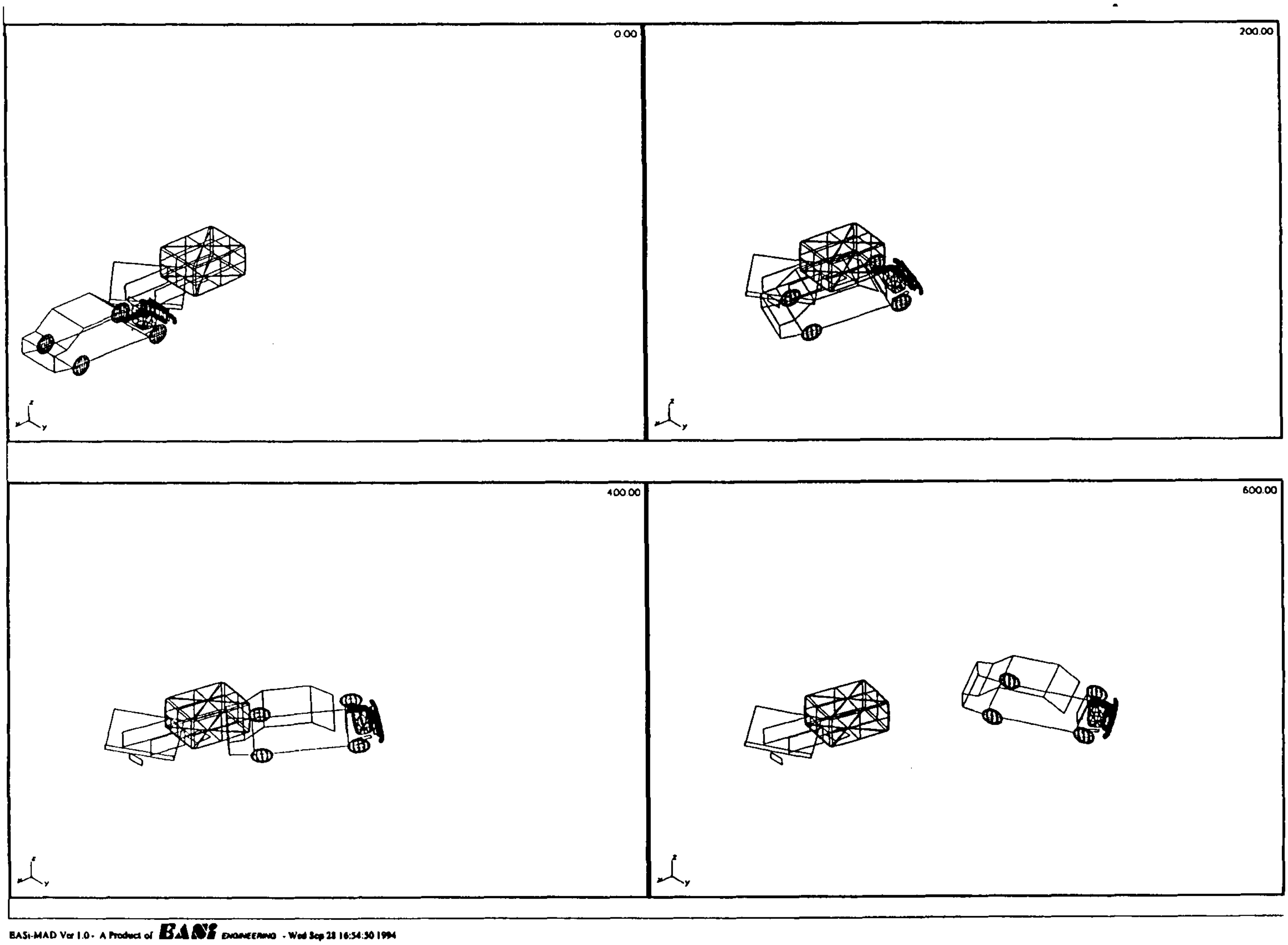


Figure 3 Angled Barrier Simulation

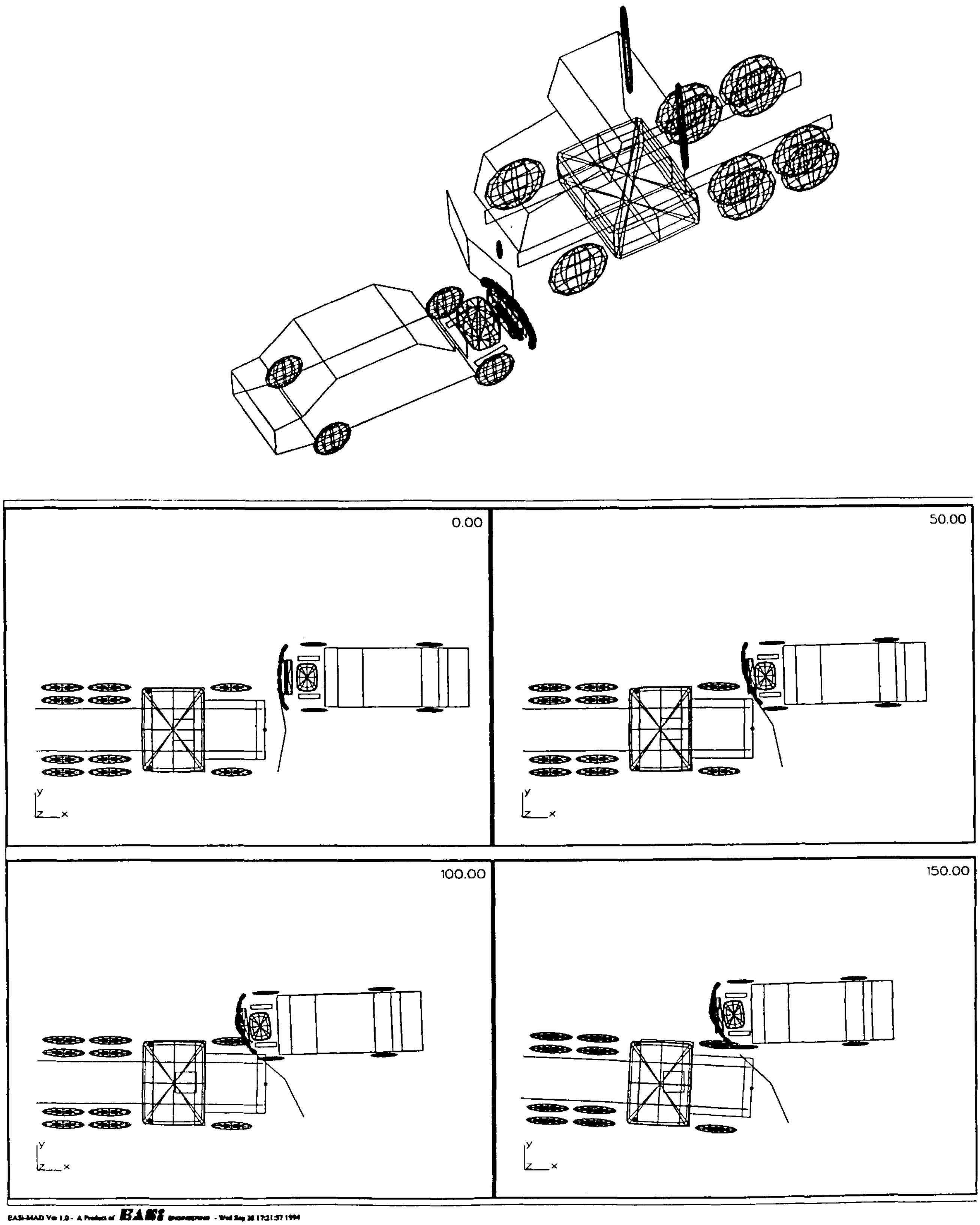


Figure 4 Preliminary Concept

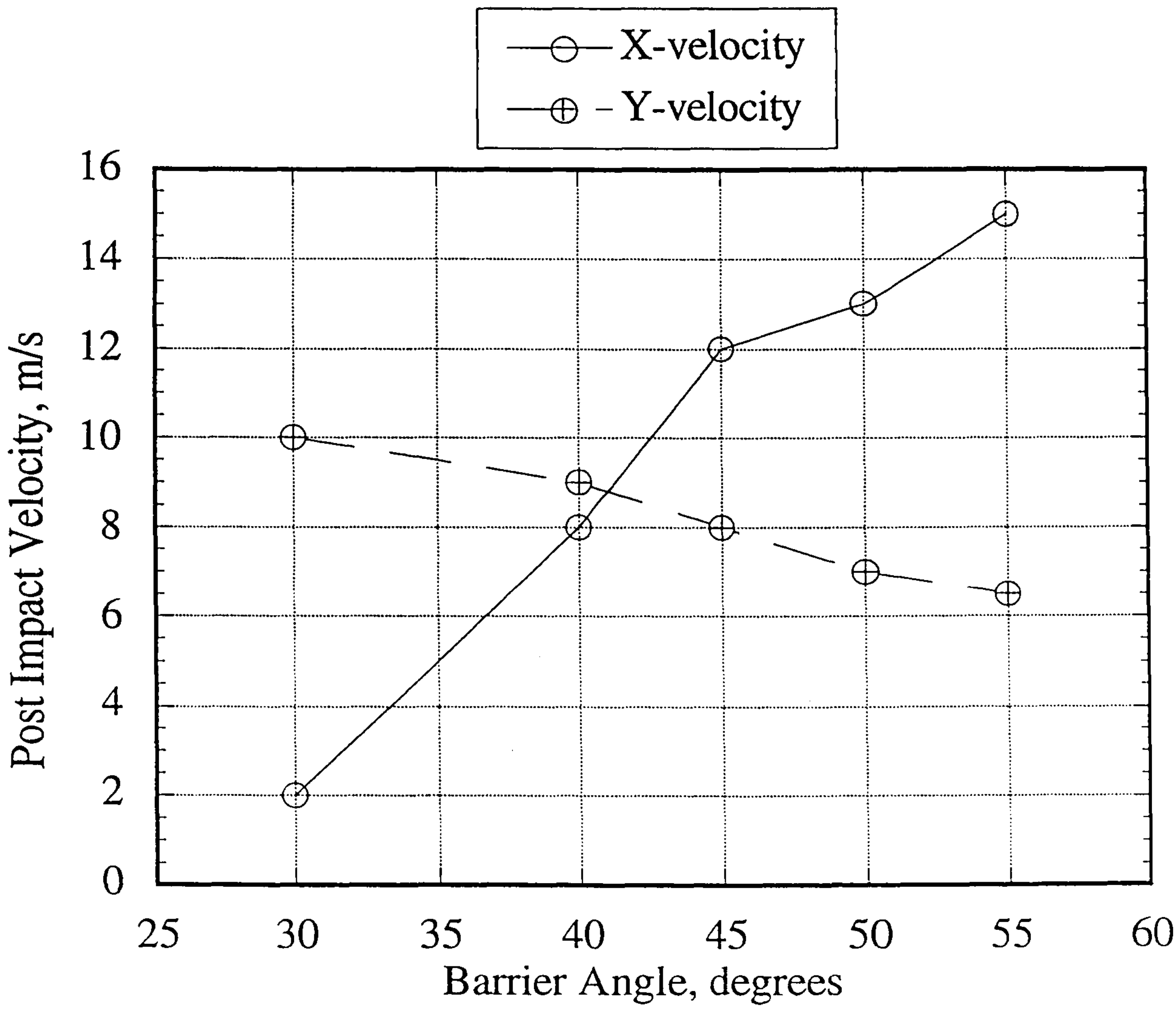


Figure 5 Effect of Barrier Angle on Post-impact Velocity

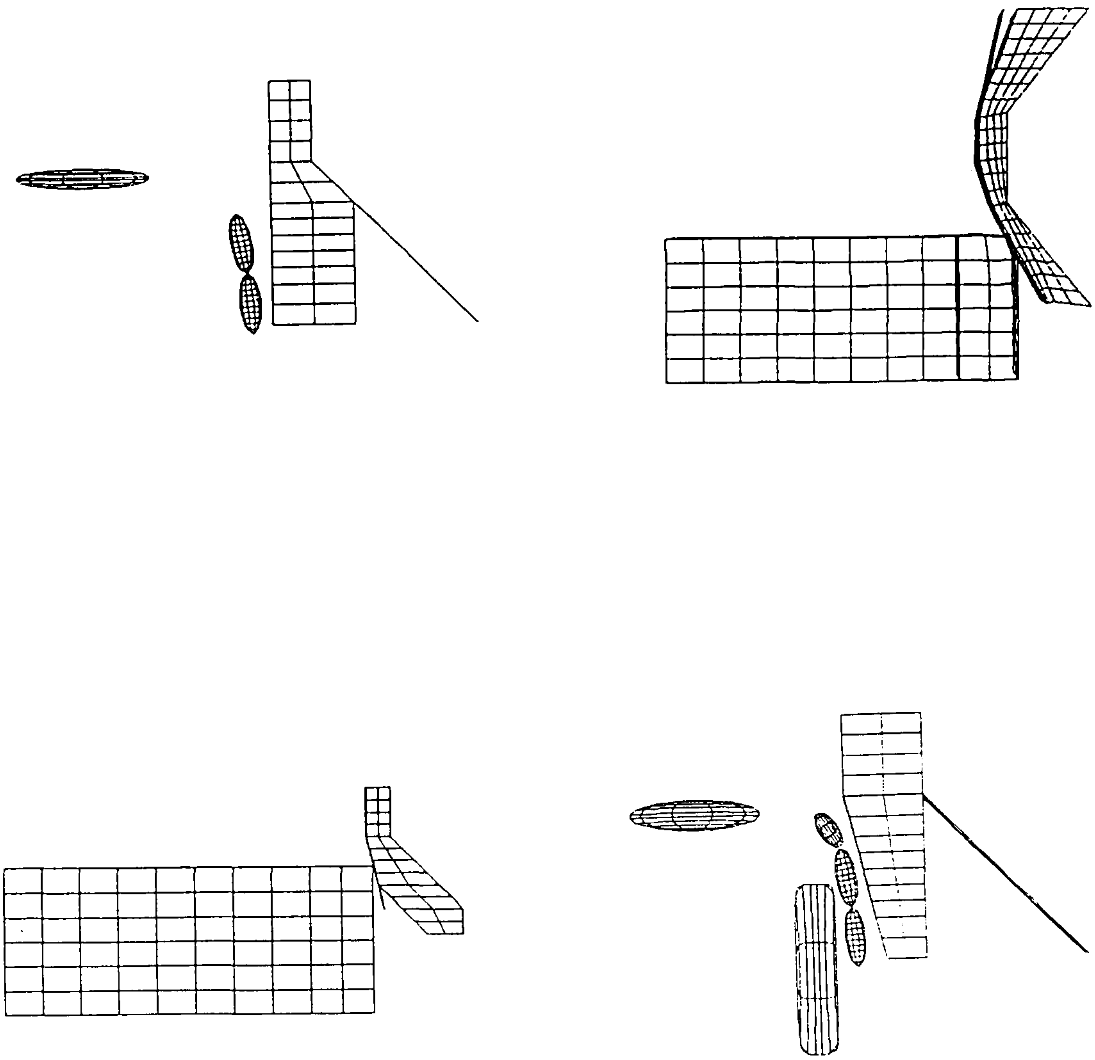


Figure 6 Design Evolution of the Bumper Concept

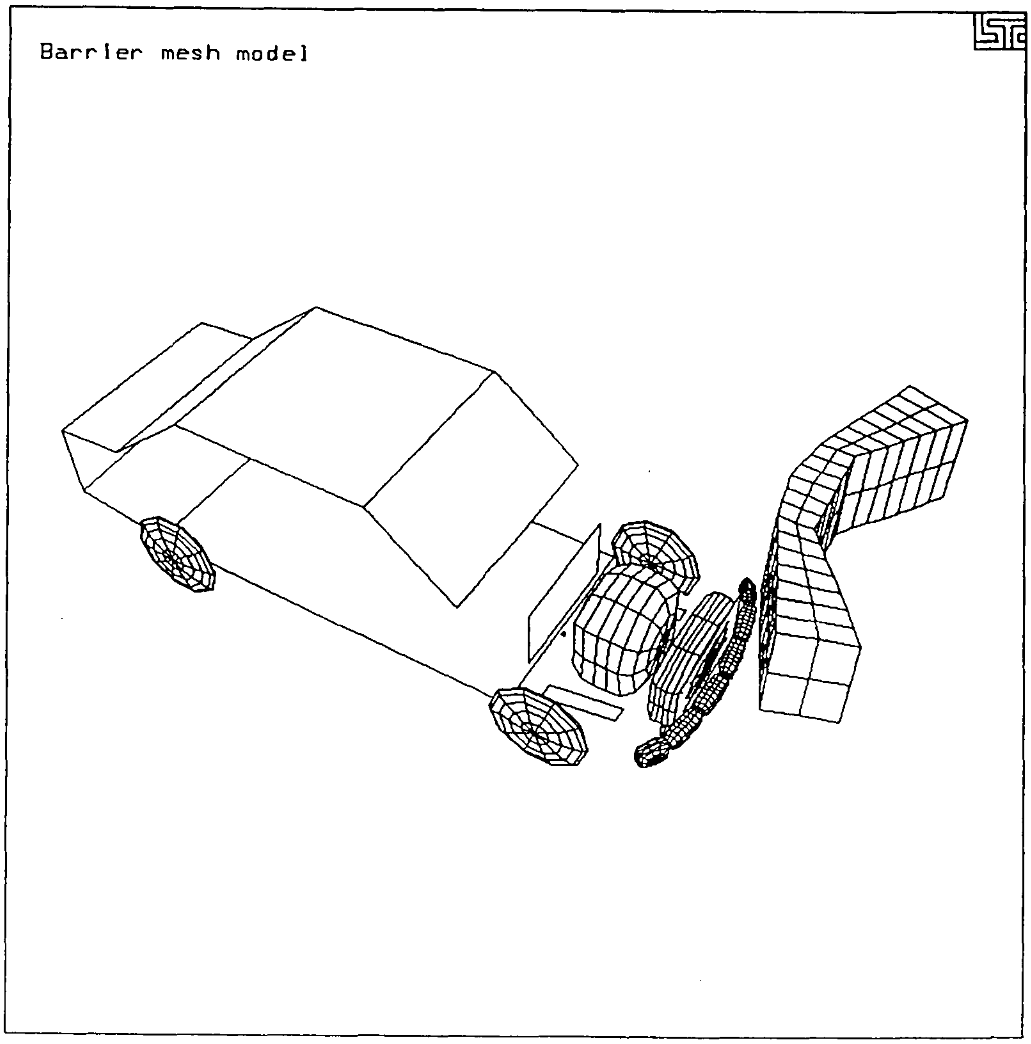


Figure 7 Coupled Simulation Model

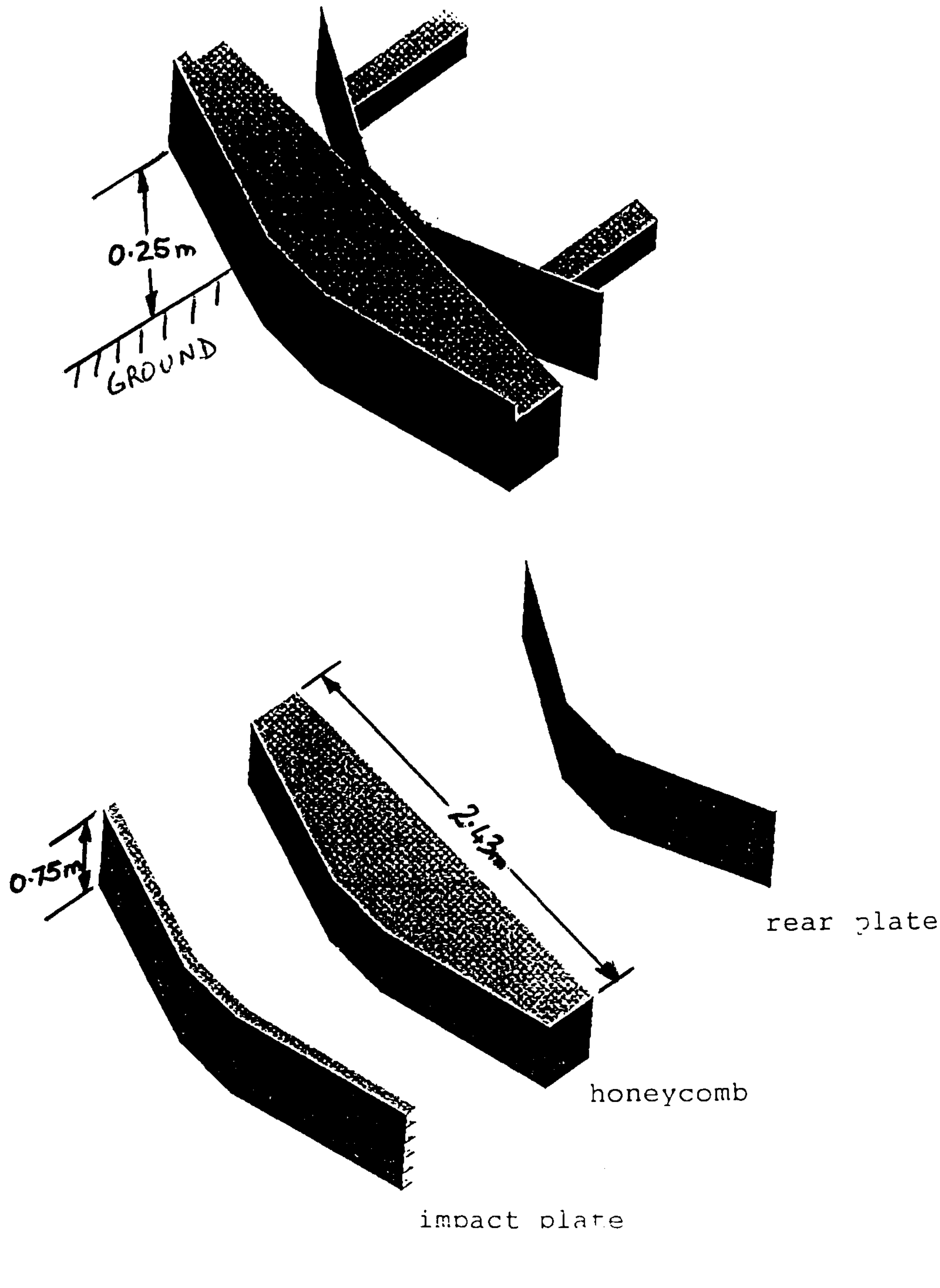


Figure 8 Design Concept Derived From Simulation

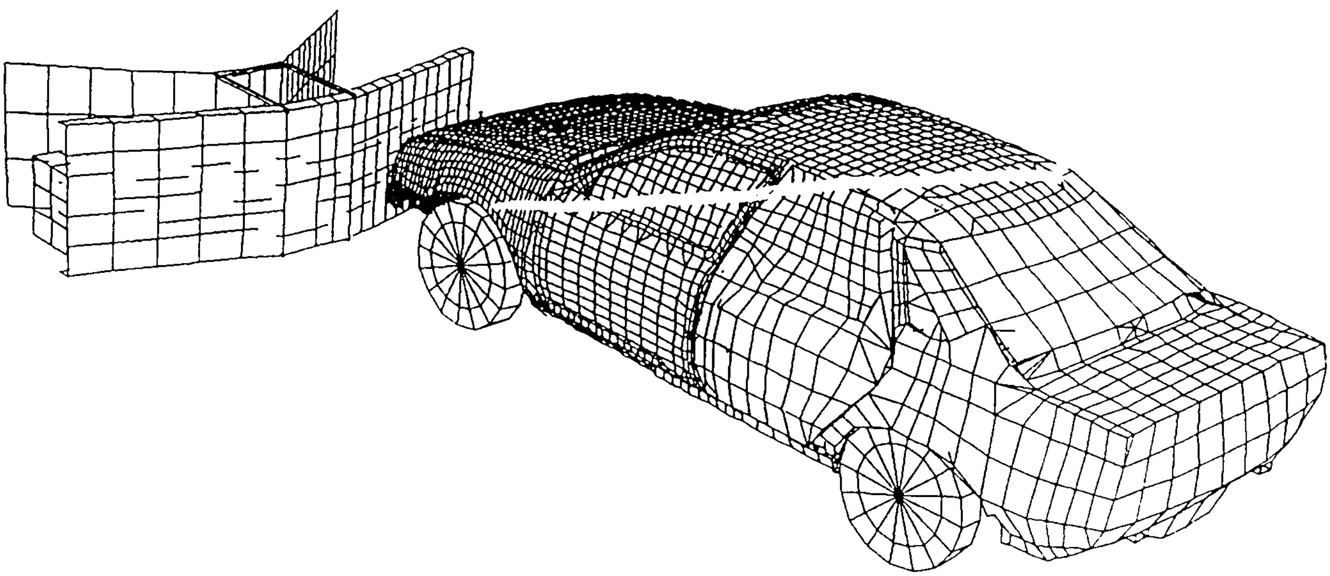


Figure 9 Finite Element Simulation Model

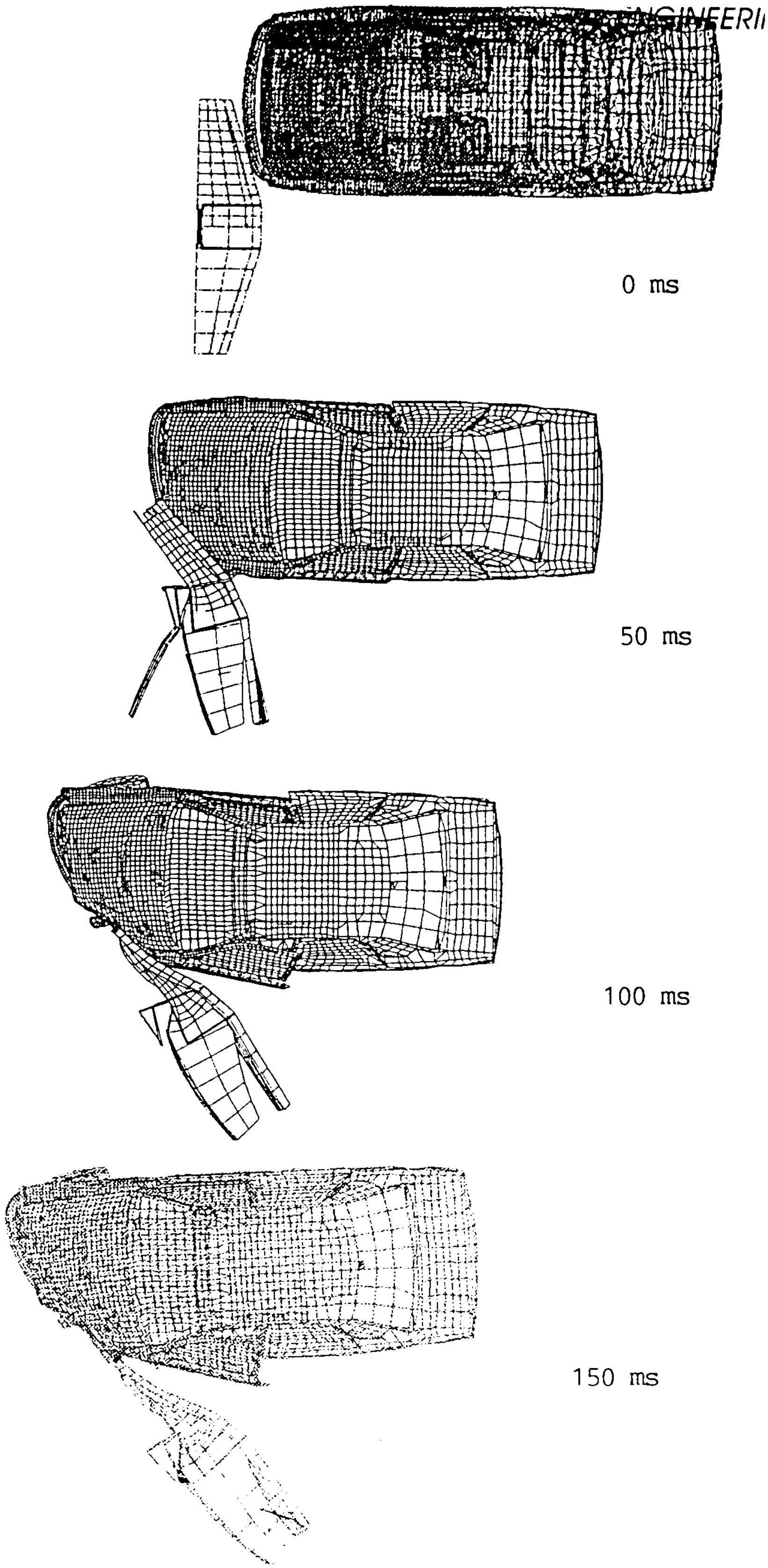


Figure 10 Verification of Simulation Using the Finite Element Car Model

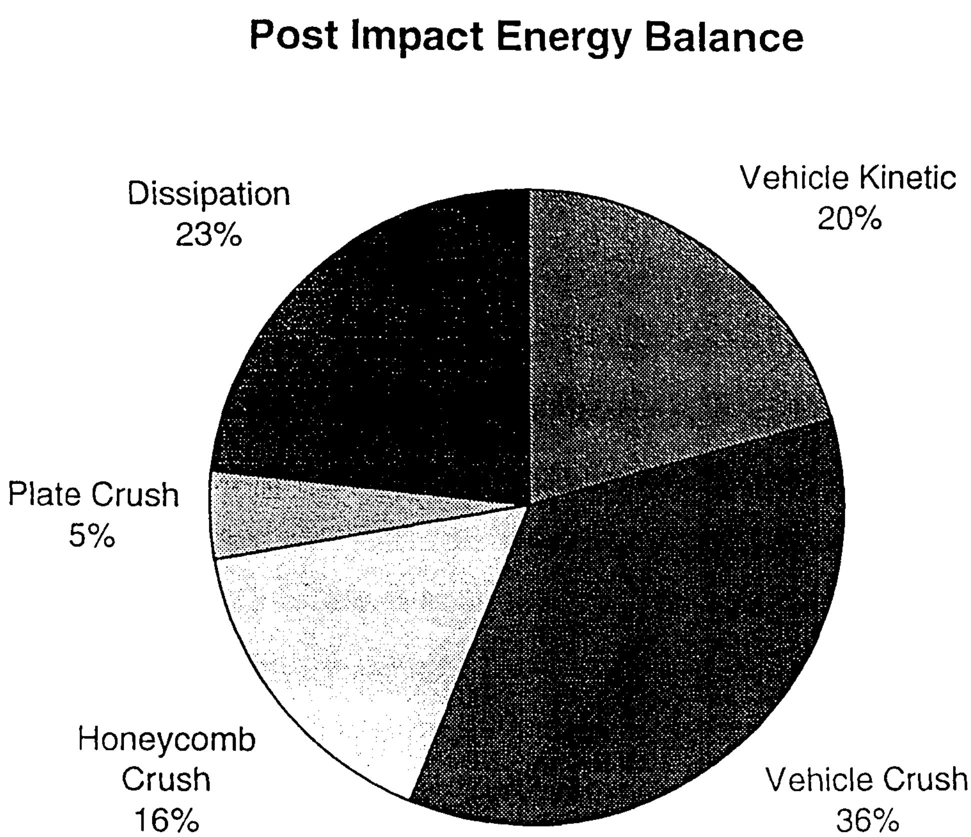


Figure 10 Conversion of Initial Kinetic Energy During Impact

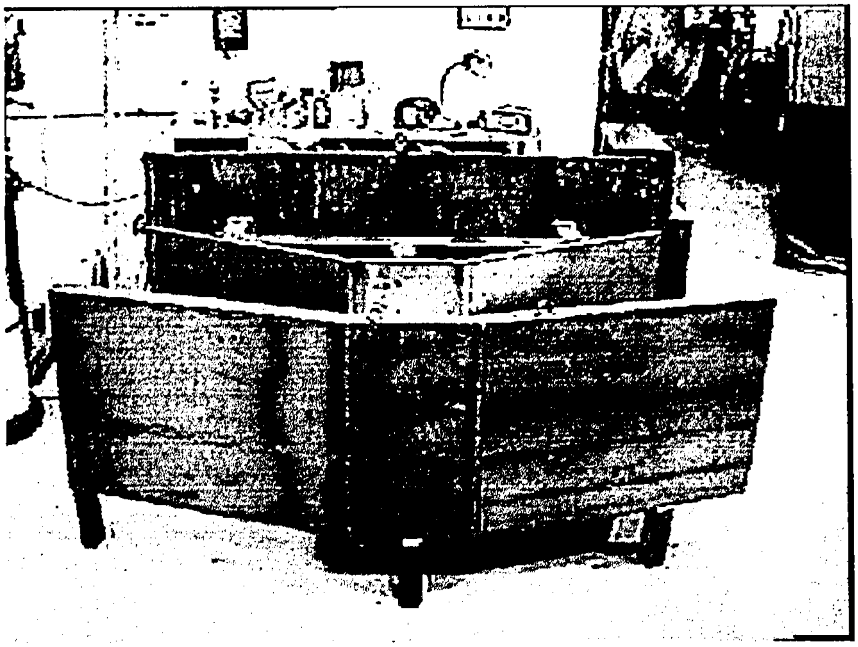


Figure 12 Modified Front-end Front View

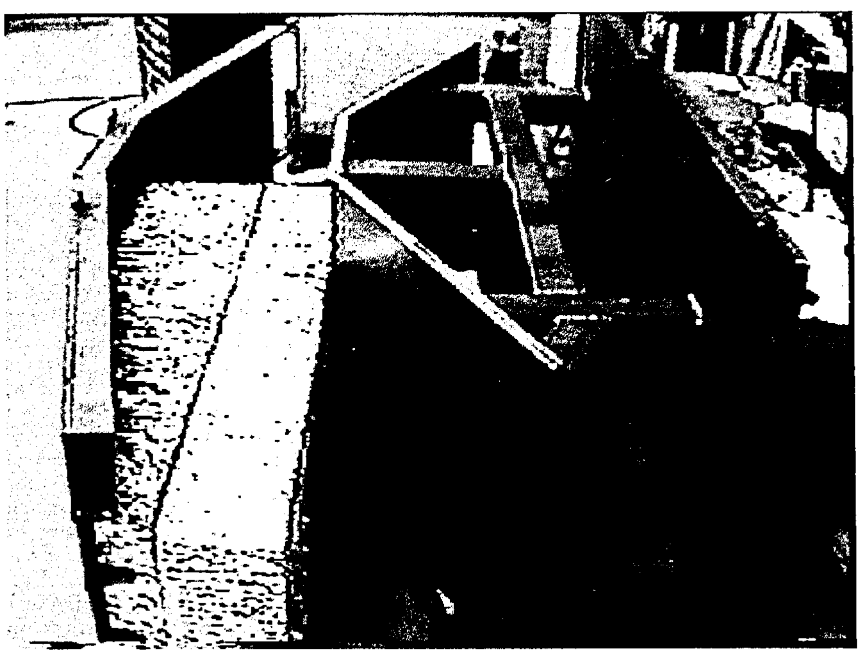


Figure 13 Modified Front-end Side View

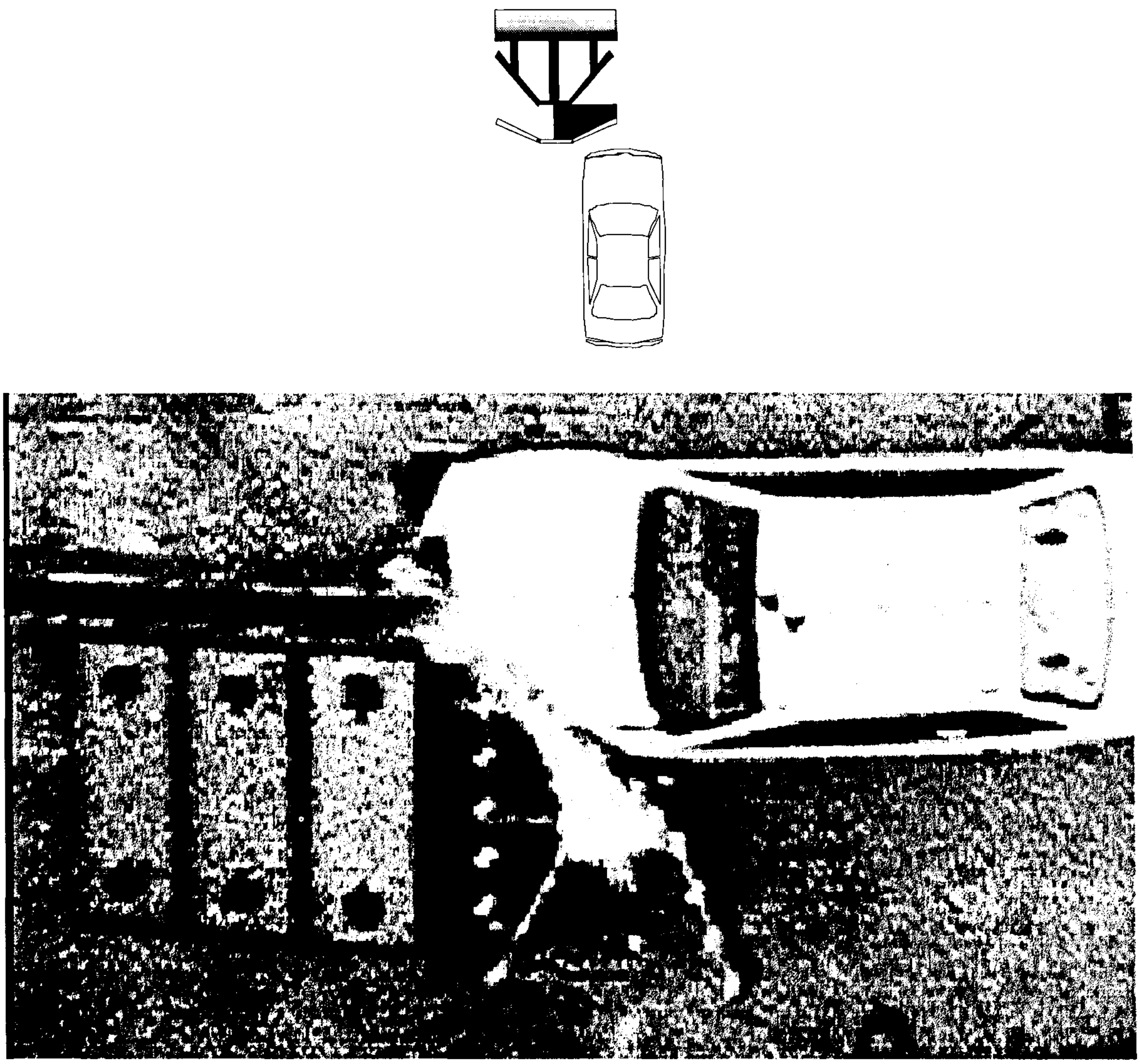


Figure 14 Test # 20

Figure 15 Truck with cab

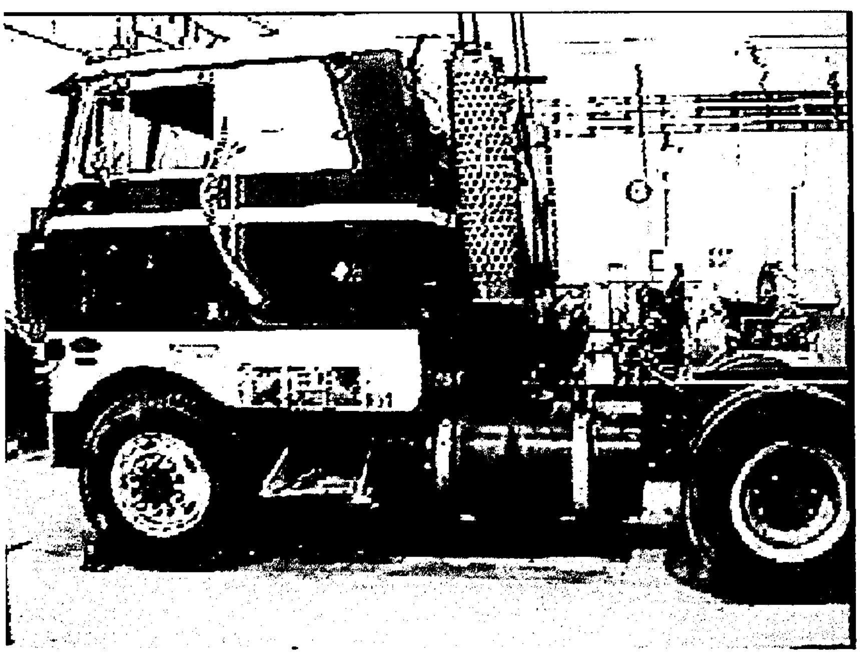


Figure 16 Truck with cab

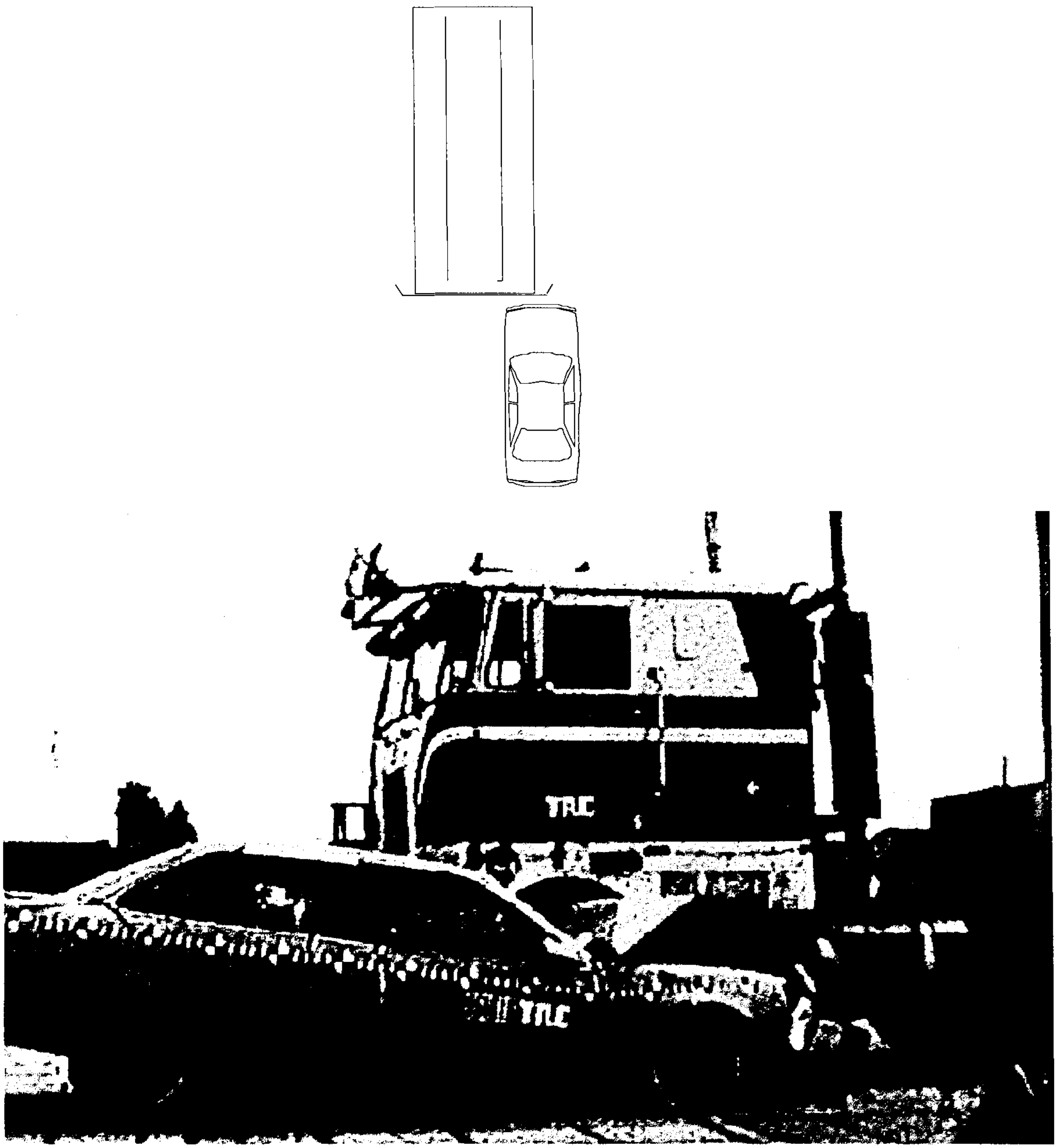
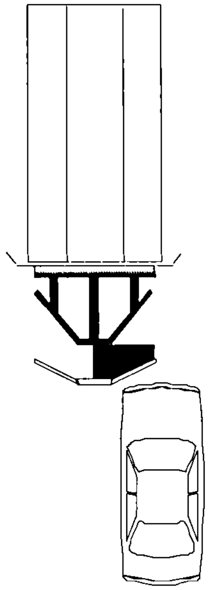


Figure 17   Test # 21



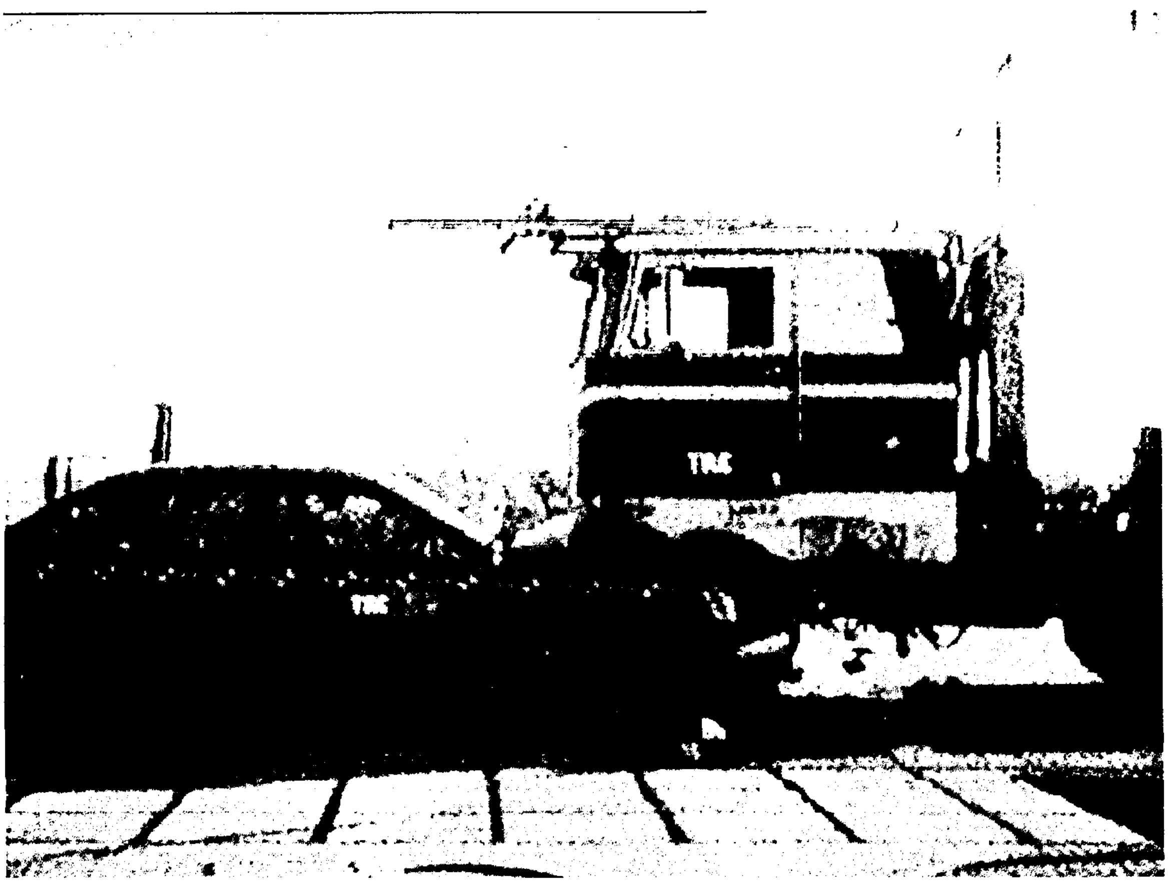


Figure 18 Test # 22

ASSESSMENT OF MEASURES REDUCING RESIDUAL SEVERE AND FATAL INJURIES MAIS 3+ OF CAR OCCUPANTS

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Paper Number 96-S4-W-18

ABSTRACT

The Accident Research Unit at the Medical University Hannover carried out an in-depth investigation of road accidents in a statistical representative manner. Each year approx. 1000 accidents are collected on order of BASt. These data are a good tool for analysing the accident scene of severe car crashes, answering the question, how car crashes of severe and fatal injury occurence look like. While many safety measures for car occupants and test procedures for car developments exist, 1/3 of all severe and fatal injured persons in road accidents are car occupants. 5.5% of all car occupants involved in road accidents suffered injury severities of more or equal than MAIS 3, but it must be the aim to avoid these victims. For this purpose the accident situations were analysed in detail.
In the study the injury mechanisms are described, measures reducing these residual severe and fatal injuries assessed and demands for test procedures and car design formulated with regard to an optimized occupant protection.

INTRODUCTION

The amount of fatally injured car passengers has steadily reduced during the past years (figure 1). 8,989 killed car passengers were registered in Germany in 1970, compared to only 3,974 in 1994 (StBA-1). This means a reduction of 56%. The amount of severely injured has also reduced by 49% during the same period of time, but the amount of slightly injured has only reduced by 14%.

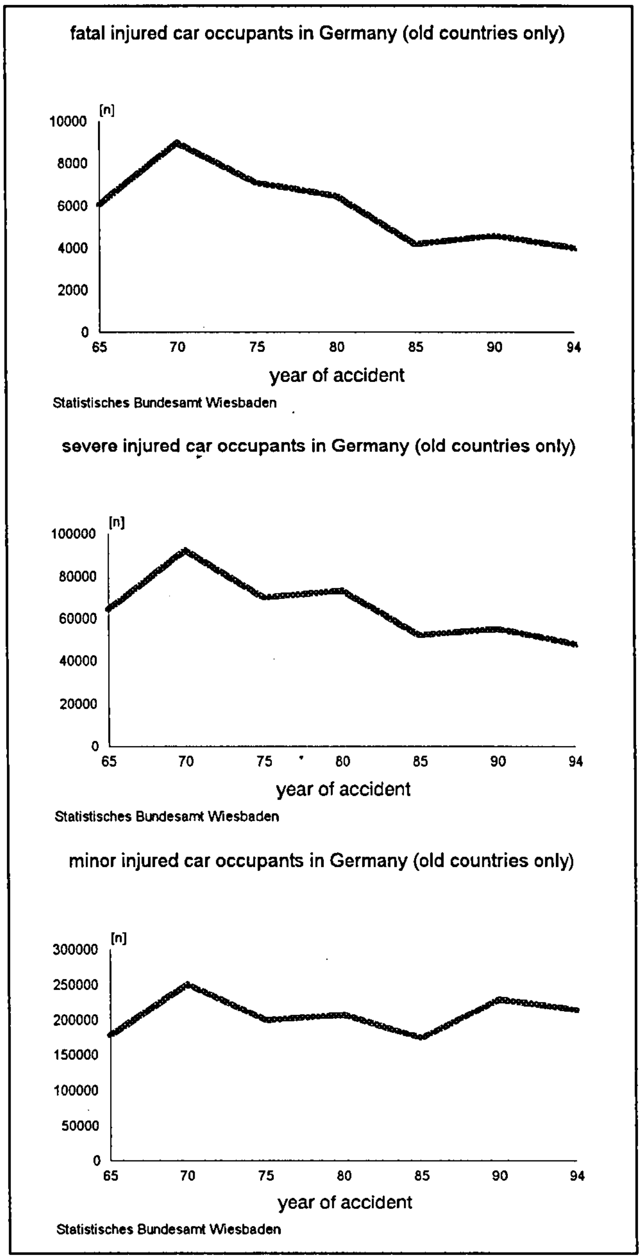


figure 1 injury situation of car occupants in Germany (old countries only) from 1965 to 1994

This has mainly been achieved by the development of vehicle safety measures such as the safety belt, the airbag and the crumple zone, as 50% of all collisions concerning injured passengers are frontal collisions, 32% are lateral collisions and 13% are rear collisions and most of the safety elements come into effect during frontal collisions. But figure 2 shows the collision situations of car drivers killed in accidents and it appears that 65% of all fatally injured car passengers are killed during lateral collisions, 34% were subjected to an isolated lateral collision and 31% were subjected to a multiple collision involving the vehicle side.

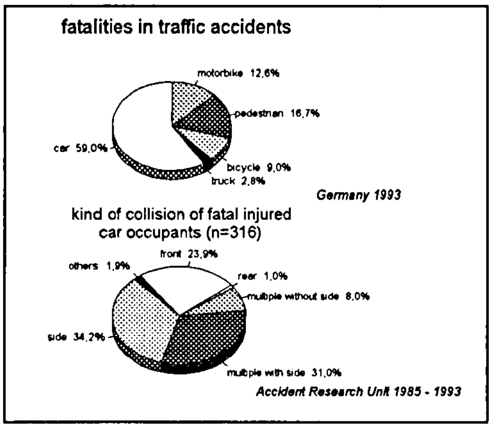


figure 2 percentage of fatalities in Germany 1993 (source: Statistisches Bundesamt Wiesbaden) and kind of collision of fatal injured car occupants (source: Accident Research Unit Hannover)

It is quite likely that passengers killed in road traffic are subject to special accident and collision situations, during which available safety precautions are either of low or even no effect. To ensure that further methods towards accident prophylaxis can be more effective, it is important to know the circumstances of the accident mechanisms for severely and fatally injured occupants. Which means that the severly injured, who are still present in todays road traffic situation, have to be surveyed within the framework of this study. The investigations at the scene of the accident in the Hannover area, in which a special scientific investigation team drives to the site of the accident and documentates its findings within the framework of a statistical sample survey, can be used for this purpose. The results of Hannovers accident evaluations can be regarded as representative.

THE INVESTIGATION METHOD

Since 1973, traffic accidents in the greater district of Hannover have been registered on the site and vehicle damages as well as injuries have been stored in a data bank, by order of the German Federal Highway Research Institute BAST. Since 1985 the accident documentation is based on a statistical random sample and the evaluation is carried out by means of a weighting method (Otte-2). The accidents evaluated in the course of the study were recorded between 1985 and 1994. The Abbreviated Injury Scale (AIS) is used to describe the accident severity (American Association of Automotive Medicine-3). Vehicle deformations are recorded and measured by photographic methods. The vehicle interior is searched for contact and collision points and deformations are represented by means of a computerized matrix system (Otte-4). An extensive reconstruction supplies details of the collision speed as well as the movement behaviour of the vehicles and the occupants. The speed variations caused by the collision delta-v calculated by a mathematical-physical collision analysis is used to assess the forces applied to the passengers. This enables a description of the injury mechanisms as well as the occured load conditions of the documentated cases within the framework of this study.

RESULTS OF THE STUDY

Assessment and Defintion of the Severely and Fatally Injured of MAIS 3+

The killed and severely injured persons found in official statistics are classified according to hospital treatment. According to the statistics, out-patients are regarded as „minor injured“, in-patients are „seriously injured“ and those who die within 30 days are regarded as „killed“. The scientific approach, on the other hand, uses the AIS scaling method. This incorporates a numeric scale of the degrees 1 (minor injured) to 6 (fatally injured) using the graduations

AIS 1	minor
AIS 2	moderate
AIS 3	serious
AIS 4	severe
AIS 5	critical
AIS 6	maximum

This scale can be used for individual injuries as well as for total injuries of the body region and for the total injuries of the person as MAIS. But MAIS does not contain any information concerning the relevant status of the event of death. It is evident that all persons of MAIS 6 injured persons died, but also 41% of MAIS 5, 10% of MAIS 4 and even 2% of MAIS 3 injured persons die as well. A comparison of the official injury severity grades minor, serious and fatal with the AIS scale is possible (Otte-5). Fig. 3 shows a good correlation of about 80%, when

MAIS 1 = minor injured
MAIS 2-4 = seriously injured
MAIS 5/6 = most seriously/fatally injured

are defined. A 95% probability for exclusively serious and most serious injuries is possible by use of the injury severity degree MAIS 3 and above, described as **MAIS 3+** during the following.

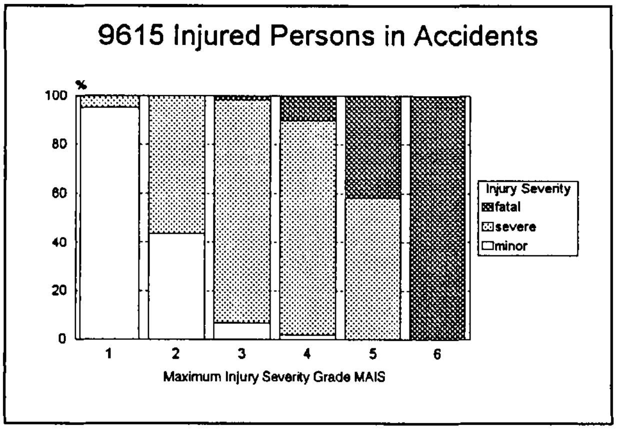


figure 3 correlation of MAIS and official injury severity grades

8,500 accidents were documentated within the framework of the Accident Investigations of Hannover during the 10 years between 1985 and 1994, these included 6,908 with car participation. 563 car passenger accidents of MAIS 3+ were registered. A total of 691 car passengers with MAIS 3+ injuries were recorded.

Fig. 4 shows the MAIS distribution of all injured car passengers, 80% suffered MAIS 1 (minor injured), only 5.5% of the involved suffered MAIS 3+ injuries, 25% of the MAIS 3+ patients died..

	total	fatalities
total	5165	70
maximum AIS		
MAIS 1	80.0%	0.0%
MAIS 2	14.5%	0.0%
MAIS 3	3.2%	3.4%
MAIS 4	0.9%	15.6%
MAIS 5	0.7%	52.5%
MAIS 6	0.7%	100.0%
MAIS 3+	5.5%	25.0%

figure 4 frequency of injury severity MAIS and percentage of fatalities for different MAIS grades

Population of the MAIS 3+ occupants

2/3 of the MAIS 3+ patients were drivers (66%), 21% were front-seat passengers and 10% were rear-seat passengers. 69% used a seat belt. 37% were up to 25 years of age, only 14% were older than 55. Which shows that high age, that leads to a reduced endurance limit, is not responsible for the severe injuries of car passengers, but rather that the corresponding accident situation causes the resulting injury. A comparison of the ages of all car passengers, including the non-injured, shows that 32% are up to 25 years of age and 12% are older than 55. But 43% of the MAIS 3+ patients had a height of more than 175 cm in relation to 30% of the MAIS 0 to 2 persons.

The accident and collision situation of the injured occupants MAIS 3+

25.5% of the cars with MAIS 3+ injured persons collided with another car, 22.9% collided with a pole and 36.8% suffered multiple collisions. When a closer look is taken at the multiple collisions in regard to the collision opponent (figure 5), car and pole collisions make up 2/3 of all collision situations of the severely injured. 56 % of the cars collided frontally and 36% laterally. This is a distribution, which is not differ from the situation of all accidents with injured people.

	n	%
total	252	100.0%
collision partner		
car	86	34.2%
truck to 7,5t	18	7.0%
truck > 7,5t	15	6.0%
pole	92	36.6%
other object	33	13.0%
two-wheeler	-	-
pedestrian	-	-
others, unknown	7	2.7%
impact area		
front	143	56.7%
side	90	35.6%
rear	5	1.8%
others	15	5.9%

figure 5 collision partners and impact areas of cars with MAIS 3+ occupants

The following collision types were established (figure 6 and figure 7). In the case of car to car collisions, the angle between the longitudinal axis of both vehicles was established and assigned to the collision points on the cars. For this reason every side of the vehicle is divided into 3 zones according to VDI (Vehicle Deformation Index - 6). The angle of the transmitted impulse was established and determined to the according impact points on the car for the representation of the collision types with poles.

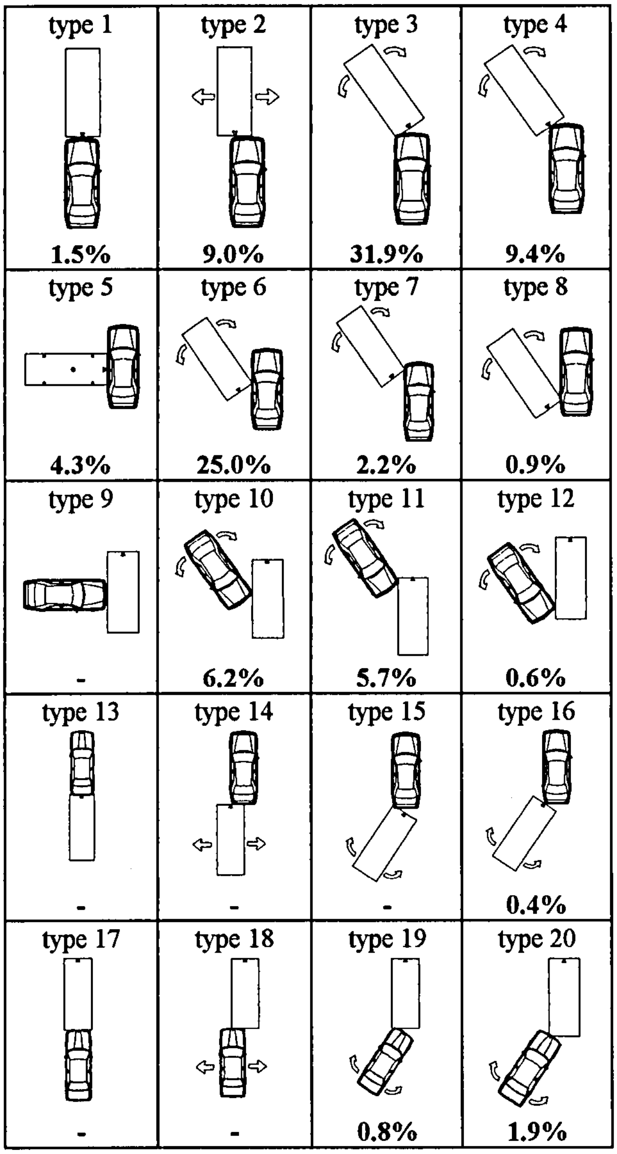


figure 6 definition of collision types in car to car accidents, the blacked car is the MAIS 3+ vehicle

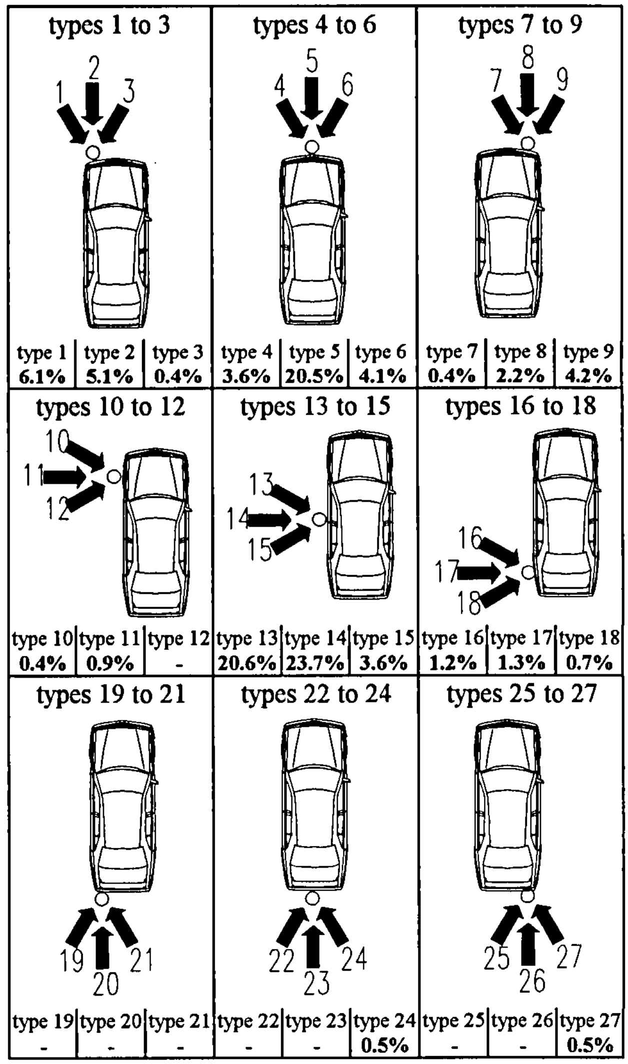
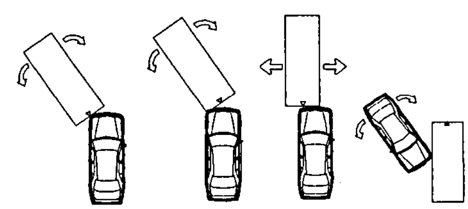
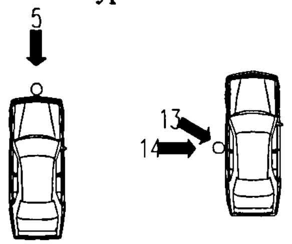


figure 7 definition of collision types in car to pole accidents

3/4 of the most seriously injured MAIS 3+ cases were assigned to frontal collisions at an oblique impact angle of the opposing car, where as in most cases the vehicle fronts partly covered each other, as well as oblique lateral impacts against the compartment area (see the following sketches)



As the main impact constellation with poles, 3 different collision types were established,



2/3 of the most seriously injured occupants were registered in these 3 main collision types. These are a pole impact against the mid of the car front as well as the rectangular and oblique impact against the lateral part of the compartment.

INJURY PATTERNS OF THE MAIS 3+ CASES

The injury cases with an injury severity grade of MAIS 3+ or higher usually suffered injuries of various body regions. 37% are regarded as so-called polytraumatised, which according to the definition, influences at least 3 different body regions, each injury being so severe that it can influence the occurance of dying (Heberer - 7). The AIS 3+ injuries were located on the head (35.1%), thorax (41.5%), abdomen (20.5%) and legs (38.9%). Figure 8 shows that patients who suffered head injuries of the severity of AIS 3+ also suffered AIS 2 and higher injuries to the thorax (55.2%), abdomen (30.8%) and legs (45.6%). 62.2% of those with serious thorax injuries of AIS 3+ also suffered head injuries of AIS 2 and above. 61.5% of those with leg injuries of AIS 3+ suffered head injuries of AIS 2+ and above. Which shows that head injuries were especially frequent among the MAIS 3+ injured. Only 20% of the passengers suffered no head injuries.

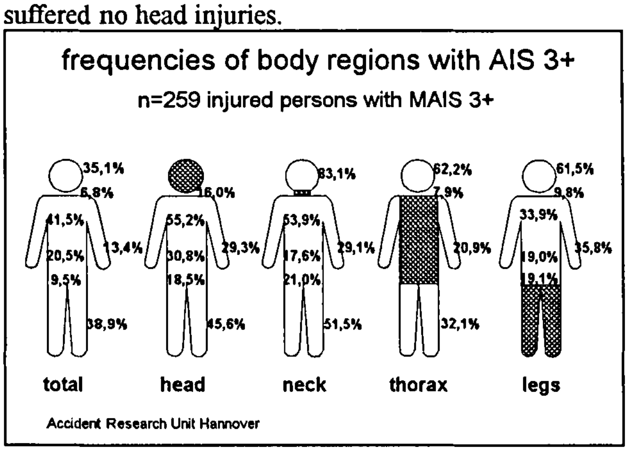


figure 8 frequency of body regions with MAIS 3+ in total (100% all AIS 3+ injuries) and for different body regions with MAIS 3+ (100% each blacked body region)

The most relevant injuries are brain traumata, which suffered 48.9% of the MAIS 3+ patients (figure 9). 21.5% occured fractures of the mid facial bones. It is generally noticed, that 1/5 of the MAIS 3+ patients suffered fractures to the spine. This pattern can be explained as bending and sharing load to the whole body and identify the body movement of the belted occupants, 21.2% of the belted MAIS 3+ patients suffered spinal injuries in frontal collisions.

		impact area	
	total	front	side
total (n)	259	149	89
skull fracture	9.6%	8.6%	10.1%
facial fracture	21.5%	30.1%	7.7%
fracture of base of skull	6.3%	6.7%	4.2%
brain injury	48.9%	44.1%	53.7%
spine fracture	19.9%	21.2%	18.1%
rib fracture (> 3 ribs)	19.0%	16.3%	21.5%
organ injury thorax	23.5%	19.4%	24.3%
intra-abdominal injury	16.7%	14.2%	17.5%
pelvic fracture	16.0%	12.0%	24.2%
organ injury pelvis	0.5%	0.3%	1.0%
closed fracture of upper leg	25.5%	32.1%	19.3%
open fracture of upper leg	4.5%	7.0%	1.1%
fracture of knee	7.1%	10.0%	1.0%
closed fracture of lower leg	9.4%	11.4%	7.2%
open fracture of lower leg	6.6%	8.1%	4.7%
fracture of foot or ankle joint	13.4%	15.6%	8.0%
fracture of upper extremities	25.5%	29.8%	15.8%
fracture of shoulder	9.0%	7.4%	11.9%

figure 9 frequencies of common injuries of MAIS 3+ occupants

In lateral collisions severe brain injuries as well as serial fractures of the ribs, multiple thoracic lesions and pelvis fractures occur very often. In frontal collisions thigh fractures could be established in 32% of the MAIS 3+ patients. This explains itself by the high deformation and load transmission about the dashboard.

BASIC CIRCUMSTANCES OF THE ACCIDENT

Level of accident severity

The data reveals that the passengers of cars with MAIS 3+ injuries are usually prone to extence accident severity (figure 10). The change of speed due to the collision „delta-v“ is an dominant accident severity parameter.

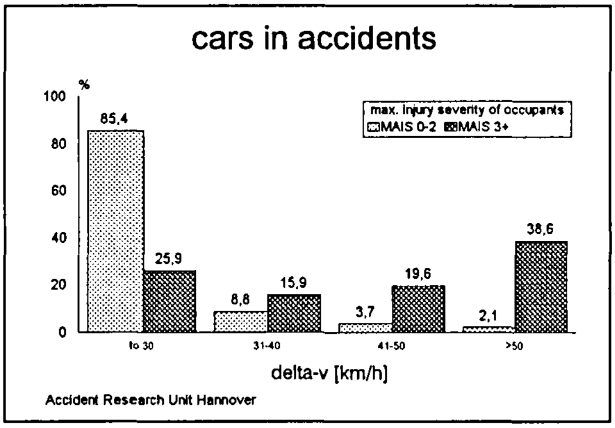


figure 10 distribution of delta-v for cars MAIS 3+ occupants in relation with MAIS 1/2 occupants

Where as accidents which lead to minor injuries have delta-v values of 85.4% up to 30 km/h, raising only by 2.1% at speeds above 50 km/h, only 25.9% of the car passengers with MAIS 3+ injuries had delta-v values of up to 30 km/h, but 38.6% even recorded values above 50 km/h.

An influence of the vehicle weight on the resulting injury severity could not be established. Roughly the same distribution of vehicle crashweight was established for cars with MAIS 3+ injured as for cars with minor injured passengers (figure 11). But as the impulse examination interrelates the mass and delta-v, it is significantly proved that with higher delta-v values the occurrence for resulting injury severity grade is higher too.

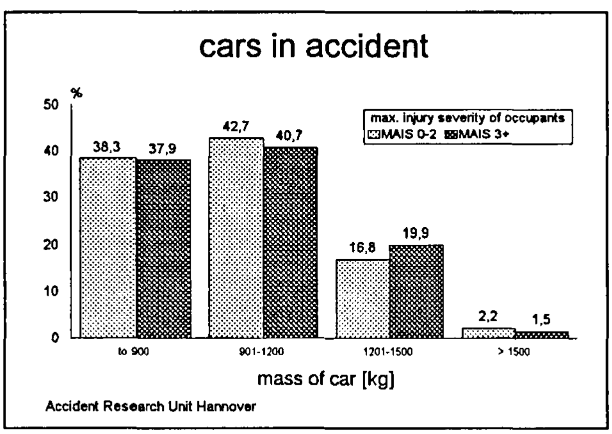


figure 11 mass classes of cars with MAIS 3+ occupants in relation to those with MAIS 0-2 occupants

The mass proportions of the vehicles colliding together were also analysed. It could be established, that for 77.2% of the MAIS 3+ vehicles a mass relation factor above 1 was often calculated (figure 12)

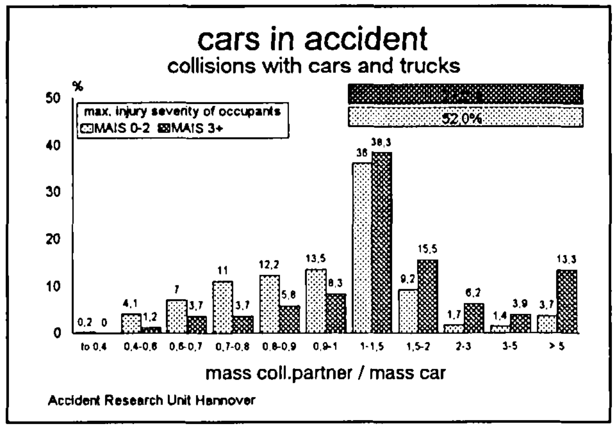


figure 12 mass ration (mass collision partners / mass of car with MAIS) for occupants with MAIS 3+ in relation to MAIS 0-2

An analysis of the age of the vehicle as a possible influential parameter only lead to a slight difference in the comparison of severe and minor injuries. Where as 59.5% of the cars with MAIS 3+ injured were more than 6 years old, 53.4% of the cars with minor injured were of the same age (figure 13).

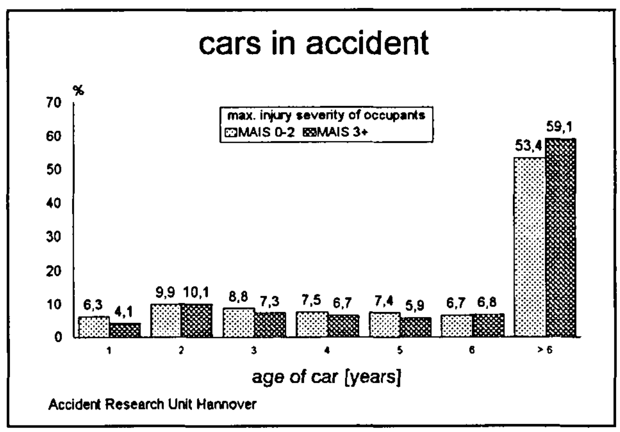


figure 13 age of cars with MAIS 3+ occupants in relation to cars with MAIS 0-2 occupants

Deformation characteristics

The deformation characteristics of the most frequent collision constellations of the MAIS 3+ injured can be reconstructed by the matrix system developed by the author. This shows the vehicle, viewed from above and divided into approximately 200 fields of equal size (Otte - 4). The accident documentation records the deformated zones by true-to-scale marking of the corresponding fields and this reproduction of the deformation pattern of every accident is stored in a computer. This means that it is possible to reproduce the deformation pattern of every case and at free choice by means of the accumulated addition of all the damaged zones as procentual frequency of the damaged zones.

The figures 14 and 15 reproduce the deformation patterns of car to car collisions involving an oblique frontal impact as well as car to pole collisions to the mid of the front as well as involving the passenger compartment in lateral collisions. It can be seen that in frontal collisions the deformation often reaches as far as the passenger compartment. 69% of the cars with MAIS 3+ injured were damaged by an intrusion of the compartment in frontal collisions, 86% in lateral collisions. 1/3 of the lateral impacts lead to deformations of more than 40% of the vehicle breath.

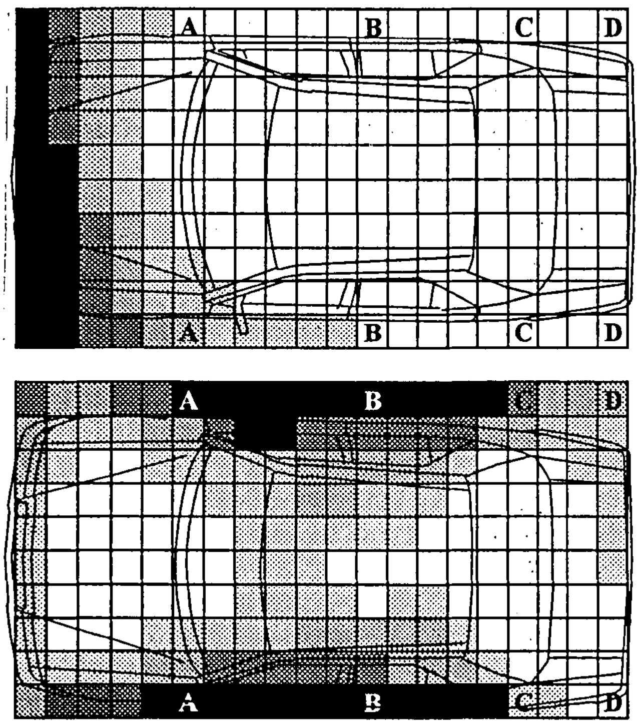


figure 14 **<u>car to car accidents</u>, upper matrix collision type 3 and lower matrix collision type 6 - reference figure 6**

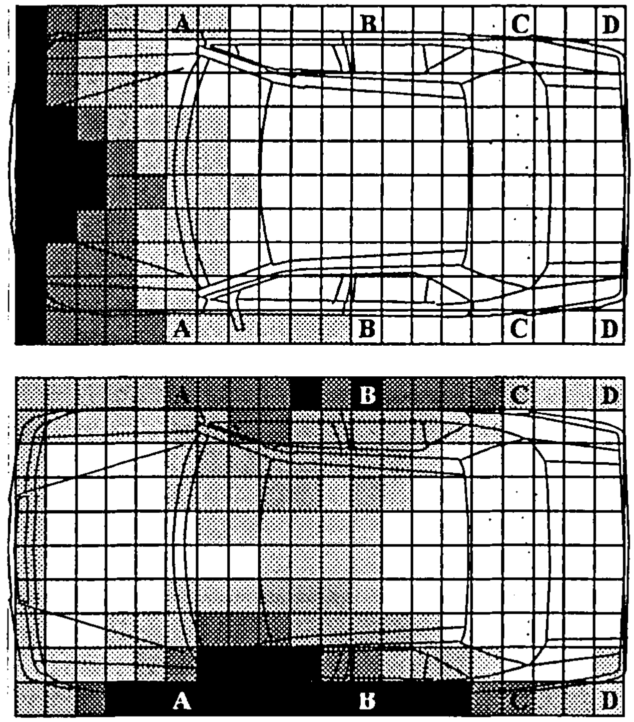


figure 15 **<u>car to pole accidents</u>, upper matrix collision type 5 and lower matrix collision types 13 and 14 - reference figure 7**

CONCLUSIONS

The detailed analysis of the car accidents, by which a passenger was severely injured with severity grades MAIS 3 and higher (MAIS 3+) shows that special collision situations were always in effect.
These are for instance

- Frontal collisions involving an oblique impact of the opposing car as well as a mid frontal impact with a pole

as well as

- the oblique lateral impact of a car as well as the rectangular and oblique impact of a pole against the compartment side

Deep intrusions as well as high delta-v values are dominant for the accident conditions. MAIS 3+ passengers are often polytraumatised and suffer serious injuries of the head, thorax and legs. The head is almost always injured and this is usually responsible for the resulting trauma consequences. The half of MAIS 3+ patients suffered brain traumata, a quarter died. There is no significant influence concerning the age of persons or vehicles, but 43% of the MAIS 3+ occupants had a body height of more than 175 cm, compared to 30% only in the group of persons with MAIS 0 to 2.

This means that measures towards the reduction or avoidance of serious injuries caused to car passengers in the course of accidents, which make up 5.5% of all injured car passengers, could be taken.
The detailed analysis of the circumstances of the accidents shows results for which the following measures should be taken into consideration:

1 Optimized front-impact test conditions

The necessity of an offset-impact is necessary to reduce the amount of severely injured car passengers. The circumstances of oblique collisions between cars by which the vehicle fronts partly overlap, a collision type which often takes place in everyday traffic, should be taken more into consideration during crash-tests. This only seems possible when either 2 vehicles are moved or when the barrier is set into motion. With a fixed barrier of offset simulation the lateral component of the impulse will not be considered in reality and the relative movement of the occupants will not be reproduced realistically. The seating position of the dummy should be foremost position.

- A crash test car against a pole, which collides mid-frontally, is also necessary. Oblique impulse angles are not necessary in this case.

2 Optimized lateral impact test conditions

- A barrier, moved under an oblique direction against the lateral compartment, should be used for crash-test conditions of a simulated collision of a car against the side of another car, where as the front corner of one car impacts the side of the other car first, which is not a crabbed impact of the whole front against the side as described in the US-test.
- The lower edge of the barrier should be positioned as high as possible in both the EC as well as US test, as the real accident situation shows that the impact height always exceeds the side sill, thus causing a deep intrusion of the passenger door.
- A single test condition for the simulation of lateral car collisions is not sufficient. More attention should be paid to the frequency and severity of object impacts, especially those involving poles.
- To enable accident conform test conditions of an impact with an object, the pole should impact the car between the A and B-pillar of the vehicle directly in front of the pelvis of the passenger. The impact direction should incorporate an oblique component from the front.
- The height of a pole-barrier can be as high as the vehicle from the door-base to the roof.

3 Optimized vehicle safety for frontal collisions

The present safety fittings of the vehicle such as the seat belt, airbag and construction measures of the car seem to be sufficient and are demonstrated in the detailed study. A few optimizations seem sensible
- in the foot region to avoid foot fractures
- in the compartment structure to avoid intrusions of the passenger compartment

4 Optimized vehicle safety for lateral collisions

- Safety precautions for lateral collisions can be supplemented by implementing various airbag systems fitted to the seat (Volvo) or the B-pole or along the upper door frame (BMW). The following proposal should lead to an improvement of vehicle safety in lateral collisions.
- To avoid deep intrusions caused by pole-impacts the lower door frame respectively the side sill and the roof edge should be reinforced to avoid an intrusion of the passenger compartment as well as a tilting up of the vehicle leading to a considerable intrusion of the upper roof area as well.
- Passenger impacts in the region of the compartment sides should be made less severe by installations of suitable airbag systems. These should be systems which are installed in the post/windscreen area to protect the head and thorax, as well as on the door/seat to protect the thorax and pelvis. A combination of both should be strived for.

5 Summary

No new cognitions beyond todays knowledge level can be derived from these facts. Many of the demands for measures of injury reduction mentioned have already been proposed in earlier publications (e.g. Otte-8,9), but they gain more importance by the extensive detailed representation and analysis of the injury and collision situation of the severly injured MAIS 3+ found in todays traffic scene. It can be suggested once again that the two aims

stable passenger compartment
energy absorption

although their safety strategy aspect may differ, should be used mutually in the concept.

The aspects of compatibility should be further pursued, as the study revealed that many of the MAIS 3+ injured were hit by vehicles of larger mass.
Softer front structures of the heavier impacting vehicle could lead to better EES relation.

On the whole, an intrusion must not be regarded as negative, as a reduction of kinetic energy by means of deformation crumpling is needed to reduce impact. But lateral impacts do not offer this facility, which means that injury sources can be found in todays vehicles, caused by relatively uncontrolled characteristics of the deformation leading to sharp-edged parts, as can be derived from the documented accidents. A reduction of injuries could be achieved by a behaviour leading to a defined, wide area surface pressure. The offset and lateral impact test conditions (ETSC - 10) which are to become obligation Europe-wide from 1998 onwards fulfil a few of these demands.

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A MATHEMATICAL HYBRID MODEL FOR EVALUATING VEHICLE PERFORMANCE IN CAR-TO-CAR SIDE IMPACTS

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ABSTRACT

A hybrid model consisting of several parts: a baseline vehicle, a EUROSID-1 dummy and a US-SID dummy, was developed using the finite element software RADIOSS. The vehicle model was developed using truss, beam, spring and beam type spring elements. Where necessary, the vehicle model was complemented with shell elements. The EUROSID-1 and US-SID dummies were one dimensional lumped mass spring damper models and consisted of spine, ribs, and pelvis. The model will be used to evaluate effects of vehicle modifications on injury risk predicted by the dummies.

The model was validated by means of mechanical tests. Generally good agreement was obtained between predictions of the model and the results from the mechanical tests. The dummy models were validated by means of pendulum tests. The baseline vehicle model with the dummies were validated by means of crash tests according to both the American crash test procedure (FMVSS 214) and the proposed European side impact test procedure.

The applicability of the hybrid approach for vehicle and occupant modelling has been demonstrated. The hybrid model provides a quick and economical analysis of a large number of design changes. The advantage of this model is that it can be used in the early design stages of passenger vehicles.

INTRODUCTION

After frontal impacts, side impacts are the second most common type of collision to cause serious or fatal injuries (AIS 3-6). The relative frequency of side impacts is 20% of all impacts and accounts for approximately 50% of the total number of accidents causing serious or fatal injuries (Harms et al., 1987; Otte, 1993). For Volvo vehicles approximately 25% of all serious-to-fatal injuries have been established as side impact accidents (Lundell et al., 1995).

Depending on the investigation, the body region rated as the most vulnerable in side impacts varies. Rouhana et al. (1985) and Harms et al.(1987) studied serious to fatal injuries and rated the chest as the most vulnerable region followed by the abdomen and then the head. In another study it was found that the abdomen was slightly more vulnerable than the chest, and the head less vulnerable than the chest (Jones, 1982).

The severity of the accident expressed in change of velocity (Δv), is one of the important parameters for deciding whether the occupant will sustain injuries or not. The crash severity for serious injuries on average varies between 27 km/h and 32 km/h, and for fatal injuries Δv is often more than 50 km/h (Harteman et al., 1976; Jones, 1982). It has also been found that there is a 10% probability of serious injuries for near-side occupants in side impacts at a Δv of 26 km/h (Mills and Hobbs, 1984).

There are at present two side impact crash test procedures for evaluating and improving the level of safety of passenger vehicles: the American (FMVSS 214) (NHTSA, 1990) and the proposed European (ECE, 1993) (Figure 1). The American test procedure simulates a road intersection crash in which the struck vehicle - the vehicle to be tested for compliance - is traveling at 24 km/h (15 mph) and the striking vehicle is moving at 48 km/h (30 mph). This is accomplished with a crabbed (27^0) mobile deformable barrier impacting a stationary vehicle at 54 km/h (33.5 mph). The European side impact test proposal prescribes a 50 km/h side impact test at a right angle with a mobile barrier of 950 kg mass.

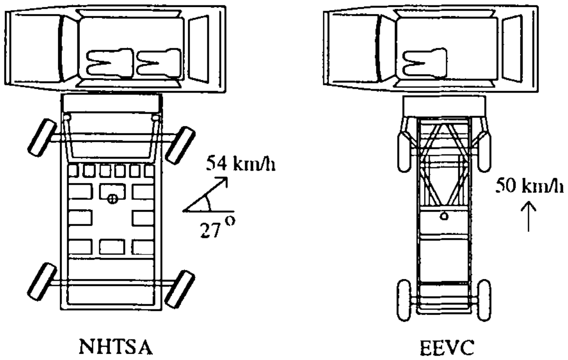


Figure 1. NHTSA and EEVC side impact test procedures.

In these tests a human surrogate, a crash test dummy is used to evaluate the risk of sustaining injuries. There are at present three side impact dummies available; the US-SID, EUROSID-1 and BIOSID. The first one is prescribed in the US regulation and the second one in the proposed European regulation.

An injury criterion is used to estimate the risk of injury. There are two injury criteria for the chest at present; the viscous criterion (VC) and the thoracic trauma index (TTI). The VC takes the visco-elastic response of the chest into account (Lau and Viano, 1986). The VC is the instantaneous product of chest deformation speed (V) and the relative compression (C). VC is included in the proposed European side impact requirements and the injury criterion level is VC≤1m/s. The TTI is the average of the maximum spine acceleration and the impacted rib acceleration expressed in g's. The American requirements use the TTI and the injury criterion level is TTI≤85 g for four door cars and TTI≤90 g for two door cars (NHTSA, 1990).

Carrying out crash tests, however, is expensive and therefore mathematical models are used extensively to evaluate vehicle crash performance. Lumped parameter models are ideal for this purpose since it is possible to carry out hundreds of runs at little expense, and investigating the effects of varying combinations of parameters. Lumped parameter models for predicting vehicle frontal crash response, using static crush data of front end components, have been in use for several years (Kamal, 1970; Prasad and Padgaonkar, 1981). These models are one-dimensional, but have proven to be useful for studying the effect of mass and stiffness changes of the various components. In frontal collisions the behavior of the structure is not influenced by the occupant. In side impacts, however, it has been shown that the dynamic response of the side structure is affected by the presence of an occupant in close proximity to the side structure (Monk et al., 1980). Hence it is necessary to simultaneously model the occupant and the structure in a side impact simulation. A number of one and two-dimensional side impact models of vehicle and structure as well as the occupant have been developed (Trella and Kanianthra, 1987; Kanianthra and Trella, 1989; Richter, 1989; Dieu and Riss, 1994). In these models, however, the effective masses of the vehicle side structures and dummy components must be established. To avoid having to estimate effective masses and still to gain the ability to predict vehicle and dummy's out-of-plane twisting/rotation motions, three dimensional models of the vehicle structure and dummy have been used. In a number of models, the possibility of three-dimensional car-to-car side impact simulations where the structure and the dummy occupant have been modeled simultaneously have been shown (Padgaonkar and Prasad, 1979; Monk et al., 1980; Padgaonkar and Prasad, 1982) In these models the occupant was modeled by four segments, the lower torso, upper torso, neck and head. The stiffness of the side structure of the impacted vehicle and the front end of the impacting vehicle were modeled. A lumped mass three dimensional lumped mass model of a baseline vehicle, a US-SID dummy and a NHTSA side impact barrier has been developed and validated by means of actual crash tests (Low et al., 1991). A similar lumped mass three-dimensional model of a baseline vehicle including a EUROSID-1 dummy and the EEVC side impact barrier has also been developed and validated by means of crash tests (de Coo et al., 1991). These models have been used to carry out various trend studies through parametric variations in the models.

In modelling a car-to-car side impact, it is vital to have an accurate description of the impacted side. It has been shown that serious or fatal injuries are much more common in impacts with door intrusion than in impacts without intrusion (Danner, 1977) and that doors that have remained upright during the crash have been able to offer the occupant much better levels of protection (Hobbs, 1989). The profile of the intruding door is controlled, to a great extent, by the characteristics of the "B-pillar" and the "sill". The B-pillar can, in general, be considered as a beam supported at its top and bottom. When loaded, it bends with the greatest deflection being near its center, where the bending moment is greatest (Hobbs, 1989). The extent to which the door follows the motion of the B-pillar is determined by the degree of interaction between the bottom of the door and the sill. If the sill prevents the bottom of the door from over-riding it, the degree of door tilt increases (Hobbs, 1989).

Unfortunately the lumped mass models provide a poor representation of parts of the structure in which dynamic effects are significant, such as the tilting of a door as it intrudes inwards. A detailed finite element model on the other hand, would provide an accurate representation of the impacted side of the vehicle. Areas directly involved in the deformation have to be meshed in fine detail, while other parts, where an accurate stiffness is less important to the analysis, can be meshed more coarse. However, the finite element models are time consuming to prepare and to execute. Therefore for research purposes, and in the early design stages of a vehicle, it is not practicable to run full finite element models, with all its complications more than a very few times. What appears to be an ideal solution is a compromise between the simplicity of the lumped parameter model and the accuracy of the finite element model, a hybrid model.

The objective of this paper is to develop a hybrid side impact model that combines the simplicity of the lumped parameter model and the accuracy of a finite element model. Parts of the model is an accurate representation of the real structure, and parts of the model is a simple representation of the real structure. The model takes into account the very complex deformations and interactions that occur inside the car structure, between the barrier and the occupant. In addition, the objective is to demonstrate the applicability of a hybrid approach in vehicle and occupant modelling of a side-impact collision.

For the development of the model the following criteria applied:

- The model must be an efficient tool for parametric studies,
- It must be possible to estimate the injury reducing benefits of padding,
- It must also be possible to estimate dummy response to structural modifications.

METHOD

The explicit finite element code RADIOSS-CRASH (Mecalog, 1994) was chosen. The important features that were used were lumped mass, discrete elements and elements where the deformation modes: tension, bending, shear and torsion were described individually.

RADIOSS solves the momentum equation

$$\frac{\partial \sigma_{ij}}{\partial x_j} + \rho b_i = \frac{\partial v_i}{\partial t}, \qquad (1.)$$

where σ_{ij} is the Cauchy stress tensor, b_i is the volume force (body force), x_j is the displacement vector, v_i is the velocity of displacement and ρ is the density. The boundary conditions on the outer surface S are,

$$\sigma_{ij} n_j = p_i \text{ on external surface } S_1, \textit{ and} \qquad (2.)$$

$$v_i = \bar{v}_i \text{ on external surface } S_2. \qquad (3.)$$

These boundary conditions and the initial conditions $x(0)$ and $v(0)$ fully describe the dynamic continuum system under consideration. When the finite element method is used to solve the momentum equation, the equation is formulated as a system of ordinary differential equations,

$$M\left\{\frac{dv}{dt}\right\} = \left\{F^{ext}\right\} - \left\{F^{int}\right\} + \left\{F^{bod}\right\} \qquad (4.)$$

where M is the mass matrix, v is the velocity vector, F^{int} is the internal force vector (stress divergence vector), F^{ext} is the external load vector and F^{bod} is the body forces vector. If the lumped approach is used, the mass matrix M is diagonal and the explicit procedure becomes computationally efficient.

The equation is solved in the time domain by explicit integration using the central difference method. For a time step Δt,

$$v_{t+\Delta t/2} = v_{t-\Delta t/2} + \frac{dv}{dt}\Delta t \text{ and} \qquad (5.)$$

$$x_{t+\Delta t} = x_t + v_{t+\Delta t/2}\Delta t. \qquad (6.)$$

The integration scheme is conditionally stable. The stability condition or COURANT condition is

$$\Delta t \leq \frac{\Delta l}{c} \qquad (7.)$$

where Δl is characteristic element length and c is sound speed.

In a passenger vehicle, the body components are thin or semi-thin walled beams to which the classical collapse theory does not apply. The semi-thin walled beams buckle within the elastic or plastic range and this has a significant effect on their maximum strength and energy absorption capacity. The classical collapse theory assumes constant collapse moments during rotation. In a passenger vehicle there are a number of structural components which make a significant contribution to the behavior of a whole vehicle during impact. The collapse characteristics of these components can be measured quasi-statically. Incorporating data from these quasi-static tests in RADIOSS can be done using the beam type spring element. In the beam type spring element the deformation properties can be defined in the individual deformation modes tension, torsion, shear, and flexion (Figure 2). For each deformation mode, data is given in a tabular form. Six tables are needed to define the deformation properties of one beam type spring element. The properties of the element can be totally elastic, non linear elastic or non linear elastoplastic.

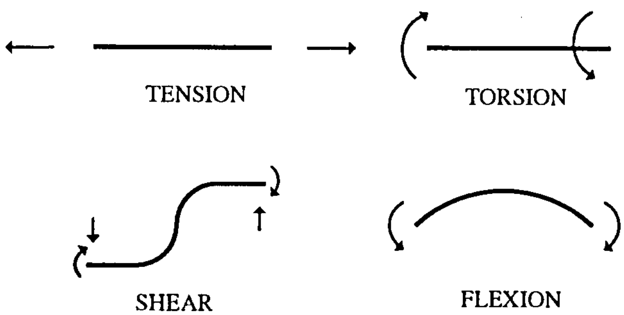


Figure 2. Deformation modes of the beam type spring element.

BASELINE VEHICLE MODEL

A combination of the lumped mass and finite element approach was realized using RADIOSS. The structure was developed using beam, truss, beam-type spring, shell, spring, and point mass elements. The model contained 191 two-node truss elements, 264 two-node beam elements, 1408 shell elements, 48 beam type spring, 12 spring, and 12 point mass elements.

The beam elements were used to model the structural members of the vehicle that were not directly involved in the deformation. Cross sectional data for these structural components were computed and used in the hybrid model.

The sheet metal parts of the non struck side of the vehicle were modeled using truss elements. Each sheet metal area was modeled by 6 truss elements. An equivalent truss cross-sectional area was computed for each area.

On the struck side of the vehicle beam type spring elements were used to obtain an accurate description of the structural parts directly involved in the deformation. Element characteristics were derived from quasi-static bending, compression, and torsion tests where the components were loaded until failure. These element characteristics were incorporated in the hybrid model by means of the beam type spring element with non linear elastoplastic element properties.

Shell elements were used to simulate the interaction between the pillars, sill, and doors in an impact. In a side impact, the impacting object transmits force to the vehicle through contact with the B-pillar and doors. The doors transmit force from the impacting object to the vehicle when the bottom of the door over-rides the sill. These interactions were simulated in the hybrid model by complementing the pillars and sill with shell elements and adding a door section. The nodes of each pillar and shell element were joined to a beam-type spring element node to form a rigid body (Figures 3 and 4). The door section was modeled as one simple open rectangular box section (Figure 5) using shell elements. In a side impact force is also transmitted from the doors to the A- and B-pillar via the hinges. Therefore in the hybrid model the box section was connected to the A- and B-pillar at the positions where the hinges were located.

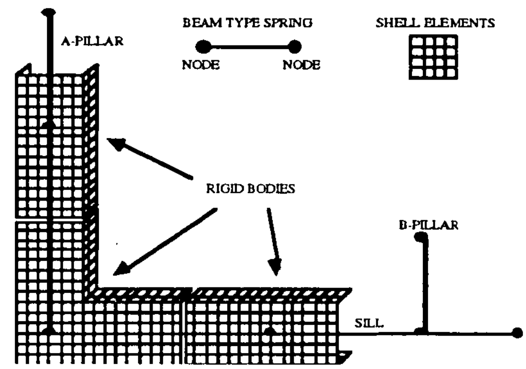


Figure 3. Schematic A-, B-pillar and sill model.

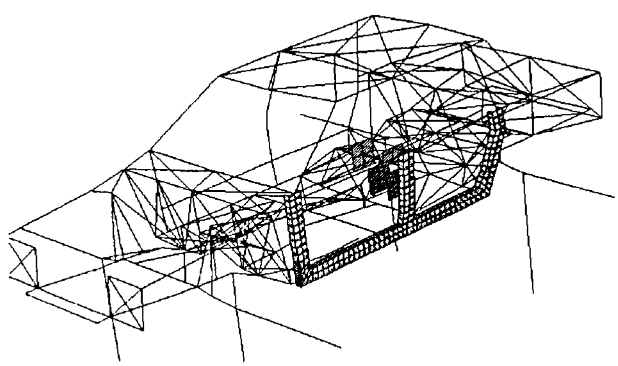


Figure 4. Baseline vehicle without door section.

Non linear springs were used to model tire friction in the impacted direction (y-dir). The springs were connected to the wheel center and to the inertial space. The tire friction force used for the rear wheels was 3.5 kN, and for the front wheels 4.5 kN (Rundkvist, 1974). Force due to the rear tire friction was transferred through the friction springs at the wheel center to the vehicle structure by three beam elements, while the force from the frontal tire friction was transferred to the structure by two beam elements.

The baseline vehicle model was completed by adding point masses for the regions of the car that were not modeled; engine, wheels, spare tire, gas tank, rear axle and seats. In a side impact the engine, gearbox, the differential, and the chassis components of the car undergo very little deformation (Tilakasiri and Dubois, 1990). The most important requirements as far as these components are concerned are the mass distribution. Therefore, these elements were modeled as point mass elements. In addition, in order for the baseline vehicle to have correct total mass and mass moment of inertia, 6 additional correction masses were added.

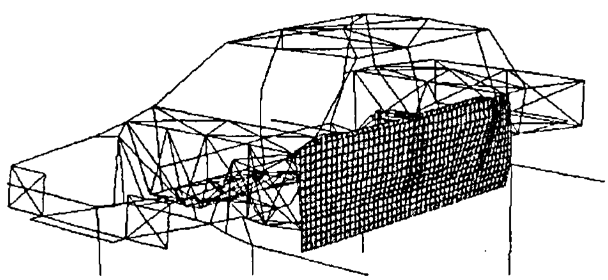


Figure 5. Baseline vehicle with door section.

BARRIER AND DUMMY MODELS

The NHTSA deformable barrier represents a typical American passenger vehicle with a simulated stiffness for the front end. The barrier was modeled using 2952 brick elements and 920 shell elements (Figure 6). The main deformable body of the mechanical barrier consists of a block of aluminum honeycomb represented in the model by the brick elements. The sheet of aluminum that covers the outside of the main barrier body was modeled by shell elements. A block of stiffer honeycomb, similarly covered with aluminum, at the lower part of the barrier represents the bumper of the vehicle. This was also modeled by brick and shell elements.

Component data for the barrier were obtained by both static and dynamic tests. The model was impacted against a rigid wall. The deceleration vs. time and force vs. displacement curves obtained were similar to those obtained during a wall-to-barrier impact test (Figure 7).

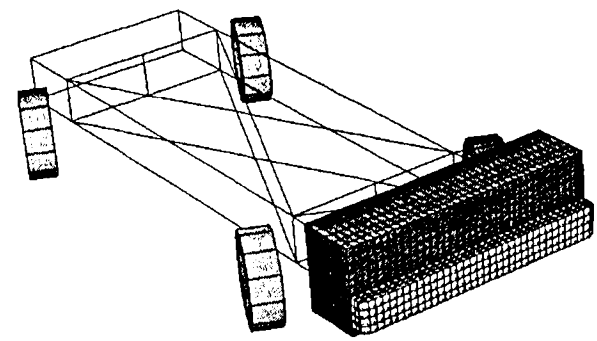


Figure 6. NHTSA side impact barrier model.

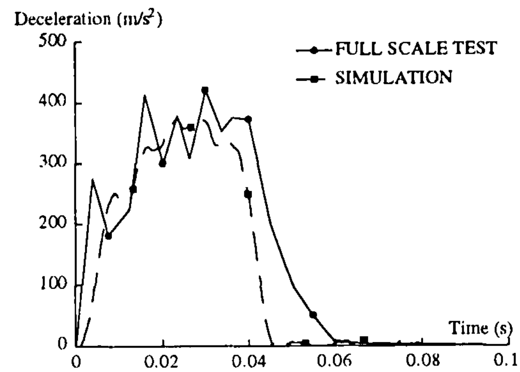


Figure 7. NHTSA barrier deceleration in a rigid wall impact test at 25 mph (40 km/h).

The moving EEVC barrier represents a typical European passenger vehicle with a simulated stiffness for the front end. The barrier consists of six composite foam blocks and two aluminum plates that cover the foam blocks. The barrier was modeled using spring and shell elements. The foam blocks and aluminum plates deform during contact with the side of a target vehicle. The compressive and shear stiffnesses of the mechanical barrier were modeled separately. The compressive stiffness was modeled by 527 springs at 90 degrees angle to the barrier face, and the shear stiffness was modeled by 340 springs at 45 degrees angle to the barrier face (Figure 8). The aluminum plates were modeled by 450 shell elements.

The force deflection characteristics of the foam blocks were obtained by both dynamic and static compression and shear tests. The barrier model was impacted against a rigid wall at 25.5 km/h. The deceleration vs. time and force vs. displacement curves obtained were similar to those obtained during a rigid wall-to-barrier impact test (Figure 9).

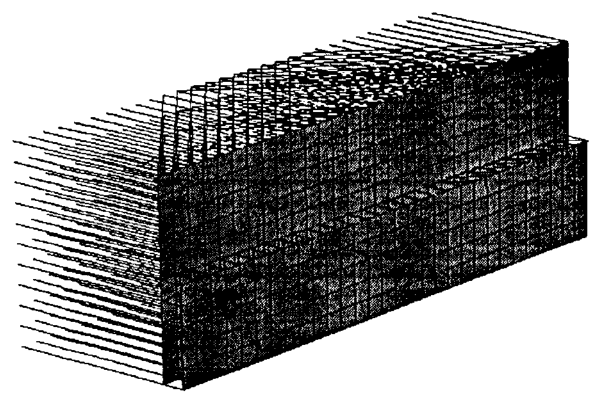


Figure 8. EEVC foam side impact barrier model.

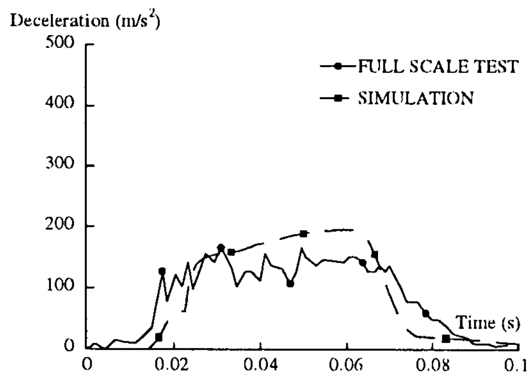


Figure 9. EEVC barrier deceleration in a rigid wall impact test at 16 mph (26 km/h).

A one dimensional model of the US-SID dummy was developed and validated (Figure 10). The mechanical US-SID consists of a head, neck, thorax, pelvis, legs, and feet. The US-SID thorax consists of five ribs and a spine. The thorax includes the mass of the arm and shoulder. The ribs connect to the spine through a shock absorber assembly. The thorax connects to the pelvis through a rubber spine. The pelvis is a steel skeleton surrounded by foam. The US-SID dummy was modeled by two body parts, the thorax and pelvis. The head, neck, arms, and legs were not included in the model since these body parts were considered to have a minor influence on dummy response. In the model the five ribs of the US-SID dummy were represented by one mass, connected to the spine through a spring damper system. There was no connection between the spine and pelvis. The pelvis was modeled as one mass and the foam was represented by a spring. The component data used in the model were obtained from an in-house measurement program of the US-SID dummy.

The US-SID model was validated by means of pendulum tests: A 23.4 kg pendulum impacted the dummy at the chest and pelvis level at 6.7 m/s. A pendulum impact velocity of 6.7 m/s was chosen since it was estimated that the energy transfer from the pendulum to the dummy would be approximately the same as from the intruding structure to the occupant in the side impact simulations. Results from the simulations were compared with results from the mechanical calibration tests. The predictions of the US-SID model was in good agreement with the results from the mechanical pendulum tests for the rib, spine and pelvis acceleration (Figure 11). Consequently, the US-SID model was considered to be able to predict the injury measures TTI and pelvis acceleration.

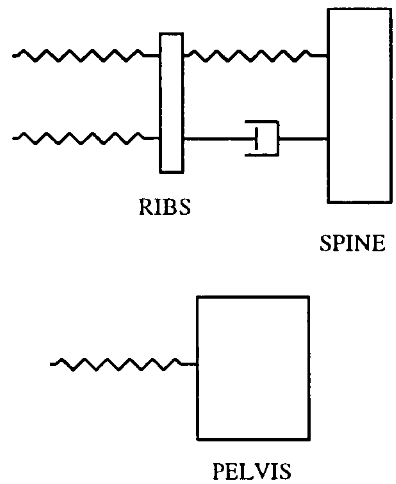


Figure 10. One dimensional US-SID thorax and pelvis model.

A one dimensional lumped mass spring damper model of the EUROSID-1 dummy was developed and validated (Figure 12). The mechanical EUROSID-1 dummy consists of head, neck, arms, thorax, abdomen, pelvis, legs, and feet. The thorax consists of three rib modules. A spring and damper system runs across each rib module. The ribs connect to the spine and the thorax connects to the pelvis through a rubber spine. The pelvis consists of a steel and plastic skeleton surrounded by foam.

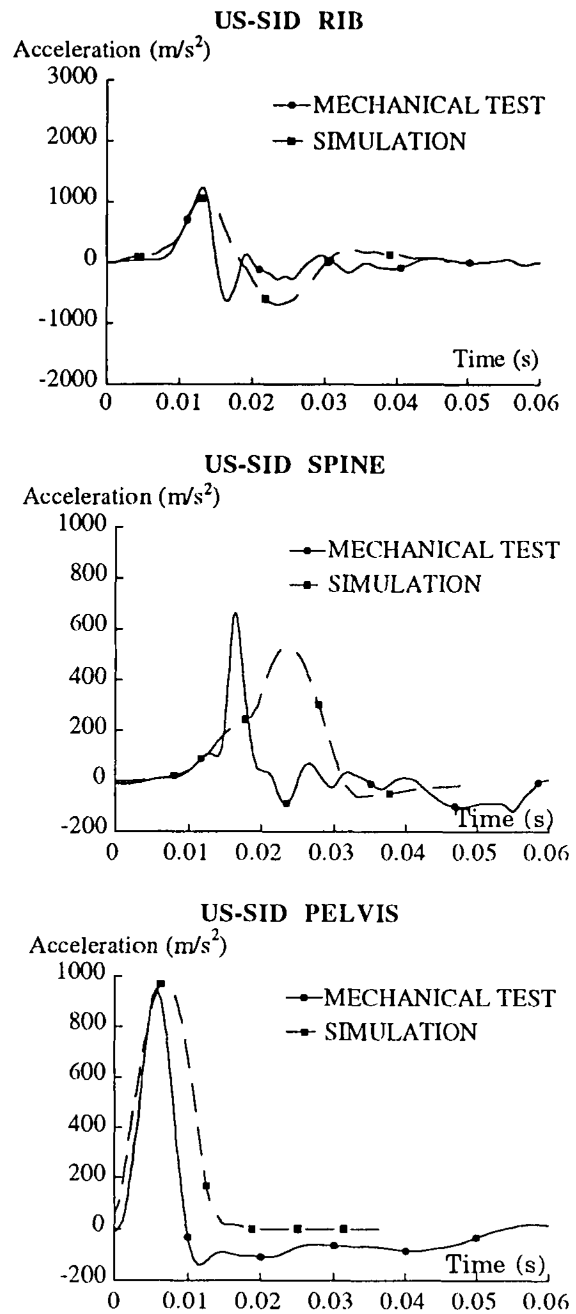


Figure 11. US-SID thorax and pelvis response in pendulum impact tests at 6.7 m/s.

The EUROSID-1 was modeled by two body parts: thorax and pelvis. Head, neck, arms, abdomen, and, legs were not included in the model since these body parts were considered to have a negligible influence on dummy response. The thorax was represented by three ribs that were connected to the spine through a spring damper system.

No connection between the spine and pelvis was included in the model. The pelvis was modeled as one mass and the foam was represented by a spring. The component data for the EUROSID-1 was obtained from a report by de Coo et al. (1990).

The dummy model was validated by means of pendulum tests. Pendulum simulation results at an impact speed of 6.7 m/s with the one-dimensional dummy model were compared with results from corresponding test with the mechanical EUROSID-1. The predictions of the model were in good agreement with the results from the mechanical pendulum tests for the rib acceleration, chest deflection, and pelvis accelerations. The shape of the spine acceleration differed somewhat between the model and the mechanical test but the peak spine acceleration values were in good agreement with one another. The peak spine acceleration is one of the components used to compute TTI. Consequently, the EUROSID-1 model was considered to be able to predict the injury measures TTI, chest deflection, chest VC, and pelvis acceleration (Figure 13).

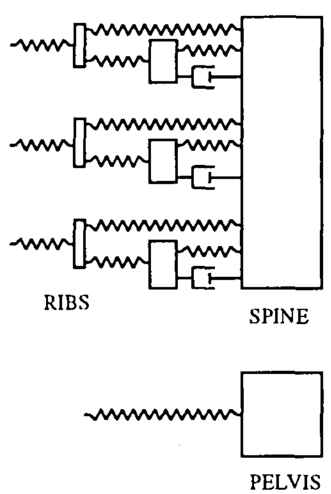
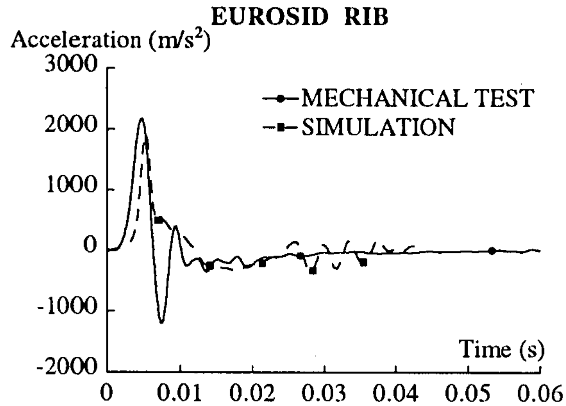
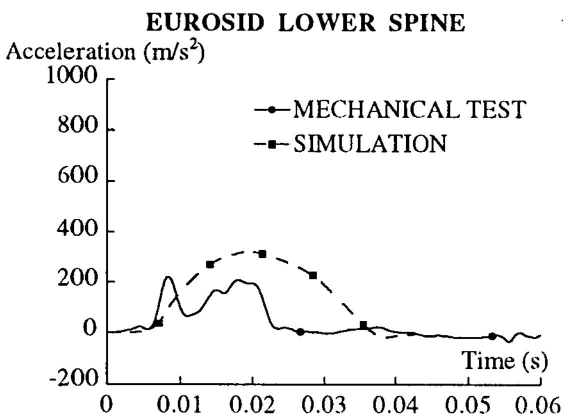
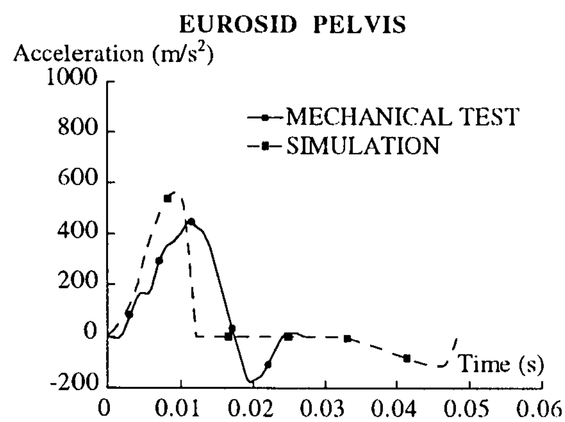


Figure 12. One dimensional EUROSID-1 thorax and pelvis model.







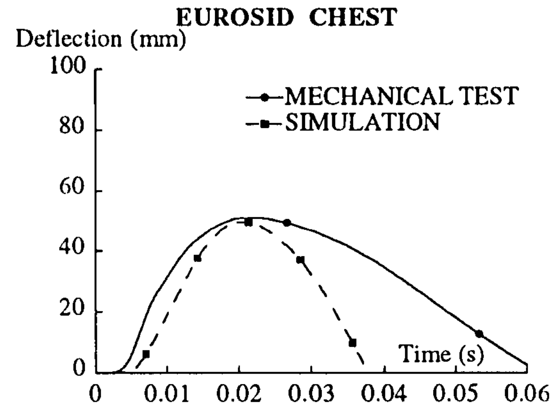


Figure 13. EUROSID-1 thorax and pelvis response in pendulum impact test at 6.7 m/s.

The dummy models were included in the baseline vehicle model. In the model the vehicle interior of was represented by a spring. Based on mechanical tests, the stiffness of the vehicle interior was estimated at 300 kN/m. The barrier, vehicle and dummy model was validated against crash tests. The tests were conducted according to the NHTSA and EEVC test procedures. The intrusion velocity of B-pillar, the velocity of the MDB, the velocity of the non impacted side of the vehicle and dummy injury measures were then compared.

RESULTS

According to the NHTSA and EEVC side impact test procedures, there was in general good correspondence between barrier-, vehicle-, and dummy results, measured in the crash tests and predicted by the model (Figures 14 and 15). The velocity of the B-pillar at the chest level in the simulation diverged from the measured velocity of the B-pillar at the chest level after about 50 ms of the crash sequence. At the chest level, the velocity of the B-pillar rose more rapid in the simulation than in the crash test. In both the simulation and the crash test, the B-pillar velocity at the pelvis level after 10 ms was approximately as high as the velocity of the barrier. The B-pillar velocity at the chest level was lower than the velocity of the barrier.

For the dummy results, in the test according to the NHTSA procedure, the shape of the predicted US-SID rib acceleration differed from the measured US-SID rib acceleration. However, the predicted peak value was in good agreement with the measured value. For the US-SID spine, the predicted peak acceleration occurred prior to the measured peak spine acceleration.

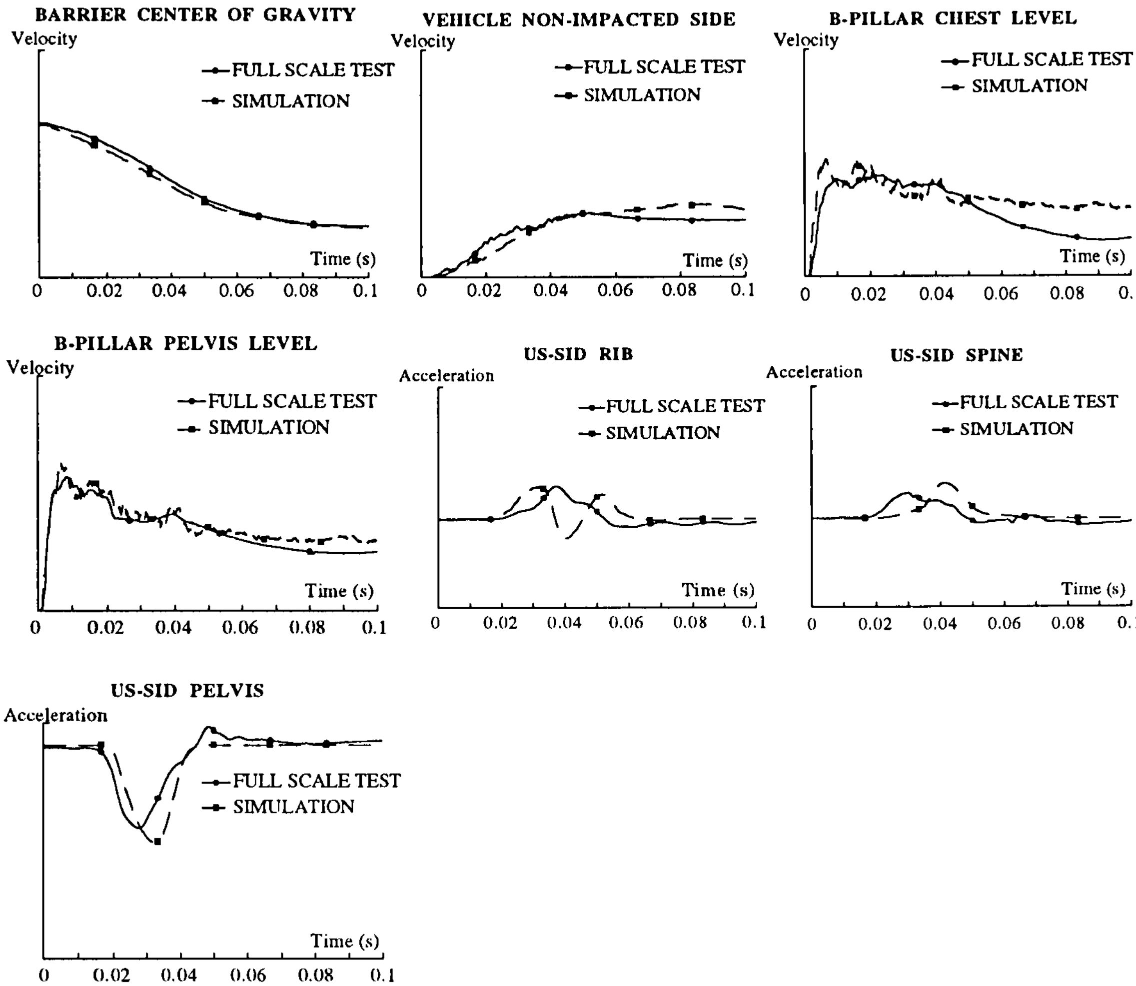


Figure 14. Results from crash test and simulation according to the NHTSA side impact test procedure.

According to the EEVC test procedure, the predicted non impacted side velocity was lower than the measured non impacted side velocity. The predicted B-pillar pelvis level velocity diverged from the measured B-pillar velocity at pelvis level after about 50 ms of the crash sequence. In the crash test, the peak B-pillar pelvis level velocity occurred prior to the peak B-pillar chest level velocity. In the simulation the peak B-pillar pelvis level velocity and the peak B-pillar chest level velocity occurred at the same time.

For the dummy results, the predicted peak rib acceleration was lower than the measured peak rib acceleration. The predicted peak spine acceleration occurred prior to the measured peak spine acceleration. In addition, the shape of the spine acceleration curve differed between the model and the test. The peak pelvis acceleration was higher in the simulation than in the crash test.

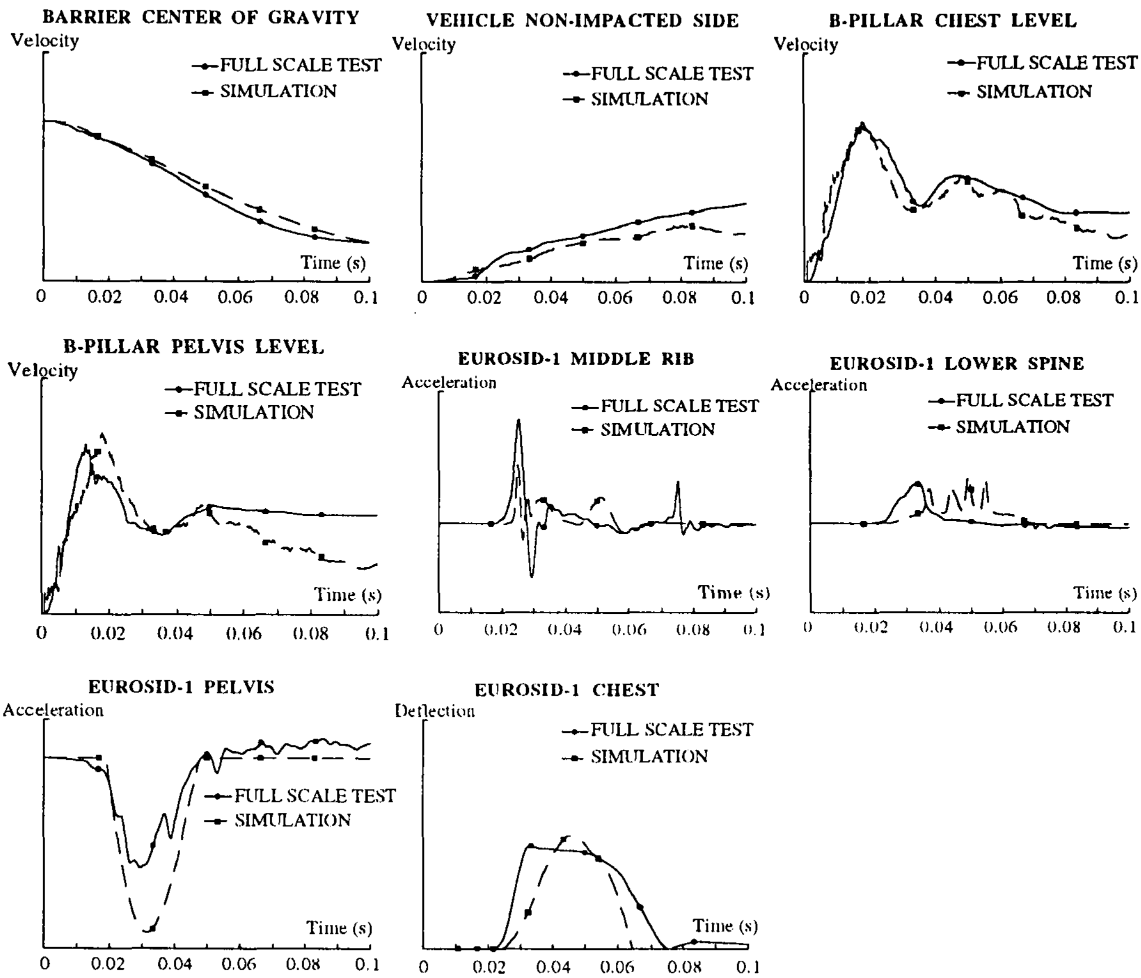


Figure 15. Results from crash test and simulation according to the EEVC side impact test procedure.

DISCUSSION

The hybrid model is an effective tool for quick and economical analysis of the direction of the effects of a large number of design alternatives. The advantage of the hybrid model is that it combines the simplicity of a lumped mass model and the accuracy of a finite element model. Parts of the model was lumped mass while critical parts of the model were modeled by detailed finite element technique. The model was simple enough to run on a computer work station, with computer running times short enough for it to be possible to carry out detailed parametric studies. An added advantage of the hybrid model is that complexity and accuracy can be greatly increased. In addition, as parts of the model with simple representation are replaced by more accurate and complex modeled parts, it is possible to see if the replacements increase the accuracy of the model. If these complex modeled parts does not increases the accuracy they can resume the simple representation.

The hybrid model will thus be of value despite the fact that, the computers are becoming more and more powerful and computing power is becoming cheaper. Today it takes less than 24 hours to carry out full finite element side impact simulations including barrier, vehicle and dummies on a standard engineering workstation. In a few years it might be possible to carry out such simulations in a couple of hours. To develop a full finite element model, geometrical data are needed. Today such data are obtained from computer aided design (CAD) drawings of the vehicle. In the early design phases no CAD drawings of the vehicle structure exist. It is therefore not feasible to develop a full finite element model at this stage. The hybrid model can, in the early design phases, be used to obtain required bending stiffness and crush forces of the vehicle structure. In the future, even with unlimited computer power, the hybrid model will be suitable to use in the early design phases.

Lumped mass models are ideal for carrying out numerous runs at little expense. Therefore lumped mass side impact models have been developed. A number of lumped mass models of moving deformable barrier (MDB), vehicle structure, and US-SID have been developed and validated by comparing predictions from the model with results from crash tests according to the NHTSA side impact test procedure (Kanianthra and Trella, 1989; Low et al., 1991; Prasad et al., 1991). Also lumped mass models of MDB, vehicle structure, and EUROSID-1 have been developed and validated by comparing predictions from the model with measurements from crash tests according to the proposed EEVC side impact test procedure (de Coo, 1990; Dieu and Riss, 1994). These models have only been validated against one side impact procedure which makes them less general than the hybrid model which was validated against both the NHTSA and EEVC side impact procedures. All these lumped mass models can not easily be interfaced with other models. Therefore, in the lumped mass models, all parameters have to be lumped mass including the barriers and dummies. The hybrid model on the other hand is easily interfaced with other models. Finite element barrier models and dummy models are easily integrated in the hybrid model. Today barrier models are readily available since most car manufacturers run full finite element side impact simulations. Finite element dummy models are at present being developed and beginning to be commercially available.

The simple lumped mass models provides a poor representation of parts of the structure in which dynamic effects are significant, such as the tilting of a door as it intrudes inwards. These effects are accurately represented in full finite element models (FE).There have been a number of finite element models of MDB, vehicle structure and occupant developed. One model including US-SID as occupant was validated against the NHTSA side impact procedure (Tsukiji and Taga, 1991). Another model had EUROSID-1 as occupant and this model was validated against the proposed EEVC side impact procedure (Steyer et al., 1989). These models provide accurate results, but the models are time consuming to develop and execute, and are therefore not suitable to use for parametric studies. The hybrid model is very suited to use for parametric studies due to the short handling and execution time.

The hybrid model developed in this study was validated against crash tests according to the NHTSA side impact procedure and the EEVC procedure. There was in general good agreement between the predictions from the model and results from the mechanical tests. In the crash tests, according to both side impact procedures, the measured barrier velocity, B-pillar velocity and vehicle velocity were all in good agreement with the predictions from the model for at least up to 50 ms of the crash sequence. During these 50 ms, all peak dummy injury measures were reached. Therefore, the barrier and vehicle models can be used with occupant models for injury prediction.

The dummies were validated against pendulum as well as crash tests. The measured US-SID peak rib acceleration level in the pendulum test was in good agreement with the peak level in the simulation. After the peak acceleration level was reached, the rib acceleration curve in the simulation diverged from the rib acceleration curve in the mechanical test. The same trend was observed when results from the crash test and the simulation according to the NHTSA side impact test procedure were compared. The measured US-SID spine acceleration level in the pendulum test was in good agreement with the spine acceleration level predicted by the model. However, the spine peak acceleration level in the simulation preceded the peak acceleration level in the test. The same behavior was observed when the measurements from the crash test with the results from the simulation were compared. The Injury measure used with the US-SID chest is the thoracic trauma index (TTI) which is computed by taking the average of the peak acceleration of the rib and the spine. The US-SID model accurately predicted the mechanical US-SID peak rib and spine acceleration levels. Therefore the mathematical dummy can predict TTI. For the US-SID pelvis acceleration, the measurements from the pendulum test and the results from the simulation were in close agreement. This was also the case in the crash test and simulation. Pelvis injury is predicted by peak pelvis acceleration level. In the test and simulation, the pelvis acceleration was in close agreement with one another. It was therefore considered that the US-SID pelvis would be able to predict injury.

For the EUROSID-1, the measured peak rib acceleration level in the pendulum test was in good agreement with the peak rib acceleration level in the simulation. In the results from the crash test according to the EEVC procedure, the predicted peak rib acceleration level was lower than the measured peak rib acceleration level. The peak spine acceleration level in the pendulum test and simulation were in good agreement with one another while the shape of the acceleration curves differed. The same behavior was observed in the crash test and simulation. There was good agreement in peak chest deflection in the results from the pendulum test and simulation. This was also the case in the crash test and simulation. For the pelvis, the results from the pendulum tests were in good agreement with the results from the simulation. The peak pelvis acceleration was lower in the crash test than in the simulation. The injury measure used with the EUROSID-1 were TTI, chest deflection, chest VC and pelvis acceleration. It was considered that the EUROSID-1 model would in terms of TTI, chest deflection and chest VC distinguish the difference in side structure behavior. Albers and Lehman, (1994) found that the results from the mechanical EUROSID-1 in repeated tests varied considerably. In identical crash tests a scatter of 78% in the measurements was observed. Therefore the accuracy of the EUROSID-1 model used in the present study was considered good.

The mechanical EUROSID-1 has an articulating shoulder and a stub arm. The position of the arm is the decisive factor in terms of whether or not the arm and shoulder are impacted by the intruding structures in a side impact. But since the arm and shoulder are not always engaged, the arm and shoulder were not included in the EUROSID-1 dummy model. It has been shown that considerable reductions in injury measures are obtained when the shoulder and arm are engaged in the impact (King et al., 1991). With the model, the protection of the occupant was evaluated when the loading on the chest was greater than when the arm and shoulder of the occupant were impacted. The mechanical US-SID has no articulating shoulder and the arm and shoulder are incorporated in the chest. In the US-SID model the arm and shoulder mass were included in the rib mass. Therefore the loading on the US-SID chest was lower than on the EUROSID-1 chest.

The simple pelvis models consists of one mass and a spring. The dynamic behavior of the mechanical EUROSID-1 pelvis is very complicated. The pelvis behavior under impact conditions is a combination of flesh compression and iliac wing deformation. It was not possible to include this effect in the simple one-dimensional EUROSID-1 pelvis model. The US-SID pelvis consists of a steel structure and flesh. The dynamic behavior of the US-SID pelvis is governed by flesh compression. Despite the simplicity of the dummy pelvis models, they were able to predict the trends in pelvis acceleration with sufficient accuracy.

The hybrid model can be used to evaluate the injury reducing benefits for the chest of vehicle modifications and padding materials. One-dimensional dummy models are good tools to use for this purpose since the response of the chest in this type of evaluation is dominated by one-dimensional compression and translation. Therefore, the one-dimensional US-SID and EUROSID-1 dummy models were chosen for this study. Since there was such a good agreement for the vehicle and barrier parameters between the crash tests and the simulations, improving the dummy models can increase the injury prediction capability of the hybrid model. Adding a head, neck, extremities and a coupling between the chest and pelvis can increase the accuracy and the injury prediction capability of the one-dimensional dummy models. Adding an arm and shoulder on the EUROSID-1 model can increase the injury prediction capability of this dummy model. Making the dummy models two-dimensional with rotation of the upper body around a longitudinal axis can also increase the injury prediction capability. It must be pointed out, however, that these dummy models only respond to design changes that result in changed velocity of the B-pillar. To incorporate three-dimensional dummy models in the hybrid model can make the models respond to other design changes than those that result in a change in the velocity of the B-pillar. Very complex finite element dummies for very detailed analysis can be incorporated in the hybrid model. The hybrid side impact model can be made anything from simple to very complex.

The hybrid model was used to establish the parameter or combination of parameters that influence the injury measures, TTI, chest deflection, chest VC, and pelvis acceleration. A 16 experiment full factorial design in four parameters at two levels was established (Tables 1 and 2). The four parameters were; barrier type, barrier impact velocity, structural collapse level and padding or panel (Table 1). A linear relationship between the parameters and the results was assumed. The factorial design was used with the US-SID and EUROSID-1.

In the experimental design the NHTSA or the EEVC barriers were used at 15 or 35 mph. The structural collapse levels were either increased by 100% or decreased by 50% (Figure 16). On the inside of the door either the standard panel or padding was used. The padding used with the US-SID and EUROSID-1 at the chest level was ARSAN 601 (103 kPa at 35% compression) and at the pelvis level ARCEL 310 was used (221 kPa at 35% compression). Zuby, (1991) found in sled tests that these padding materials produced the lowest injury measure for respective dummy. When padding was used the complete space between the dummy and the door was filled.

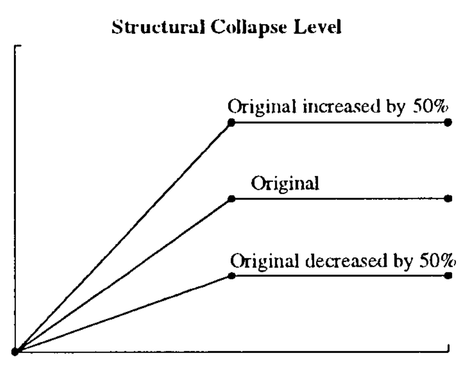


Figure 16. Structural collapse level used in the factorial design.

Table 1.
Four factors used in the 16 experiment full factorial design at two levels

	+	-
Barrier type	NHTSA	EEVC
Barrier vel (mph)	35	15
Structural collapse level	100% increase	50% decrease
Door interior	Panel	Padding

Table 2.
16 experiment full factorial design matrix

Run	Barrier	Velocity	Structure	Door interior
1	+	+	+	+
2	-	+	+	+
3	+	-	+	+
4	-	-	+	+
5	+	+	-	+
6	-	+	-	+
7	+	-	-	+
8	-	-	-	+
9	+	+	+	-
10	-	+	+	-
11	+	-	+	-
12	-	-	+	-
13	+	+	-	-
14	-	+	-	-
15	+	-	-	-
16	-	-	-	-

TTI and pelvis acceleration was evaluated for the US-SID and EUROSID-1 and chest deflection and chest VC for the EUROSID-1. Only the four parameters that had the strongest influence on the injury measures are presented.

In the results from the full factorial design it was observed that all injury measures for both dummies showed a strong response to barrier impact velocity (Figures 17-19). However, the impact velocity can not be altered through design changes to the vehicle. The US-SID TTI showed strong response to the interaction of the velocity and door interior. In addition there was a decrease in TTI for the US-SID with padding. The EUROSID-1 showed strong response to barrier type and to the interaction of the barrier type and barrier velocity. Also the failure level of the structure showed strong response to EUROSID-1 TTI. The parameters that effectively reduces TTI vary depending on the test method used. A design modification found in one test method to reduce TTI may in the other test method increase TTI. Therefore to avoid suboptimisation the vehicles need to be tested according to both the NHTSA and EEVC side impact procedures.

The chest deflection of the EUROSID-1 showed strong response to the failure level of the structure the barrier type and the interaction of the velocity and the failure level of the structure (Figure 18). The chest VC for the EUROSID-1 showed a strong response for the door interior and the interaction of the velocity and the door interior. In addition the failure level of the structure showed strong response to the EUROSID-1 chest VC (Figure 18). To reduce chest deflection and chest VC, strengthening of the structure is the first step and the second step is to add padding. In a previous study it has also been established that the first step in improving the occupant's protection in side impacts is to reinforce the vehicle structure to reduce the door-to-occupant impact speed (Mellander et al., 1989).

Pelvis acceleration for both the US-SID and EUROSID-1 showed strong response to the failure level of the structure, the interaction of barrier type and failure level of the structure and the interaction of the velocity and the failure level of the structure (Figure 19). To reduce pelvis acceleration stiffening of the vehicle structure is the most important modification.

The impact velocity was shown to be the most important injury causing parameter. By reinforcing the structure to significantly reduce intrusion was also shown to reduce injury measures. However, just strengthening the structure may not lead to minimize the injury measures but may provide a guidance to the influence of the structural stiffness on the injury measures. To minimize the injury measures, the stiffness of the structural members need to be tuned to each other in order for the deformation of the vehicle to engage as large an area as possible of the vehicle side in an impact.

The hybrid side impact model is a good tool for simple, quick and economical analysis of the direction of the effects of a large number of design changes. The model is well suited for parametric studies. For example running 80 ms of a crash sequence requires approximately only one hour on a standard engineering work station. A wide range of impact conditions such as barrier-to-car, car-to-car impacts. Pole-to-car impacts are also possible to simulate as well as to estimate the potential injury reducing benefits of padding or side airbags. The hybrid side impact model can be used to guide the design engineer in understanding the effect of changes in the vehicle structure and to select promising design directions already at an early stage of the design work.

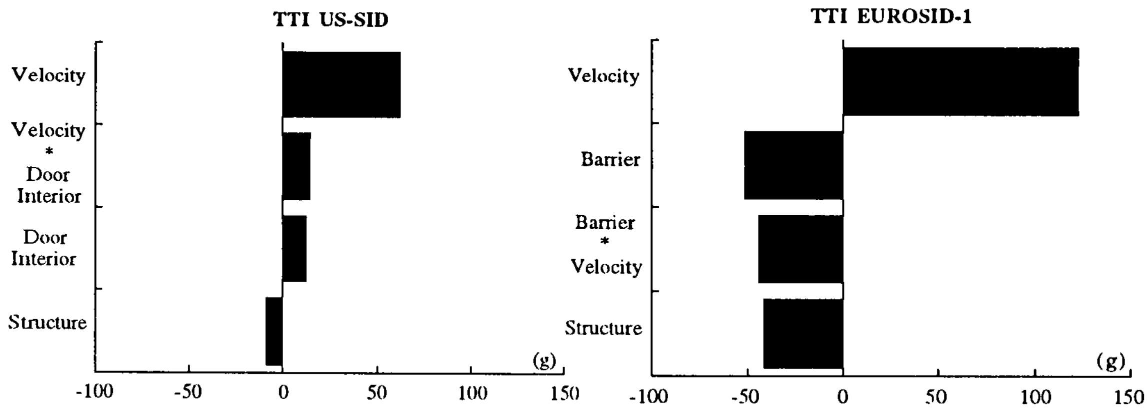


Figure 17. Influence of factors on TTI.

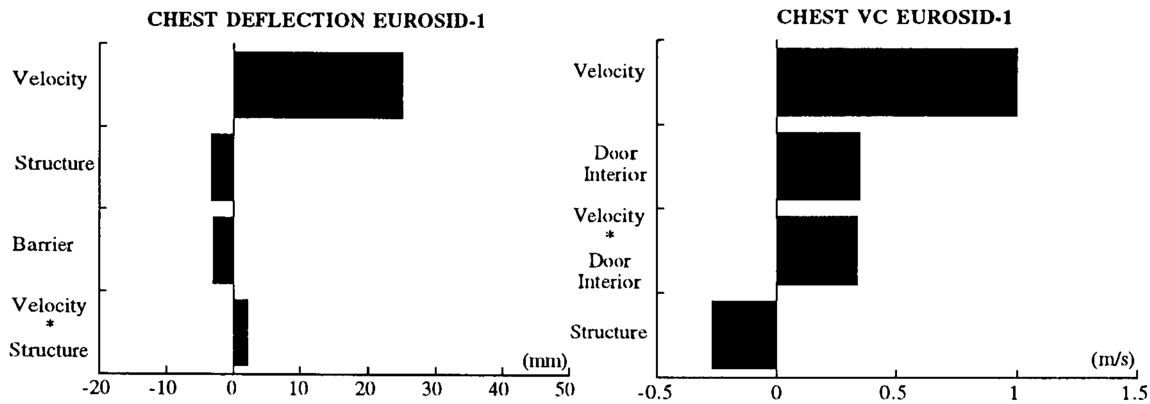


Figure 18. Influence of factors on chest deflection and chest VC.

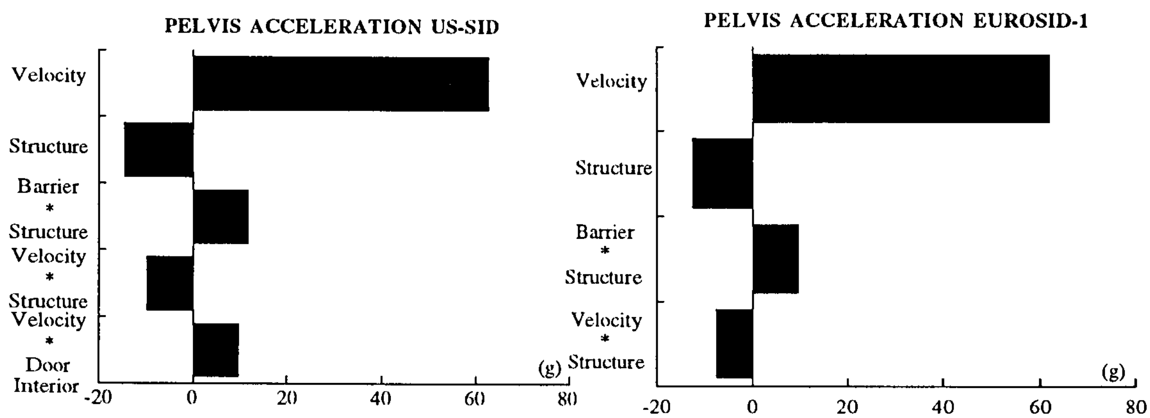


Figure 19. Influence of factors on pelvis acceleration

ACKNOWLEDGMENT

The author acknowledges the Swedish National Road Administration for sponsoring this work and Renault for supplying the EEVC foam barrier model.

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OPTIMISATION OF CRASH PULSE THROUGH FRONTAL STRUCTURE DESIGN

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ABSTRACT

Structural analysis techniques are now available that allow the development of safer vehicle structures. The basis for the development strategy is new knowledge of the biomechanics of injury and of the spectra of real life accidents occurring in Australia, plus new computer simulation techniques. An optimised crash pulse is proposed, which analysis suggests will provide reduced injury risk throughout the spectrum of frontal crashes. Design of a vehicle with a front structure having this characteristic could result in reduced injury risk to occupants in the complex interaction between the occupants and the vehicle during the variety of crash types and speeds that occur on Australian roads.

INTRODUCTION

Motor vehicle accidents are causing increasing public concern, as evidenced by the frequent media coverage whereby they are identified as a major cause of injury and death, resulting in substantial personal, social and economic cost to the community.

The three crash parameters which influence the injury risk to vehicle occupants are the severity of the crash, the behaviour of the vehicle structure and restraint system, and the vulnerability of the passengers. The crash severity is basically determined by the collision speed, and the stiffness and mass of the car or obstacle struck. The behaviour of the vehicle system is determined by the way energy is absorbed, by the strength of the passenger compartment, and the characteristics of the restraint system. The vulnerability of car occupants is determined by their seating position, their health, size, sex and age.

Legislative regulations in all the industrialised countries are setting requirements for the performance of vehicles in crash tests. Australia has adapted the United States Federal Motor Vehicle Safety Standard FMVSS 208 for a full frontal impact. Recognising the high seatbelt wearing rates in Australia (over 95% in front seats) in contrast to the USA (less than 50%), Australian Design Rule ADR 69 incorporates the same test procedure and injury limits as FMVSS 208, but specifies restrained dummies (1). Beyond government legislation, a number of organisations have conducted tests at a higher speed with the objective of evaluating relative safety performance, on the assumption that measurements made during a higher speed barrier test would indicate improved field performance.

The behaviour of the vehicle structure during a collision, so-called vehicle crashworthiness, has a major influence on the occupant injury risk. During a collision, the part of the structure that is in contact with the impacting or impacted object deforms. The energy required for this deformation decelerates the remaining, still undeformed part of the vehicle, including the passenger compartment. The deceleration/time signature of the passenger compartment is referred to as the crash pulse. It is ultimately the shape of this crash pulse that determines the magnitude of the injury risk. Understanding the relationship between the biomechanics of collision injury and the crash pulse of the vehicle structure provides the opportunity for improvements in vehicle crashworthiness. Modern methods of computer simulation of the occupant kinematics and of the behaviour of the crushing structure, combined with experimental tests, make it possible to optimise solutions to the complex problem of occupant protection in car collisions.

Safety Design Objectives

The objective in vehicle safety development at Holden is to protect car occupants by minimising their injury risk. At the moment, media focus is on safety devices, such as new seat belts or airbags, and on the test results generated by the New Car Assessment Program (NCAP). This focus does not recognise that the injuries that occur on Australian roads are the result of a spectrum of crash types and severities, of vehicle responses and of occupant vulnerabilities.

In developing a vehicle to provide the maximum injury risk reduction benefit to the community, this broader consideration must be given to the problem. The complete vehicle, with its structure, its restraint system and its occupants, has to be considered as a total system. In frontal crashes, two separate parts of the structure perform different functions. A sequential process is required to first develop the front structure, the pulse-creating crush zone that absorbs the crash energy, and then the protective, rigid passenger cell that supports the loads generated, and finally to optimise the restraint system for the broadest spectrum of passenger protection.

Crashes in Australia

Achieving safety for vehicle occupants requires more than satisfying the legislated safety regulations. It is necessary to understand the types of crashes that occur, their frequency and the associated injury risks. The frequency of distribution for the point of impact has been identified in data collected by Vic Roads and the Federal Office of Road Safety in the period from 1/1/89 to 31/12/92 (2). The data base contains 171,502 recorded accidents, including all levels of injury, ie. from no injury to fatality. After some redistribution of the frequencies to make the classes of impact points comparable with the US statistics (3), the results shown in Figure 1 were obtained (US results in brackets).

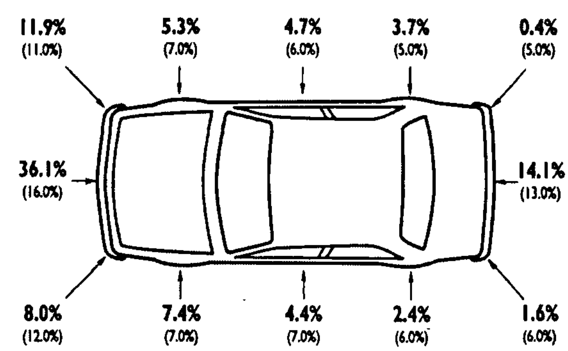

Figure 1. Impact Directions in Australia and USA

The break-down of the distribution from Figure 1 into more general categories of frontal, side and rear impacts is tabulated below, and shows the statistics for serious and fatal accidents in Victoria.

Table 1.
Comparison of Injuries From Crash Directions, Victoria and USA

CATEGORY	FRONT %	SIDE %	REA %
All Injuries, Victoria	56	28	16
All Injuries, USA	39	38	24
Serious and Fatal, Victoria	62	32	6
Fatal, Victoria	51	46	3

The comparison with the US statistics is valuable because this information is often used in discussions, when information about Australian conditions is not available. It can, however, be seen that there is a substantial difference between the Victorian and the US distributions. This emphasises the value of optimising the frontal collision performance, when frontal collisions in Australia cause about 60% of all serious and fatal injuries.

Crash Speed

The severity of a crash is determined by the direction of impact, the collision speed and the shape, stiffness and mass of the car or object struck. Most collisions occur at low speed, as is shown below. For many years the focus of road safety initiatives has been on reducing road fatalities. More recently a more comprehensive evaluation is being made, involving societal harm, which considers, in addition to fatalities, the effect on society of all injuries suffered in road crashes. The frequency of injuries of greater severity than AIS 2 is shown in Figure 2 below (4). This category includes all injuries classified as serious, severe, critical and maximum injury (virtually unsurvivable) (5). This emphasises that appropriate priority needs to be given to reducing the injury risk over a large range of collision speeds.

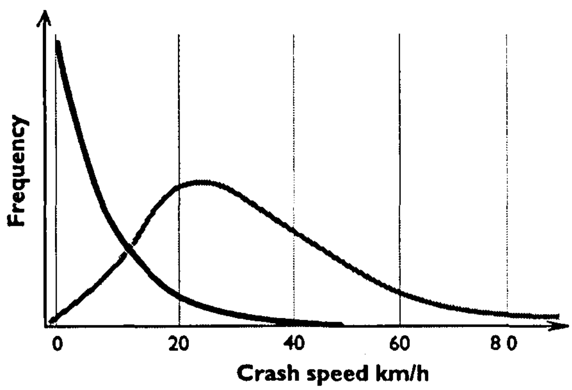

Figure 2. Crash Frequency Against Speed

Injury Risk

Injury risk relates to the field performance of the vehicle, since the vehicle will be used by occupants who vary in health, size, sex and age, and who will be involved in collisions at various speeds, at various angles and overlaps, and with obstacles and other vehicles of various sizes and stiffness. The vehicle manufacturer must satisfy the government regulation for performance at 48 km/h frontal collision with 50th percentile male dummies. It must also account for the needs of all of its customers, many of whom are small females, or large males, or older people with more fragile bones, and who will become involved in a range of accidents. Vehicle system performance must result in minimum injury risk to all vehicle occupants in all accident situations.

A measure of soft tissue injury risk is provided by the Viscous criterion, defined as $(V*C)_{max}$ (6). This provides a method of measuring in test dummies the risk of soft tissue injury. The maximum value of velocity and displacement of soft tissue areas of the dummy body measured during a test indicate the risk of real life injury. The measurements are not part of ADR 69 or NCAP tests, however they relate to injury risk to people in real life crashes. This evaluation was applied to the results of analysis described later in this paper.

Crash Pulse

A vehicle crash pulse is the deceleration time signature generated when a collision occurs. This pulse determines the occupant loading and hence the injury risk for a passenger in a particular vehicle involved in an accident. In a collision, the vehicle's kinetic energy ($\frac{1}{2}mv^2$) must be dissipated by deformation of the vehicle structure (Fx). An initial response to this engineering challenge might be to achieve this energy dissipation with maximum structural efficiency by designing a crush zone with a constant force characteristic, to suit the government regulation ADR 69 test speed of 48 km/h, or the NCAP test speed of 56 km/h.

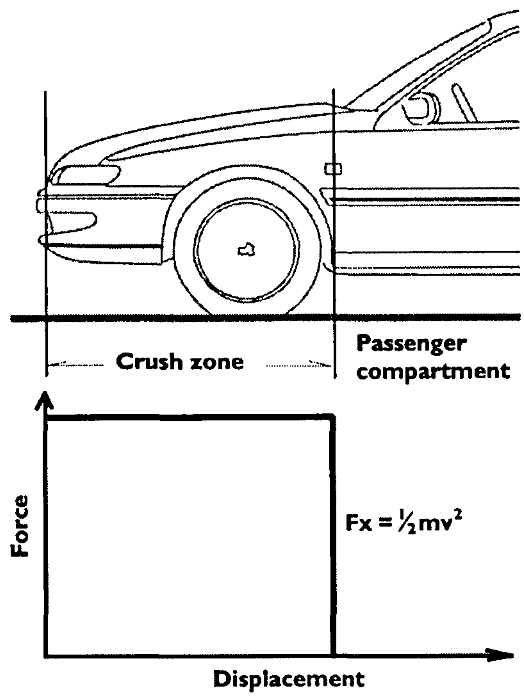

Figure 3. Theoretical Crush Characteristic

This approach can be seen to be inappropriate if the performance at collision speeds higher and lower than the design speed is considered.

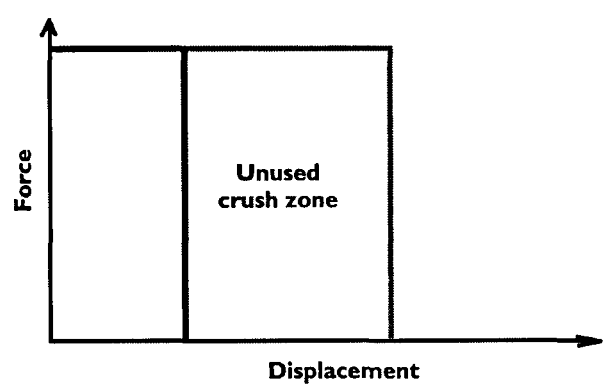

Figure 4. Crash Pulse for Low Speed Collision

In a low speed collision, the high force causes a high deceleration that unnecessarily risks injury to older people commonly involved in low speed collisions, as illustrated in Figure 4, above.

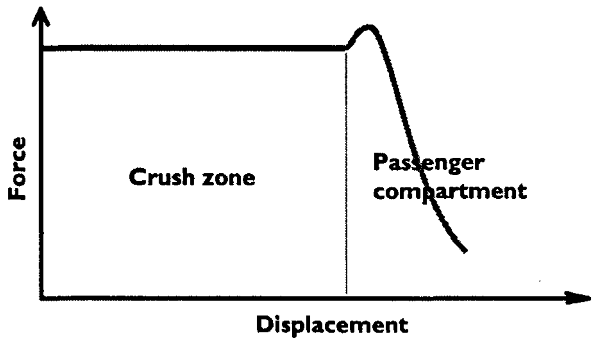

Figure 5. Crash Pulse for High Speed Collision

At higher speed, the kinetic energy exceeds the energy that can be absorbed by the crush zone. The excess energy will cause the passenger compartment to collapse and place the passengers at a high risk of being crushed, as shown in Figure 5.

To provide safety benefits to vehicle occupants involved in collisions in the field, the spectrum of collision types and speeds must be considered. An "optimised" crash pulse can provide benefits in low speed and high speed collisions, as well as providing improved performance at typical collision speeds. The crash pulse must accommodate three conflicting requirements:

1. Minimum vehicle damage in low speed crashes.
2. Minimum deceleration and hence occupant loading for the most frequent crashes.
3. High energy capacity for high speed collisions.

These requirements are shown schematically in Figure 6.

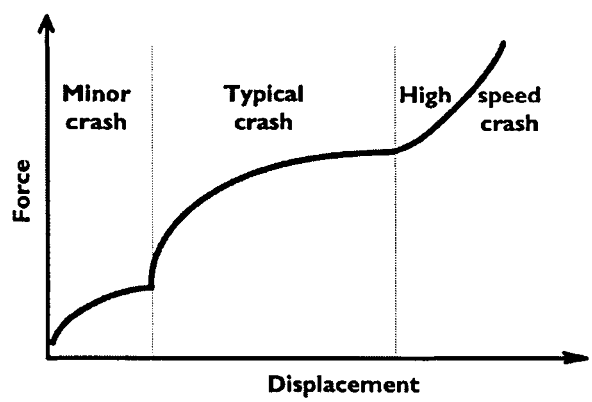

Figure 6. Idealised Crash Pulse

Modelling of deforming structures is complex, because the deformation being modelled changes the behaviour of the structure from one instant to the next. Modelling of this type requires substantial computer power. Development of vehicle crash pulses with characteristics similar to that proposed above is now possible as a result of recently developed computer modelling techniques utilising high powered work stations. These techniques allow the exploration of structural characteristics that have previously been impractical.

Front Structure Development

The design of a new model vehicle provides the opportunity to utilise computer modelling techniques to develop the vehicle's front structure to give improved crash performance. Models for frontal collision of up to 80,000 elements can be used for simulating offset collisions. As an illustration of the opportunities available to change the crush force/deflection characteristics of the front structure of the vehicle, the effect of changing the engine side rail section and weld pattern is shown in Figures 7 and 8 (7). The results demonstrate the potential for utilising changes to shape and weld flange location to increase or decrease the required crush force by 30%.

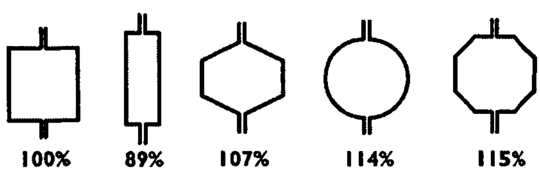

Figure 7. Crush Force for Various Sectional Shapes

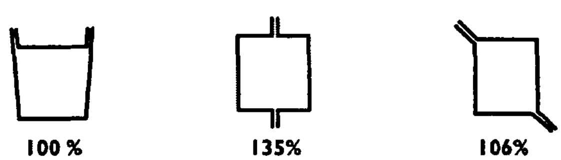

Figure 8. Crush Force for Various Weld Flange Configurations

Design of the energy absorbing front structure must consider not only full frontal collisions, but also the full spectrum of crash types - offset, angle, pole, etc, their relative frequency on Australian roads and the injury risk involved. Minimum injury risk design must thus be the optimum compromise considering the frequency of each collision type, and the injury risk involved, in order to achieve the minimal social harm.

"Smart" restraint systems are being developed which modify the performance of seat belts and airbags to suit the needs of the occupants. They sense passenger presence and position, and crash severity, and control airbags with variable pressure characteristics, selecting the required airbag performance to suit the particular crash severity and occupant position. This approach may enhance an optimised crash pulse, or compensate for a less efficient crash pulse.

With the increasing trend towards smaller vehicles, alternate vehicle structure designs are being explored which may lead to very rigid structures, with energy absorbing attachment of the occupants to the passenger compartment. This approach may lead to a crash pulse with different characteristics to that being proposed.

Crash Pulse Evaluation

Crash pulses are generally described as either "soft" or "stiff", depending on the length of the deceleration pulse. Three sets of generic deceleration curves were generated, representing soft, stiff and optimised crash pulses, for full frontal collisions with a barrier at speeds of 30, 45 an 55 km/h, as illustrated in Figures 9, 10, and 11.. These pulses were then applied to a computer model of a 50th%ile male seated in a Commodore in the mid seating position, restrained by the production seatbelt system, utilising the occupant simulation code MADYMO.

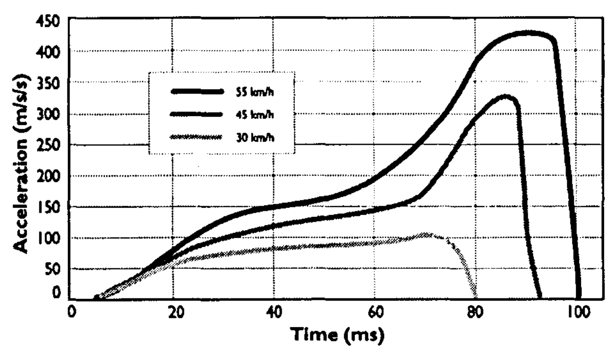

Figure 9. Optimised Crash Pulses

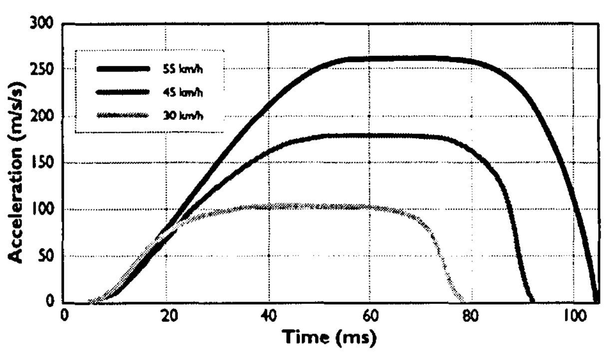

Figure 10. Soft Crash Pulses

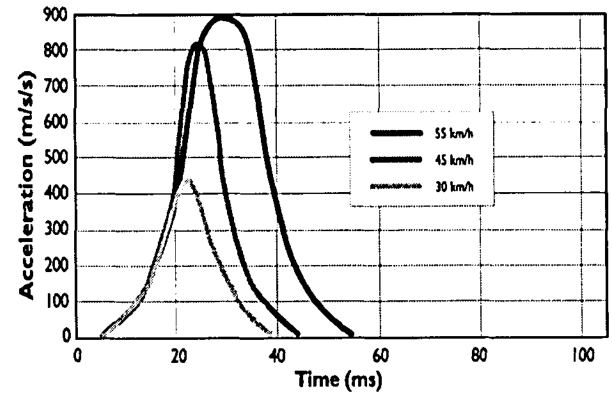

Figure 11. Stiff Crash Pulses

The resulting thoracic loading was then evaluated by considering the magnitude of the Viscous Criterion measured in the computer model thorax.. This measure provides a most sensitive index of thoracic injury risk (risk of injury to the heart and lung and abdominal viscera) as a result of seat belt loading. Head and neck loading were not considered in this evaluation, as these are predominantly the result of occupant kinematics and head contact, and hence do not directly correlate with changes to the crash pulse. The results of this analysis are shown in Table 2, and indicate the potential for injury risk reduction throughout the crash speed range, plus the ability to extend the crashworthiness of the vehicle to higher speeds through optimising the crash pulse.

Table 2.
Viscous Injury Risk

SPEED	SOFT PULSE	OPTIMISED PULSE	HARD PULSE
30 km/h	0.14 m/s	0.14 m/s	0.19 m/s
45 km/h	0.26 m/s	0.22 m/s	0.48 m/s
55 km/h	0.36 m/s	0.27 m/s	0.89 m/s

Factors Affecting Frontal Crash Pulse

There is a spectrum of competing requirements for the design characteristics of the front structure, the global load paths and subsystems, in order to absorb crash energy efficiently, with minimum peak deceleration, and appropriate energy absorption for high and low crash speeds, and for full frontal, offset and pole crash types. Computer analysis and optimisation techniques are now enabling the design of vehicle structures which provide improved safety for the whole community.

Optimisation of Occupant Protection

Techniques have been developed to optimise the restraint system performance to achieve minimum societal harm (12). The gains achieved from this optimisation are very dependant on the vehicle crash pulse. Development of a crash pulse with the characteristics proposed in this paper requires an understanding not only of the crush characteristics of the vehicle front structure, but of the distribution of crash loads and energy, and the dynamic interaction of sub-systems within the structure. Examples are the interaction of the front structure and the engine, and in the case of a rear wheel drive vehicle, of the role of the driveline in the distribution of crash loads and energy.

CONCLUSIONS

Opportunities exist for reducing the injury risk in frontal collisions by optimising the crash pulse to achieve reduced occupant loading over the spectrum of collision types.

Car design for safety is a complex process with many interdependent performance parameters influencing the outcome, ultimately the injury risk to the vehicle's passengers. Improvements are a result of comprehensive computer modelling, experimental testing and evaluation, and careful weighing of the relative benefits throughout the field accident spectrum. To think that safety is merely a matter of "bolting in" a new seat belt or airbag, or of achieving good performance numbers from one synthetic test such as NCAP, is not to comprehend the lengthy development required to ensure that reducing the injury risk for some passengers in one crash situation does not create an increased risk for other passengers in another crash situation. The optimum safety is provided by the design that gives the lowest risk to the community of vehicle occupants.

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RACE CAR SAFETY DEVELOPMENT
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Paper Number 96-S4-W-23

ABSTRACT

Insights are provided into the engineering challenge involved in providing crash protection to the driver when modifying production cars for racing conditions. Building on knowledge gained from passenger car safety development, the higher performance parameters associated with high speed racing crashes are reviewed and proposals developed to minimise the injury risk to racing car drivers.

Structural modifications to improve front, rear and side crash performance are proposed, utilising the opportunity to make modifications not practical in a production vehicle, but which can be implemented in a racing car. The role of the seat and seatbelt system in driver protection is discussed, and the potential for major gains in safety are discussed.

INTRODUCTION

The modification of production cars to suit motor racing poses a number of technical challenges, not the least of which is the design of changes to make the car crashworthy under racing conditions. Most frontal impact crashes on normal roads occur at relatively low speed, and safety systems for passenger vehicles are optimised for these speeds. Racing accidents on the other hand may occur at high speed. A high speed crash creates a large amount of crash energy, and this energy must be absorbed with minimum risk of injury to the driver, either from the rapid deceleration, or from collapse of the passenger compartment. There is the potential to modify race cars based on production vehicles to better manage the increased crash energy involved in a high speed crash.

Recent motor race crashes, both in Australia and internationally, have encouraged consideration of the potential for safety improvements to racing cars. One of the most popular forms of motor racing in Australia is known as Group A Touring Cars, and is based on current production Holden and Ford full size passenger vehicles. Safety in production cars has increased markedly over recent years with the introduction of driver and passenger airbags and extensive computer simulation of vehicle structures. There is thus the potential to extrapolate the safety technology developed for passenger vehicles and to apply it to the motor race versions of these vehicles. The aim of racing car safety improvement should be to minimise the injury risk to all drivers in the range of crash types in which there is a chance of being involved.

PHYSICS OF A CRASH

A crash occurs when a cars strikes another object, either another vehicle or the perimeter of the circuit. The severity of the crash is determined not by the vehicle speed prior to the crash, but by the change in speed (ΔV), and the rate of change in speed, or deceleration. The deceleration is determined by the mass and stiffness of the object struck. Prior to the crash, the car has energy of momentum, related to its mass and velocity ($\frac{1}{2}MV^2$). To stop the car completely, the crash must dissipate all of this energy. Part of the energy is dissipated by deforming the car structure, and part of the energy is absorbed by the object struck. When the object struck is very rigid, such as a concrete barrier, all of the energy must be absorbed by the car. Figure 1 shows the force generated in crushing the front structure of two cars.

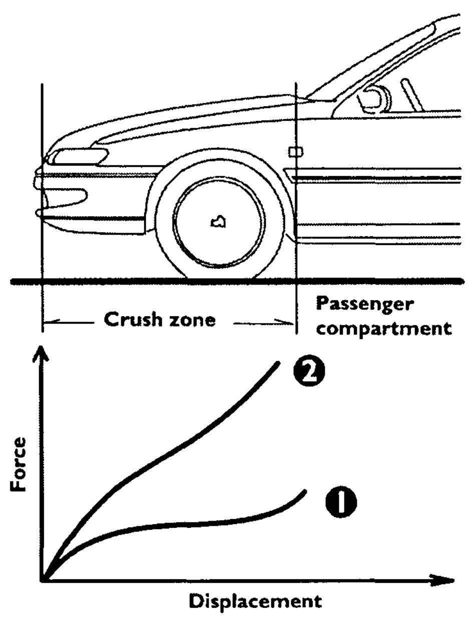

Figure 1. Frontal Crush Force

During a crash, the crushing continues until the crash energy is absorbed, that is, until the energy of momentum ($\frac{1}{2}mV^2$) equals the energy required to crush the front structure ($\int f dx$). The driver is at risk if all of the crash energy is not absorbed in the front structure, and the passenger compartment then begins to deform, or if the crush loads become so high that the resulting crash deceleration experienced by the driver causes injury.

MOTOR RACE CRASHES

Comprehensive data is now being collected about the type, severity and injury outcomes of crashes on Australian roads (1). Holden has a national Accident Investigation Program to analyse the crash performance of Commodores (2). A critical characteristic of this program is the evaluation of the crash severity, ΔV. No suitable data is available on racing crashes in Australia which adequately identifies the crash severity. Although General Motors has a program in the USA of fitting crash recorders to Indy cars, little data is available on sedan car crashes. It is possible, however, to make some comparisons between crashes on the race track and crashes on public roads. For example a Touring Car crash into a concrete barrier can be compared with a frontal barrier test. The direction and speed of impact of potential race crashes can be estimated from a knowledge of a circuit and of potential car speed. The front structure of a passenger car is designed to minimise the injury risk to occupants at around 50 km/h, the speed at which most crashes occur on public roads. The racing speed range for a touring car is 80 to 300 km/h. The structure of a production car is not designed to manage the collision energy associated with crashes at racing speeds.

INJURY RISK

Racing car drivers involved in crashes are at risk of injury from several causes. If the crash forces exceed the passenger compartment strength, there is the injury risk associated with the vehicle structure collapsing in onto the driver, causing crushing injuries. Racing seat belt harnesses are relatively stiff, and couple the driver firmly to the vehicle. If vehicle deceleration during the crash is extreme, the driver is at risk of thoracic injury caused by rapid compression of the chest cavity, causing viscous injury to the soft tissue of internal organs. Although the driver's torso is closely coupled to the vehicle by the seat belt system, the driver's head is unrestrained and can flail about in a severe crash. This situation is exacerbated by the mass of the crash helmet, usually an additional 20% of head mass. The loads applied to the driver's neck, a relatively fragile part of the human body, are consequently increased, and this can cause neck injury.

In recent years Formula 1 drivers have been seen to survive some very severe crashes, despite being subjected to forces which appear to be beyond the limits established by current biomechanical knowledge. The injury tolerance of the human body is imprecisely defined. The first scientific measurements began in 1954 when Colonel John Stapp volunteered to be strapped to the front of a rocket sled that decelerated from 1029 km/h to rest in 1.4 seconds and suffered no permanent injury (3,4). The additional factor in these recent crashes which enabled these drivers to survive is believed to be their young age, small stature and extreme fitness. The majority of drivers competing in Australia's Touring Car Championship are larger, older and less fit than their Formula 1 counterparts, and are hence more at risk of injury. The crash performance of Touring Cars needs to suit the lower injury tolerance level of the drivers of these cars.

VEHICLE STRUCTURE

Changes could be made to the structure of Touring Cars which would reduce the driver injury risk in severe crashes. These modifications need to combine changes to manage the significantly higher levels of crash energy involved in severe motor racing crashes compared with the types of crashes that occur on public roads, plus changes to strengthen the passenger compartment to support the increased crash loads without collapsing onto the driver.

Front crash requirements

In order to manage the increased energy of a high speed race crash, the front structure could be modified to absorb increased energy. Modifications could increase the energy absorbed by existing structure forward of the firewall.

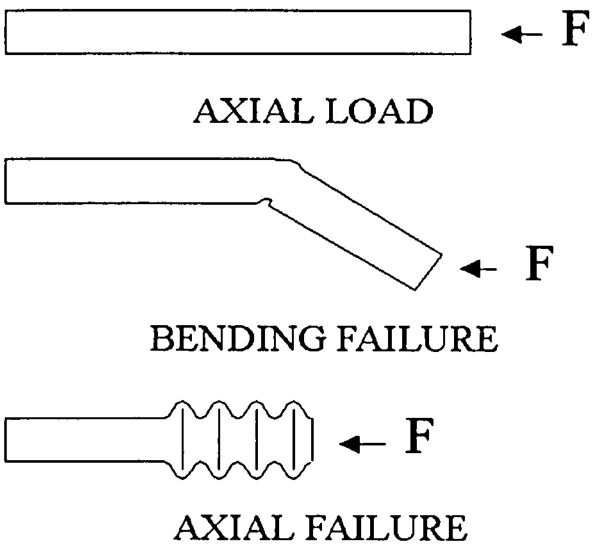

Figure 2. Collapse Modes of a Column

Figure 2 illustrates how an existing design can absorb more crash energy if it is supported to collapse in the most effective way. The use of structural foam in the engine side rails, while not absorbing energy itself, can support the structure to allow it to collapse efficiently.

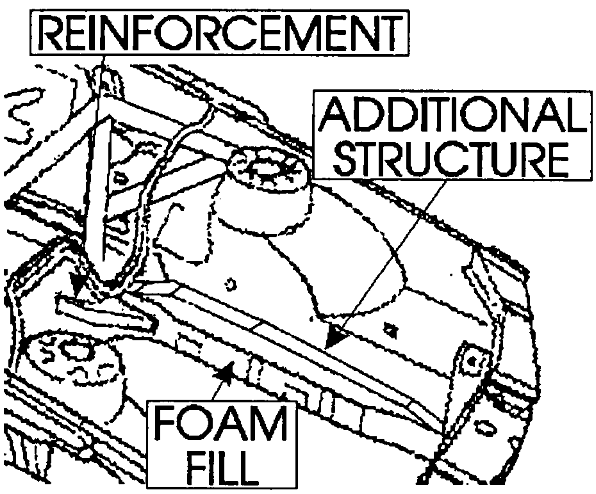

Figure 3. Front Structure Changes for Increased Crash Energy Management

Other modifications could add deformable structure to absorb additional energy, and add structure to the passenger compartment in order to support the higher loads generated in the front structure. Figure 3 illustrates the use of additional structure to absorb additional crash energy, providing progressively increasing stiffness. The additional reinforcement of the passenger compartment in order to support the increased loads is also illustrated, both local reinforcement to the lower frame, plus use of the roll cage to support the strut tower. The progressively increasing stiffness is required to ensure that minimum loads are generated during lower speed crashes, with minimum injury risk resulting, but increasing stiffness to manage the increasing crash energy associated with high speed crashes. This is illustrated in Figure 1, where curve 1 represents the production vehicle, and curve 2 represents the characteristic required for a Touring Car (5).

Rear crash requirements

As most seat belt systems have the upper straps anchored to the rear of the vehicle, it is important that all crash energy is absorbed by the body structure rearwards of the belt attachments, otherwise the restraint of the driver is compromised. This area of the body must absorb crash energy in the same way as the front structure, and similar modification is needed. In a rear crash, the seat must support the driver, performing the function of the seat belt system in a frontal crash. Appropriate attention is given to ensuring that seat belt attachments are adequate to support crash loads. This is not usually the case with the driver seat installation. A significant seat mounting structure is required to support the loads generated in a rear or side impact crash.

Side impact crash requirements

A side impact crash is significantly different from a frontal crash. In a frontal crash, the driver is decoupled from the vehicle crash. The deformation of the structure is remote from the driver, who experiences a later, secondary crash with the restraint system. The two events require separate engineering solutions. The front structure can be designed to manage the crash energy at minimal force levels. The restraint system can then be designed to transfer the minimum loads to the driver. In a frontal impact the crush space can be as much as 800 mm. In a side impact, the distance between the outer door surface and the occupant can be as little as 200 mm. In a side impact crash, the driver becomes coupled to the vehicle crash when he contacts the door. Figure 4 shows the velocities of the striking vehicle, the struck vehicle and the in-coming door of a production vehicle. Impact between the driver and the door occurs when the door velocity is a maximum

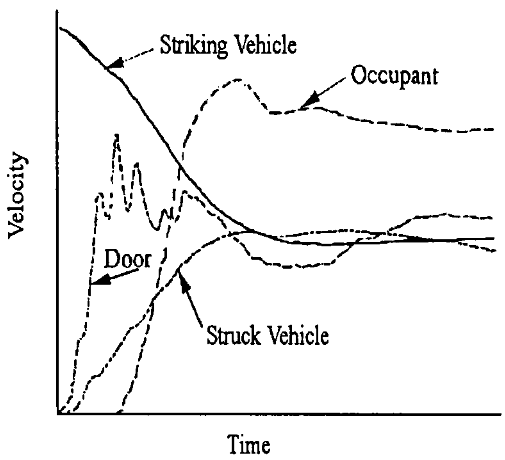

Figure 4. Velocity Comparison - Struck and Striking Vehicle, Door and Occupant

The relative velocity between the driver and the door (ΔV) is a measure of injury risk. The relative velocity can be minimised if the door intrusion is minimised, if the space between the driver and the door is maximised, and if the driver is well connected to his car, and moves with his car away from the striking vehicle during the crash. The driver's head, neck and upper torso must be supported by the seat, and the intruding door surface must be flat and padded. Note that padding represents a compromise, as although the padding can reduce the peak acceleration at impact between the driver and the door, it also results in earlier contact and thus potentially increased relative velocity.

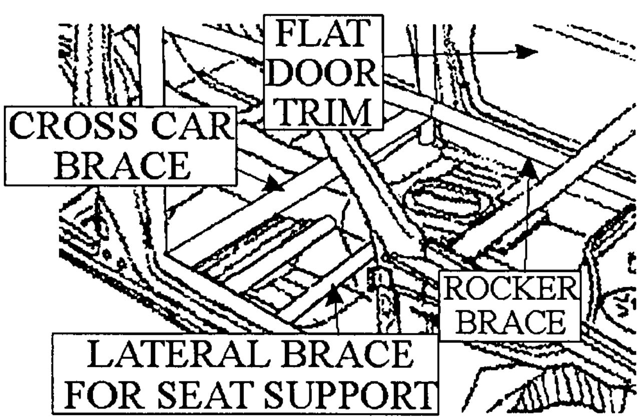

Figure 5. Cross Car Reinforcement, Door Trim Modification and Rocker Reinforcement

Minimising door intrusion requires that the side structure of the car is increased by utilising roll cage structure across the door opening, tied to the A and B pillars, lateral reinforcement between the B pillars, and lateral reinforcement of the floor to support the seat installation, which moves the seat inboard as rocker deformation occurs. These modifications are illustrated in Figure 5

RESTRAINT SYSTEM

The restraint system, which incorporates the seat belt and the seat system, should distribute loads to the occupant and incorporate compliance in the seat belt webbing to reduce peak deceleration while preventing driver contact with the steering wheel or other surfaces within the cockpit. The driver's mass, size and age influence the performance requirements of the restraint system. Motor sport authorities have recently made webbing width of 75 mm mandatory in Formula 1 to better distribute loads over a drivers torso. The difference in stiffness between 50 mm and 75 mm webbing may however increase the peak deceleration forces seen by the driver. Compliance in the restraint system allows the loads applied to the driver to be reduced from the peak deceleration experienced by the vehicle. Compliance is limited by the space available before the driver contacts any interior surface. Increased compliance can be achieved by the use of more compliant webbing, or by load limiting devices that allow webbing pay-out above a load limit. Optimising the compliance of the restraint system will allow the loads applied to the driver to be minimised.

Correct seat belt geometry is required to ensure that crash loads are distributed into the drivers body in a way which minimises injury risk. The way in which the belt passes over the driver's torso determines the load distribution and hence the injury risk. Some forms of motor sport mount the driver's helmet to the roll cage or body structure. Most sedans mount the seat belts to the rear floor or parcel shelf. These methods do not take into consideration the deformation of the structure and therefore there is a risk of belts either loosening or tightening during a crash and causing injury.

SEAT

After the seat belt system, the seat is the most important part of the driver restraint system. Beyond supporting the driver during cornering, the seat must be able to support significant crash loads. In a frontal crash, the seat base must support high loads applied by the belt system, and support the driver's back and neck during rebound. During a rear crash, the seat back must support very high loads without failure, of the same magnitude as loads applied to the restraint system. Seat support for the driver during a side impact crash is critical for safety, and the seat belt system provides little support for the upper torso during side impact.

The driver's seat should be located as close to vehicle centre as possible, maximising space between driver and side of the vehicle. It is also important to move the driver with the vehicle in a side impact, requiring improved seat attachment to the vehicle structure, improved structure supporting the seat, and improved seat structure.

Future seat design should take into consideration these driver protection roles. The seat should provide lateral support in a side impact, head/neck support for both frontal and side impacts, and integral seat belt mounting to minimise the possibility of geometry based injury discussed previously. Infant capsules provide a useful example for seat design. An infant is completely decoupled from the occupant compartment, the restraint system mounting points move with the capsule, and it provides excellent lateral support for body and head. The development of an integral driver's seat, seat belt and mounting system is a future race car safety challenge.

ROLL CAGE CONNECTION TO STRUCTURE

The roll cage can make the maximum contribution to driver safety if it is continuously connected to the body structure, and thus contributes the maximum to structural integrity. The centre hoop of the roll cage must be well rearwards of the drivers head, to ensure that no head contact is possible even in a severe crash. The bar represents a dangerous contact point, and use of rubber foam as a protective cushion is ineffective.

STEERING COLUMN

In the process of modifying a production car for racing, it is common practice to replace the original collapsable steering column with a rigid column. Use of the original equipment collapsible steering column should be reconsidered, as a rigid column represents a dangerous contact for a driver in a severe crash.

HELMET

In a well designed race car system, the driver's helmet has a limited role, as the driver should be supported to prevent any head contact. In this situation, the helmet represents both a protection system and an injury risk. Although the driver can be held very rigidly to the car by the seat belt system, the head is restrained only by the drivers neck. In a severe crash, the drivers neck may be subjected to an additional 200 kg loading due to the helmet mass, significantly increasing the risk of neck injury. The recent extension of the cockpit sides of Formula 1 cars should limit the neck injury risk in side impact crashes in these cars. A helmet tether system developed for Indy car drivers is currently being tested for suitability for Touring Cars by Holden.

Humans sense acceleration and movement through canals in the inner ear and it is possible that the increase in head inertia could dampen the drivers ability to sense vehicle dynamics, possibly reducing performance. Drivers of well designed cars of the future may be able to do away with helmets, or utilise light weight helmets, as head protection will be supplied by the passenger compartment.

TRACK SAFETY

The design of race circuit safety characteristics is an important part of minimising the injury risk to drivers and spectators. There is a realistic limit on the absorption characteristics of a racing car. Above and beyond these physical limits to race car crash performance the track safety characteristics will determine the risk to drivers.

Many people have commented that racing accidents result in the car having a glancing blow with the wall and therefore high speed crash protection is not needed. This may be true in most crashes, however the recent spate of deaths in racing has demonstrated that severe crashes can occur, independant of whether they are labelled freak accidents or just statistical distribution. It is inevitable that these serious crashes will occur.

CONCLUSIONS

Production car research provides some valuable insights into the areas of development required for improved race car safety. Potential exists for improvements in Touring Car design to provide better crash performance in front, side and rear crashes. This includes changes to the deformable structure, the passenger cell, and the driver restraint system. Understanding of the concepts of injury risk reduction needs to be developed beyond the current perception that strength and reinforcement equate to safety. An understanding of crash energy management , and the limits of human tolerance to injury caused by force, deflection, velocity and acceleration can lead to the development of improved safety for racing drivers in Australia.

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A THEORETICAL DEVELOPMENT OF DEFORMABLE BARRIER TESTS WHICH ACCOUNT FOR COMPATIBILITY

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ABSTRACT

The aspect of compatibility between large and small cars has become a main topic in recent accident research. The likelihood of collisions between vehicles of considerable different size is triggering latest discussion on new legal requirements and test procedures.

Computer modelling applicable in crashworthiness has advanced substantially due to the availability of mature software and powerful hardware. Design guide lines derived from theses calculations are no longer restricted to A-B-comparisons and rough trend analyses. This paper describes a compatibility study based on full scale finite element and lumped mass model calculations involving models exceeding 140000 elements.

A head-on car-to-car offset crash is compared to a car-to-deformable-barrier simulation. The cars have a weight ratio of approximately 1.5:1. We describe the efforts of obtaining a single deformable barrier which gives similar deformation and deceleration profiles for both vehicle as in case of the car-to-car collision.

INTRODUCTION

During recent years one can observe a very vital discussion on the appropriateness of certain frontal impact test procedures. While to date all legislative regulations in effect prescribe impact test procedures against rigid barriers the so-called Offset-Deformable-Barrier (ODB) configuration has received considerable attention as a potential candidate for the next generation legislative test procedure (ref. /1/).

Given the fact that a frontal collision in a real world situation is less likely to occur against a rigid obstacle than against another vehicle (ref. /2/) a barrier which is deformable by itself seems to be appropriate to supplement or even substitute current test practice. Defining the exact configuration, however, has proven to be a challenging task. While the experience with rigid barrier tests is relatively large, little is known about deformable barriers. Furthermore, a new test procedure must provide substantially better possibilites to assess a vehicle structure with regard to its protective capabilities and must be clearly a step forward in terms of a better representations of real world collsion situations. Otherwise it would not be worth the effort.

While at first glance the ODB is aimed to mimic similar deformation and deceleration pattern as a car-to-car head-on offset crash of vehicles of the same type it is envisaged that the ODB should be able to assess the vehicle structure with respect to its compatibility.

Since European legislation has abondoned the Step I of the new frontal impact test procedure where the so-called 30°-ASD (Anti-Slide-Device) barrier was planned for use, the pressure on keeping the introduction date of the previously called Step II procedure employing an ODB has increased, thus leaving little time for implementation of advanced compatibility aspects which have been considered so far only to a very limited extent.

There are several aspects of compatibility. One aspect of compatibility is what one might call ''Stiffness Distribution Compatibility''. A possible consequence of an unevenly distributed stiffness can be a front member penetrating the deformable barrier and, therefore, not participating in the overall energy absorption. The result is a heavy loaded passenger compartment. Such a behavior may be also the result of poorly connected load paths. The deformable barrier will be able to detect such a defeciency. An even distribution of stiffness will conseqently result in a better energy absorption in case of a head-on crash. However, a stiffness distribution of vehicle frontal structure is never very homogeneous neither are frontal structures of different cars neccessarily similar, i.e. they usually do not match. Therefore, the evaluation of different frontal vehicle structures by a single barrier must remain a quite crude assessment.

In side collisions compatibility should try to match stiff areas in the front end of one vehicle and stiff areas in the side of the other vehicle. This is particularly true since vehicle front structures have become, in general, much stiffer due to more demanding market driven requirements (ref. /3,4/), which has consequently in turn generated higher stiffness requirements for vehicle side structures. In addition the location of greatest stiffness of the impacting vehicle should be low, i.e. in areas where the direct load is less likely to cause the injuries to the occupants of the impacted vehicle. This leads to the conclusion that lower load paths like subframes should be employed to the largest possible extent. This is particularly true for large and heavy vehicles. A direct important consequence is that the ODB design must favor the use of lower load paths.

The third aspect of compatibility deals with the mass aggressivity. This is the topic we will discuss in detail in the course of this paper. Our plan is as follows: in the next section we will outline the basic mechanics of the mass aggressivity. In the following section we develop simple lumped parameter models based on full scale finite element calculations to investigate various parameters of an ODB. This section is concluded with development guide lines for a deformable barrier. The last section will end the paper with some general conclusions.

BASIC MECHANICS OF MASS AGGRESSIVITY

For the sake of simplicity we consider a frontal head-on car-to-car collision as a one-dimensional problem as follows (Fig.1):

Figure 1

The equation of momentum for this simple model writes:

$$m_1 V_1 + m_2 V_2 = (m_1 + m_2) V^* \qquad (1)$$

V^* is the common velocity of the two vehicles at the time of the closest distance of the centers of gravity, i.e. immediately prior to rebound. We will consider throughout the rest of the paper that both vehicles will have the same initial velocity at the time of impact (but with opposite directions):

$$V_2 = -V_1 = V_0 \qquad (2)$$

We can then solve for V^*:

$$V^* = V_0 (m_1 - m_2) / (m_1 + m_2) \qquad (3)$$

If both vehicles have the same mass ($m_1 = m_2$) then V^* is zero. In case of the lighter vehicle V^* accounts for the additional ride down velocity for the lighter vehicle, i.e. the overall ΔV for the small vehicle m_2 is:

$$\Delta V = V_0 + V^* \qquad (4)$$

If the car-to-car collision has to be simulated by a barrier test this additional velocity has to be considered to have an equivalent ride down profile. As an example if we consider two vehicles with a mass ratio of 1.5 and a impact velocity of V_0 equal 50kph V^* would be equal 10kph. The fact that the heavier vehicle causes a larger ΔV for the lighter vehicle is what we call mass aggressivity.

LUMPED PARAMETER STUDY ON AN OBD

The development of an ODB has been pursuit so far by running car-to-car offset crash tests and adapting overlap and impact velocity and to some extent a few physical parameter of the barrier of a corresponding car-to-ODB test in such a way to obtain a similar deceleration profile and deformation pattern as in the car-to-car crash test. These tests involved cars of the same type and mass thus leaving the aspect of compatibility aside.

In this section we outline a development procedure for an ODB which accounts for compatibility in terms of mass aggressivity: from a car-to-car head-on offset crash simulation of vehicles of different size and mass we derive simple lumped parameter models which serve as a basis to develop a characteristic of a barrier which will yield the same deceleration profiles and hence occupant loads as the car-to-car crash simulation. Based on this force-defelction characteristic an ODB model will be developed.

We consider a car-to-car head-on offset crash between a large vehicle and a small vehicle. The vehicles are represented by detailed finite element models. The large vehicle consists of 76000 elements the small one is built up by 66000 elements. The overlap for the large vehicle is 45% and for the small one is 50%. The initial velocity is 50kph for both. The mass of the vehicles is 1652kg and 1073kg, respectively. Figure 2 and 3 show the undeformed finite element models of both cars. These models have been used throughout the development process extensively and can, therefore, be regarded as verified to the largest possible extent.

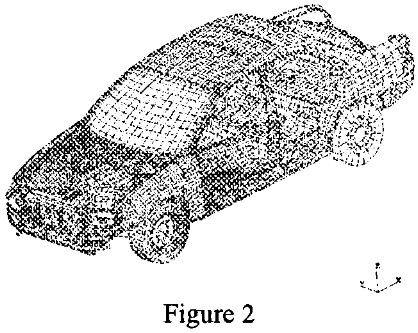

Figure 2

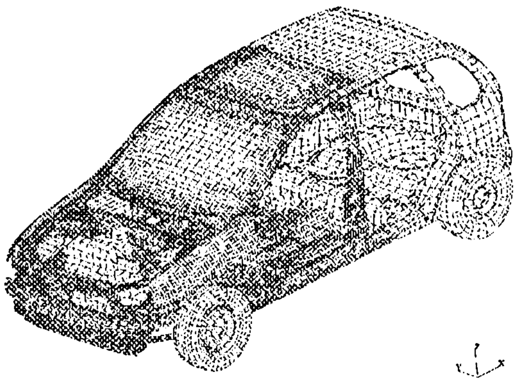

Figure 3

Figure 4 depicts the deformation at the time of rebound:

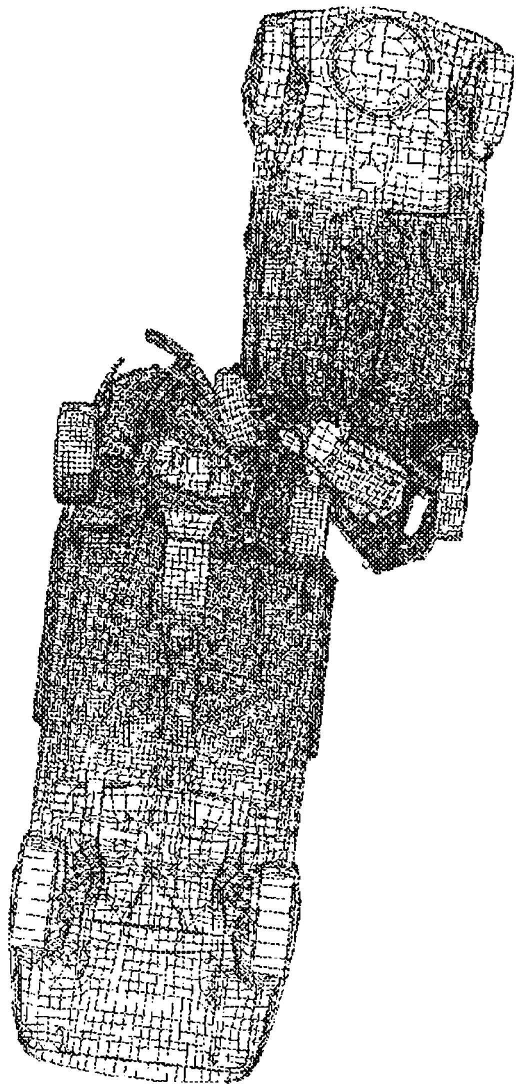

Figure 4

To simplify the investigation both vehicles will be substituted by a very simple model: each car is replaced by one elasto-plastic spring and a mass. To obtain a force deflection curve for each vehicle we run a full scale analysis for each vehicle against a rigid wall with an angle of impact and an overlap such to obtain a similar deformation pattern as in the car-to-car case. From these analyses we extract force-deflection curves, i.e. wall force over vehicle motion measured at the tunnel. The lumped models will be used as follows: first we check the validity by comparing the ride down of the lumped model with those in the full scale model in case of the car-to-car. Figure 5 shows both runs in one plot.

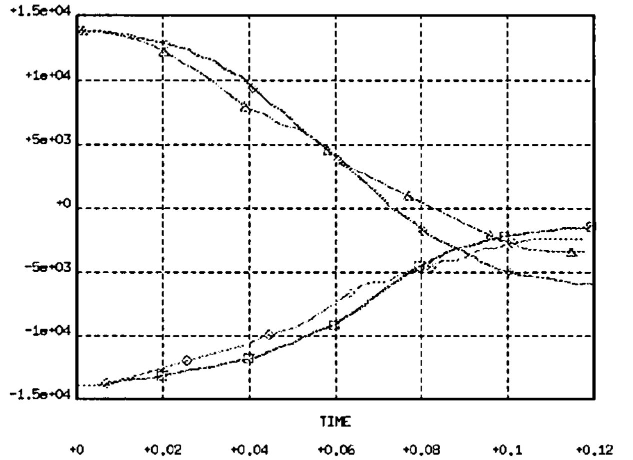

Figure 5

Both velocity profiles are similar. Although, the curves could be matched somewhat better we consider the achieved accuracy as high enough to proceed with our study, since it is only demonstrative nature. In the next step we consider a lumped model simulation where both vehicle are impacted against a deformable barrier represented by an elasto-plastic spring. The initial velocity of both vehilces has been set according to eqn. (4), i.e. the velocity of the small vehicle has been increased by 10.6kph and that of the larger one model has been lowered accordingly.

We consider the charcteristics, i.e. the load-deflection curve of the OBD as very simple (Fig. 6):

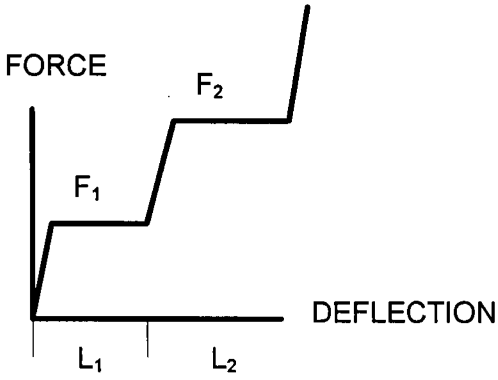

Figure 6

The barrier has two plateaus F_1 and F_2 which expand over L_1 and L_2, respectively. The physical barrier which is to be simulated by this function consists of a main body with a certain crush strength. This crush strength is determined by the force F_2 and the effective area of overlap. The depth of the barrier is equal to L_2. The barrier has a bumper attached to it at the front. The height ratio of bumper and barrier effective height determines F_1. The bumper depth is equal to L_1. A schematic side view of the barrier is depicted in Fig. 7.

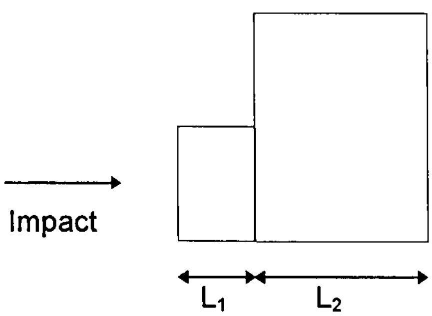

Figure 7

Of course more elaborate configurations could be considered but this for the sake of simplicity and due to fact that similar barriers are in fact currently considered for application we will restrict ourselves to this set of parameters.

We choose to match the deceleration profile of the small vehicle car-to-barrier with the car-to-car simulation by tuning the barrier parameters accordingly. However, it seems to be impossible to match both deceleration profiles - car-to-barrier -, i.e. for the small and the large vehicle employing only one set of barrier parameters. This appears to be quite obvious since the large vehicle experiences a low stiffness from the small vehicle compared to its own stiffness and vice versa. Therefore, it seems to be unfeasible to create a load case suitable for various vehicle types and consistent velocity ride down, i.e. a single barrier which accounts for compatibility. However, it should be possible to match the deceleration profiles if we scale the barrier stiffness for the large vehicle by a factor, i.e. we multiply F_1 and F_2 by a constant. This can translated into the physical model by variing the effective area, i.e. adapting the amount of overlap. Fig. 8 depicts the deceleration profiles of the lumped parameter models for the car-to-car and the car-to-ODB case. The two car-to-ODB profiles match quite well with the corresponding car-to-car profiles.

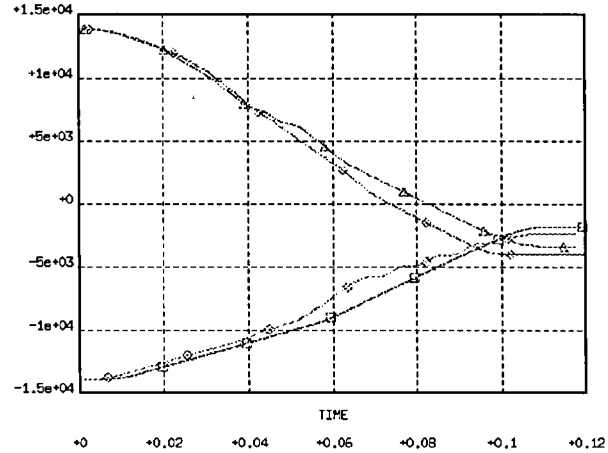

Figure 8

The final step is carried out by translating the parameters of the ODB of the lumped model into a physical model. The depth of barrier main body and bumper correspond to L_2 and L_1, respectively. The height and the width of the barrier is set to 650mm and 1000mm, respectively. An overlap of 40% for the small vehicle is chosen to mimic a similar deformation behavior as in the car-to-car simulation. The crash strength is then finally set to 100psi which yields an effective crash area of $0.36m^2$. The following table depicts the physical parameter of the deformable alluminum honeycomb barrier:

Bumper depth	90mm
Barrier depth	200mm
Barrier width	1000mm
Barrier height	650mm
Bumper height	330mm
Crush strength barrier	100psi
Crush strength bumper	250psi

The overlap ratio between the small and the large vehicle is set equal to the multiplier for the loads which have been obtained from the lumped parameter model. The full scale finite element simulation carried out with the small and the large vehicle yielded deceleration profiles as shown in Fig. 9.

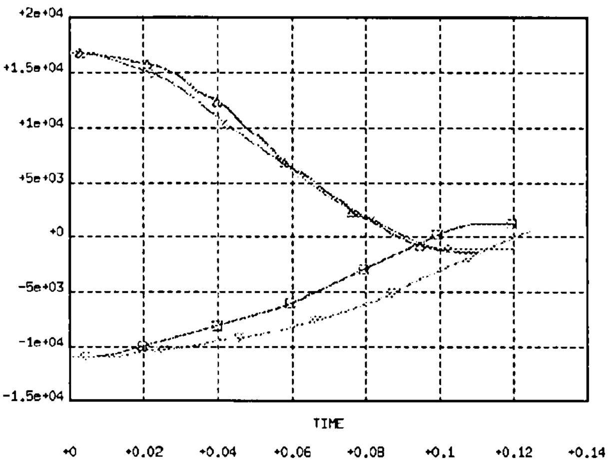

Figure 9

There is a remarkable deviation between the lumped parameter model and the finite element model calculation for the large vehicle but a very good agreement for the small vehicle. Several trial runs with different amounts of overlap were conduction in order to improve the behaviour for the large car but none of them were successful. This leads to the conclusion that there are local deformation effects present which are not captured by the lumped parameter model. However, having derived the lumped parameter model from rigid wall full scale simulation with a rough adaption of overlap an angle of impact to match the car-to-car deformation pattern and reproducing the deceleration profiles of the car-to-car full scale simulation with the very simple spring model but failing to do so for the ODB - at least for the large vehicle - prooves the ODB more sensitive for particular package arrangements. This can be interpreted as a drawback of the deformable barrier compared to the rigid barrier.

CONCLUDING REMARKS

The study has shown that computer simulations using validated full scale finite element models can be used to derive parameters for barrier developments very rapidly.

The method for developing a deformable barrier for conducting crash tests is quite straightforward: take a car-to-car crash test simulation and substitute one vehicle by the barrier. Then adjust initial velocity, overlap and barrier parameters in a way to match deformation pattern, ride down profiles, and absorbed energy. However, this can even easier be obtained with a rigid barrier which is demonstrated by current practice. The deformable barrier can in addition account for deformation of the opponent vehicle. To account for compatibility the barrier must offer even more advanced capabilities. One aspect is to employ the same barrier for different vehicles which complicates the derivation of an appropriate set of parameters for the barrier as demonstrated in this study. One advantage of the ODB is that it favors a frontal design with a good load spreading. However, it can do even more. In a car-to-car crash the lighter vehicle experiences a larger ΔV compared to the initial velocity. To have a similar ride down compared to a test with vehicles of the same type the opponent vehicle must provide additional deformation in order not to increase the average deleration level. This poses constraints on the design of the front structure of heavier vehicles, i.e. these vehicles must exhibit a low enough stiffness in the event of the collision with a smaller vehicle to yield the required deformation length. Such a characteristic, in turn, could be assessed by a deformable barrier.

In conclusion the ODB can offer advanced features for assessing the protective capabilities of passenger cars including compatibility aspects. However, the approriate design of such a barrier is a very complex task and can by far not be regarded as completed. Computer simulations are viable aids for such a development which has been demonstrated once again in this paper. It will be necessary to employ these methods to meet the required quality and the ambitious timing.

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EVALUATION OF CRASH COMPATIBILITY OF VEHICLES WITH THE AID OF FINITE ELEMENT ANALYSIS

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Paper Number 96-S4-W-25

ABSTRACT

With increasing safety awareness on the part of vehicle users and the constant improvements in passive safety design during recent years, the question of the compatibility of vehicles in collision with other road users has gained in importance.

After the definition of compatibility and a statistical accident analysis, this article deals with the potential of computer simulation as an aid to development.

Evidence is presented to show that correct results can be obtained by computer calculation of the EEVC side impact with an Audi A4 and EuroSID dummy. Further results are calculated for a car-to-car crash between a SEAT Ibiza and an Audi A4 with occupants.

The article concludes with a summary of the advantages of this method and an outlook for possible approaches in future.

INTRODUCTION

In addition to the aviation and space industries, the motor vehicle industry is among those in which product development depends primarily on computer-aided calculating and simulation procedures.

It is therefore no accident that these branches of industry in particular have made the strongest contributions towards the development of computer-aided simulation processes. Just as aircraft construction stimulated the use of the Finite Element Method (FEM), so solutions for the description of structures with non-linear effects taken into account were found in motor vehicle construction. By means of Computer Aided Design / Computer Aided Engineering / Computer Aided Manufacturing (CAD/CAE/CAM) systems, it becomes possible to convert physical models efficiently into mathematical ones and to resolve them at the prevailing level of product complexity. Engineers therefore have methods at their disposal which relieve them of routine work and provide them with scope for more creative activity.

Whereas the "trial and error" method was predominant in the past, that is to say product development made use of complex and time-consuming experimental procedures, the definition and development of a vehicle today have to be largely complete before prototypes are built. Subsequent testing is solely intended to confirm that the functions operate correctly and to coordinate various systems.

The introduction of computing methods has become of particular significance in the rating of vehicle structures for crash safety. A high level of passive safety, that is to say the protective function exerted by the vehicles in the event of an accident, is obtained by arranging for the bodyshell to deform in a planned manner, with a rigid occupant cell to act as a survival zone and the use of protective systems such as seat belts and airbags.

In addition to protection of the vehicle's own occupants, which has reached a high standard on account of the relevant legislation (e.g. FMVSS 208) and rating procedures (e.g. NCAP), increasing significance is being attached to protection of the other party involved in an accident. With declining tendency in the countries of the European Union (EU), some 2900 people are killed in frontal car-to-car accidents and 2200 in car-to-car side-impact accidents.. If measures could be taken on vehicles to ensure that no occupants were killed in frontal car-to-car accidents with a speed difference below 100 km/h, or in side-impact accidents with an impact speed lower than 50 km/h, this would mean 675 fewer fatalities per year.

The VW Group offers vehicles in a wide variety of size categories for sale. There is accordingly much motivation to investigate the important topic of compatibility between different vehicles However, analysis of car-to-car accident interactions involves particularly complex experimental effort and expense. This paper indicates how computer simulation can be adopted in order to obtain a realistic analysis of these highly complex occurrences. Calculation of variants and extensive optimization calculations can be conducted in this way, in order to determine the principal compatibility parameters and introduce improvements on vehicles.

COMPATIBILITY AND DATA ANALYSIS

When a single-vehicle accident occurs, injury of the car's occupants depend on the collision mode, ón the impact velocity, on the personal condition of the occupant and on the inherent safety of the car. The inherent safety of the car is the ability to avoid injuries to occupants, when a collision mode at a specific velocity occurs. This ability is measured by vehicle behavior and interpretation of the dummy kinematics and load. In literature the inherent safety of a car is called occupant protection.

In a car-to-car accident, collision modes, impact velocities, and the personal condition of driver and occupant can be observed in the same manner. But injury to the car's occupants does not only depend on the inherent safety of the car, but also on the structural behavior of the other car. It is obvious that an accident between a heavy and a light car may be different from an accident between two light cars. An accident between a rigid and a yielding car might be different to an accident between two yielding cars. The distribution of the stiffness at the front of a vehicle might influence the interaction between the structures of the two colliding vehicles. In car-to-car accidents we therefore also study partner protection, the ability of the other car which collides with the car under examination to avoid injuries in the car which it strikes. This ability is called partner protection. Lack of partner protection is sometimes referred to as aggressiveness.

The goal of compatibility is to bring these two issues together, to enhance partner protection without decreasing occupant protection or to optimize occupant protection in such a manner that the overall safety of the vehicle is maximized.

Figure 1 shows the situation as identified in the Volkswagen accident data base. This is a data base of 8566 vehicles (53 % are not Volkswagen or Audi models).

The vehicles were divided in 6 groups: A0 such as the VW Polo, Opel Corsa etc., A such as the VW Golf, Ford Escort, B such as the VW Passat, Audi A4, MB C-Class, C such as the Audi A6, BMW 5xx, MB E-Class, D such as the Audi A8, Volvo 7xx, MB S-Class and Bus - MPV, comprising minibuses like the VW Caravelle and similar products. Accidents between those types of vehicles are checked and it is registered what the striking car was when the belted driver of the struck car was injured. All collision modes are included. The data are checked for all belted drivers, injured and not injured, and for the different MAIS classes.

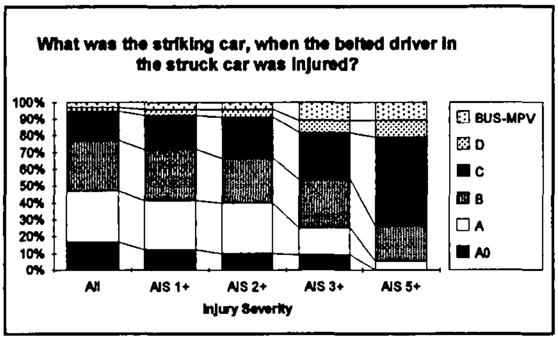

Figure 1. Distribution of the size-groups of the striking vehicles when a belted driver in a struck vehicle was injured.

It can be observed that MAIS 5 and 6 occurs to a level of more than 90 % in crashes when a B-, C-, D-, or bus-type vehicle strikes another vehicle, although these groups are less than 55 % of the striking vehicles for all drivers. This indicates that there is a need to study the structural interaction of vehicles of differing masses.

Figure 2 shows how many occupants are involved in accidents when a vehicle of a certain size-group is involved. It counts all the people involved: those sitting in the vehicle in question, those sitting in the other vehicle, the struck vehicle and those hit by the vehicle as pedestrians or cyclists. The occupants of a struck vehicle that is not a passenger vehicle are counted as occupants of other vehicles. This figure shows that per involved vehicle of the size-group under consideration there are no differences between the size-groups when the number of involved people is considered.

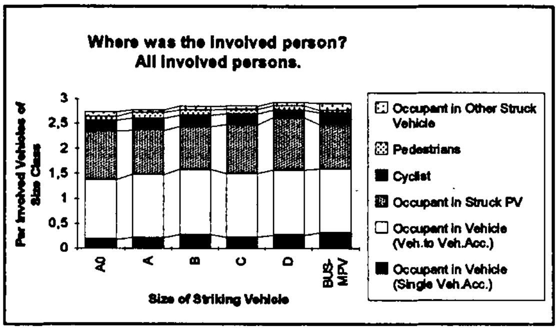

Figure 2. Involvement in an accident of a vehicle of the size-groups A0 (small) to D (luxury) and Minibus.

This changes completely when the same figure is computed for severely injured persons. Figure 3 shows that the rate is then completely different between the size-groups.

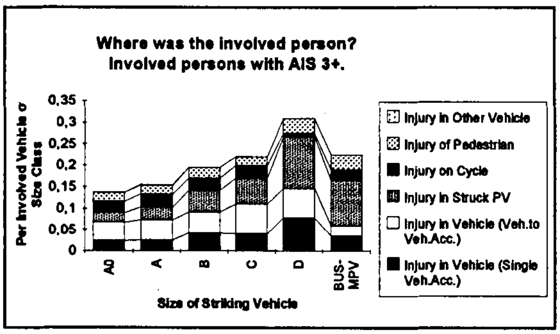

Figure 3. Severe injuries in an accident involving a vehicle of the size-groups A0 (very small) to D (luxury) and Minibus.

For the D class, we have a maximum in all rates. So when a D-class vehicle is involved, more severe injury must be expected.

Another interesting influence of structural stiffness can be identified from theoretical studies which Volkswagen already presented to a former ESV conference: Frontal impact involves a certain amount of deformation energy in the front structure. Since deformation energy is the product of deformation travel and average force level, increased frontal safety may for some vehicles also include higher force level in the frontal structure. This calls for higher force level in the side structure of the struck car in order to maintain the safety standard in the event of side impact. Fig. 4 shows the change in intrusion due to the stiffness of the front structure of the striking vehicle.

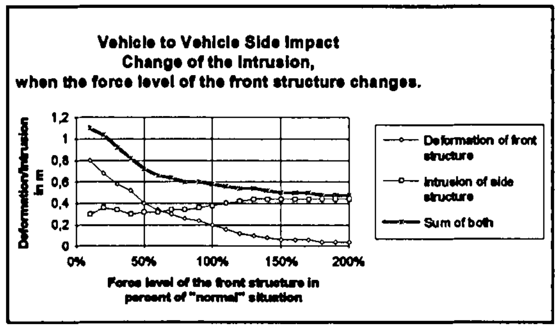

Figure 4. Computer simulation of car-to-car side impact.

The influence of the force on the intrusion of the side of the struck vehicle can clearly be seen. Fig. 5 shows the consequences. The higher stiffness of the front structure requires a higher stiffness of the side structure structure in order to maintain the safety standard. Otherwise the dummy load in the struck vehicle will increase and thus the hazard to the occupant of the struck vehicle. So stiffer front structures may compensate for the effect of the changes in side protection.

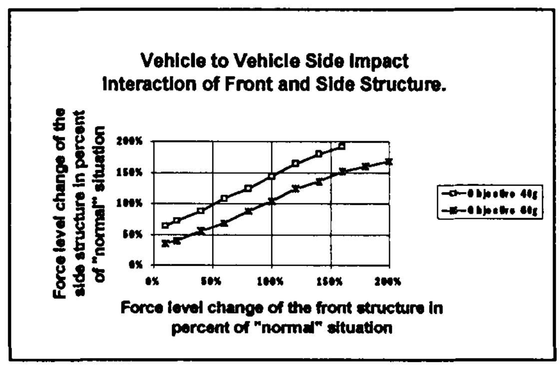

Figure 5. The higher aggressiveness of stiffer front structures can only be compensated for by stiffer side structures.

This and other observations were the reason for Volkswagen, Audi, Seat and Skoda to form an internal expert group, to study vehicle compatibility phenomena. Car-to-car side impact is an interesting area for such research and from the arguments mentioned above, we expect a lot of advantages for compatibility in side impact. But side impact and all kinds of car-to-car impact cost a lot of money, especially when the tests have to be conducted on prototypes. The ability of computer simulation to act as a substitute for actual car testing therefore has to be studied.

NUMERICAL EVALUATION OF CRASH COMPATIBILITY

Numerical evaluation of the complex occurrences in a car-to-car accident calls for the correct modeling of all relevant assemblies and part-systems. By way of an example, this paper analyzes the behavior of an Audi A4/Seat IBIZA side-impact crash with regard to compatibility. For this purpose, the simulation model is built up as follows:

1) Modeling and validation of the Audi A4 side structure with the EuroSID dummy, including the door trim and door padding, in a side crash according to EEVC conditions.
2) Checking the Seat IBIZA frontal crash model by means of a 55 kph „auto, motor und sport" offset-test.
3) Combining the models and calculating structural behavior and EuroSID loadings in a car-to-car crash in which the Seat is driven at a right angle into the side of the Audi in accordance with EEVC experimental conditions.

EEVC Side Impact with an Audi A4

The explicitly non-linear PAMCRASH finite-element program is used for structural crash simulations. Dummy loads are usually calculated very quickly using multi-body programs such as MADYMO, although it is necessary to satisfy various prerequisites and assumptions.

The number of unknown quantities for describing the same problem rises due to the high expectations and high levels of precision to be satisfied by the calculation results. The model size is again increased by allowing for an FE dummy, the non-load bearing door components and all the interactions during the crash. Therefore for parameter studies it is necessary to simplify the model. Only as many unknown quantities as are actually required should be introduced. This reduction is possible by introducing simple elements and analytical solutions in subsections of the structure. Another advantage is that this approach offers clarity and comprehension concerning active interrelationships during the crash.

Simplified structural model - Figure 6 shows the simplified model of the Audi A4.

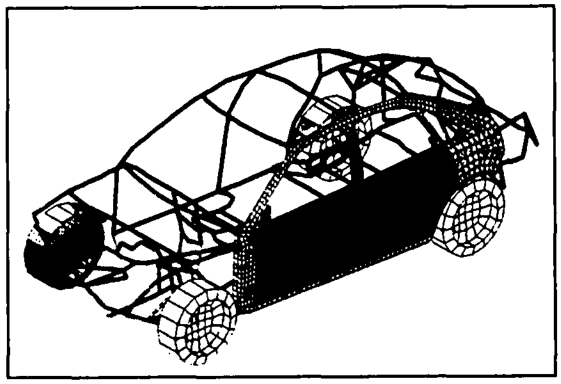

Figure 6. Reduced Model of the Audi A4-Structure.

The contact area on the left side is unaltered as a shell structure whereas the right side and the front and rear areas away from the contact zone can be defined as rigid bodies connected by plastic beams.

Figure 7 shows the structural deformation of the outer door panels of the simple and of a complete FE-model at different moments in time.

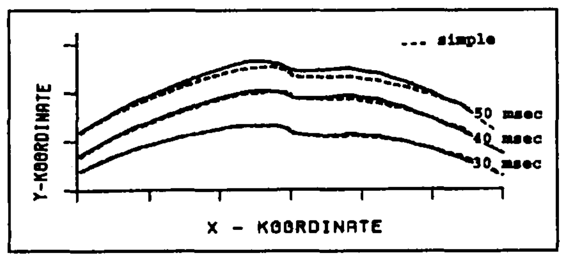

Figure 7. Deformation of the Structure. Comparision: Simple Model - Complete FE-Model.

It shows a good degree of agreement. In the simplified model, computation time is reduced by about 70 % and is therefore highly suitable for fairly rapid validation calculations.

Components such as door trim and side padding did not need to be taken into consideration in the evaluation of structural behaviour. The door components do however have to be registered during simulation if it is intended to make statements concerning occupant loads.

Door trim - The door trim is in direct contact with the dummy. The surface geometry, the edges of the transitional areas and attachment to the structural parts behind them can considerably influence occupant loads, particularly in the area of the ribs. Other crash parameters include properties of the materials which are used to construct the door trim.

To record the dynamic door trim behaviour, a special test device was set up in which all the other door components were rigidly clamped. The dummy's contact areas were then subjected to static and dynamic loads using a half ball. The appropriate types of load were simulated. Figure 8 shows the results.

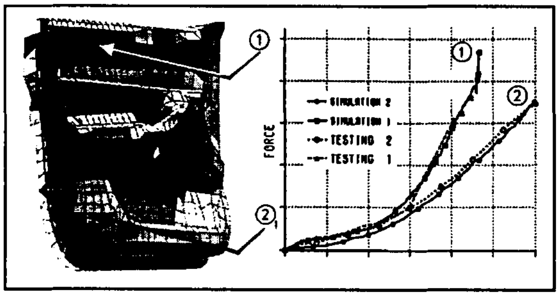

Figure 8. Characteristics of the Door Panel. Contact Areas of the Dummy 1: Ribs, 2: Pelvis.

There was a good degree of agreement between the calculation and the experiment. The door trim has a linear material property, but no dependency on the strain rate was found.

<u>Side padding</u> - Side padding is used to delay the impulse transfer from the penetrating structure to the occupant, thus accelerating the occupant more gently. Unfortunately there is currently no material which would provide an optimum result for every load speed. It is therefore necessary to make compromises; the material's properties ought to be dependent on the distortion rate. The padding material's properties are ultimately coordinated on the basis of experiments or calculations while evaluating dummy loads.

A special test device was prepared for determining the correct parameters during the simulation. The foam samples were subjected to loads at different speeds. In Figere 9 the results of the pendulum experiment are superimposed at an impact speed of 8 m/s.

The results from the simulation agree well with the test values, both during the load phase and during the phase in which the padding was relieved of load.

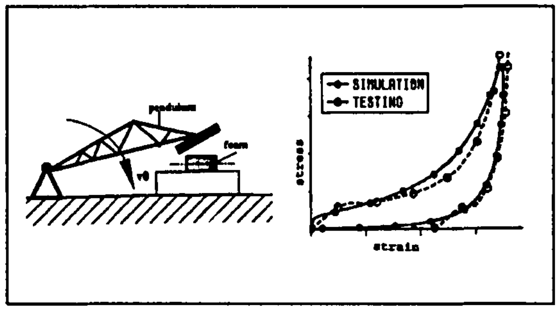

Figure 9. Pendulum Test of Padding Samples. Comparision: Simulation-Test, v=8 m/s.

Calculations were then made on the basis of the set material parameters for the pelvic padding using complex geometry. The deformation rates were varied. The results are shown in Figure 10, in which the strain rate dependency can be clearly identified.

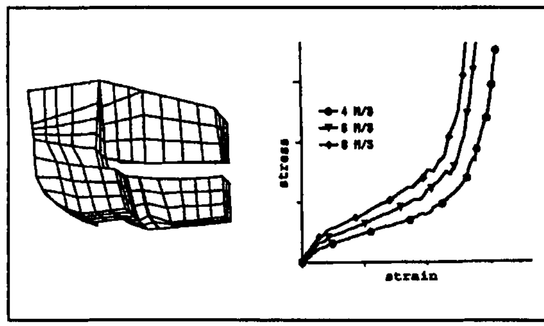

Figure 10. Pelvis-Padding and Characteristics.

<u>Dummy model</u> - Standard dummies are used to evaluate the severity of occupant injury during a side impact. Dummy loads must not exceed the legally stipulated limits. The side-impact measures taken are evaluated inter alia according to the injury criteria. The side-impact dummies based on the planned ECE regulation and US standards not only differ in their behaviour but also in the various evaluation criteria. The ECE dummy - the EuroSID - has additional measuring points. Forces and paths are analyzed in the ECE dummy, whereas in the US version the accelerations are evaluated.

The EUROSID1 dummy model from TNO is used in the simulation based on the MADYMO multi-body program. Currently no dummy models validated for all types of load are available for numerical calculations using explicit FE codes.

A model is currently being developed for the EuroSID at ESI. AUDI has received a version of this dummy model, Figure 11.

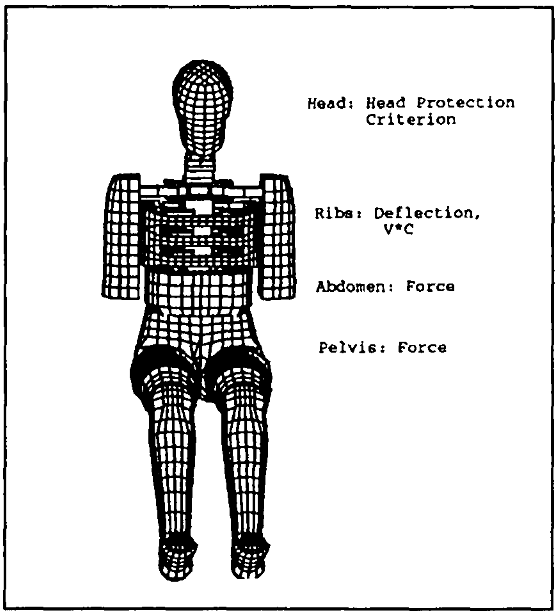

Figure 11. FE-Model of the EuroSID-Dummy.

A series of improvements, particularly in the pelvic area, has been incorporated. The validations were performed using the AUDI sled tests for doors.

Build-up of a complete side impact model - Once all the subcomponents were validated to an adequate degree of accuracy, they were combined into an overall system, taking account of all the contacts and attachments, Figure 12.

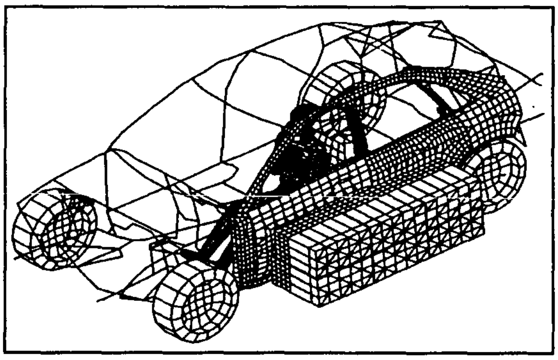

Figure 12. Side Impact Model of the Audi A4.

As all the major components in the model are taken into account, comparisons can be drawn with the experiment. The dummy loads of pelvic force, abdominal force and rib deflection are shown in Figure 13.

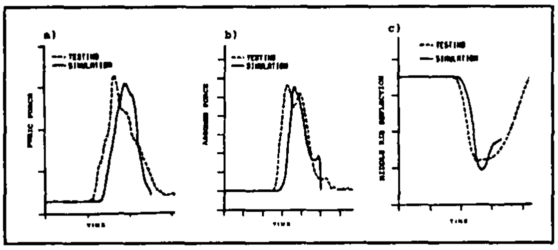

Figure 13. Loads on the Dummy. Comparision: Simulation - Test. a:Pelvis Force, b:Abdomen Force, c:Rib Deflection.

During the simulation, records of the timing patterns from the experiment largely came out well. The contact times differ to an extent, causing the patterns produced by the experiment and the calculation to shift in time. The reason for this shift is probably the divergent dummy positioning during the experiment.

Frontal Crash with a Seat Ibiza

The preliminary structure of the Seat Ibiza FE model includes the power unit, running gear and other assemblies such as the radiator. Figure 14 shows the deformation of a validation simulation in accordance with the german „auto motor und sport“ test procedure.

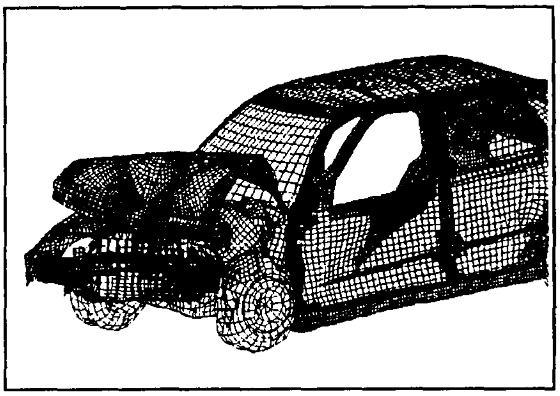

Figure 14. ams-Simulation of the Seat Ibiza.

Since both the deformation pattern and vehicle deceleration coincide well with the experimental results, it is possible as the next step to combine the Audi A4's FE model with that for the Seat Ibiza.

Analysis of the Compatibility A4/Ibiza

After the interaction of all part-systems has been coordinated by means of the simplified Audi A4 side-structure model (Figure 12), the car-to-car side crash compatibility calculations are undertaken with the full Audi model. This ensures that in the complex car-to-car crash load situation the reduced FE structure model cannot introduce any errors. The complete model, Figure 15, with both vehicles and one EuroSID in the Audi A4, consists altogether of approximately 110,000 shell elements.

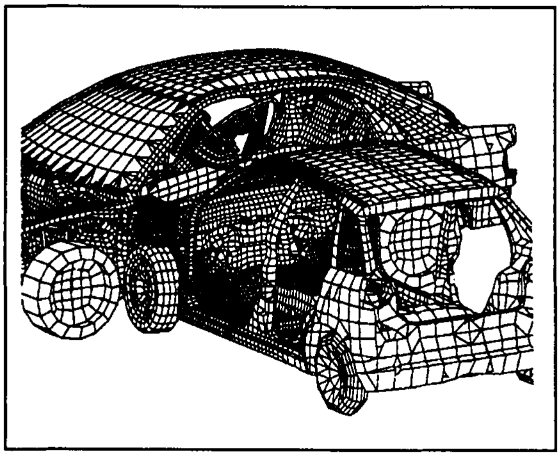

Figure 15. Complete FE Model for Analysis of the Car-to-Car Compatibility Crash.

The following conditions were chosen:

- EuroSID in midway seat positions (forward-back and height adjustment)
- The mass of the Audi A4 is 1498 kg
- The mass of the Seat Ibiza is 1392 kg
- The center of the Ibiza's front-end structure strikes the R-point of the A4 at 90 degrees
- The Ibiza strikes the stationary A4 at 50 km/h.

DISCUSSION

In order to evaluate the loads on the occupants, an experiment was carried out with precisely the same peripheral conditions as in the simulation. Figure 16 shows the deformation pattern. In view of the extensive validation already carried out, combining the various part-models presented no problems.

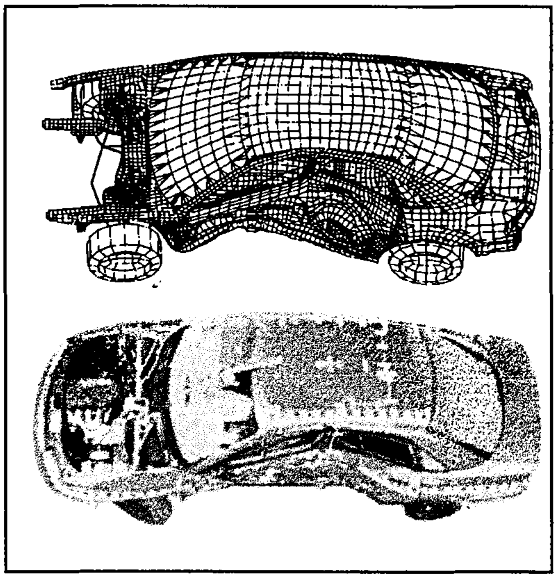

Figure 16. Deformation pattern Simulation/Test.

An extract from the resulting values is shown in Figure 17.

		Limit	Test	Calculation
HPC	Head Performance Criterion	< 1000	85	131
RDC upper/middle/lower rib	Rib Deflection Criterion	< 42 mm	36/32/34	38/29/23
VC upper/middle/lower rib	Viscous Criterion	< 1 m/s	0,62/0,50/0,49	0,83/0,54/0,40
PSPF	Pubic Symphisis Force	< 6 kN	-	4,9
APF	Abdominal Peak Force	< 2,5 kN	1,2	1,7

Figure 17. Comparison between experimental and calculated EuroSID loads (Audi A4).

The maximum values for all body areas were distinctly below the proposed limits in the planned ECE legislation. Both the elapsed time and the maximum values in the calculation and the experiment coincided very well.

Comparison of the HPC head values shows that the calculated value is about 50 % higher than the experimental result. This is caused by the FE dummy's head just touching the door frame, where it is just clear of it in the experiment. However, the value is in any case so low that even in the simulation the head deceleration value can be regarded as non-critical.

With regard to structural deformation, the simulation reveals the same tendencies as the experiment (Figure 16). Intrusion is very uniform, an indication of the strong side structure of the Audi A4 and the good behaviour of the Ibiza's frontal structure. A difference is evident in the sill area. During the experiment the Audi A4's rear door became snagged in the sill, so that it was almost entirely prevented from sliding over it. The same snagging effect cannot be identified in the calculation. In this case, more precise modeling of the relevant contact zones is needed, and also a more accurate representation of the Seat's tire, in order to reproduce this effect realistically.

CONCLUSION

The work presented here shows that calculation of car-to-car accidents as a means of investigating compatibility is a tool capable of analyzing deformation behavior and the resulting loads on dummies in the preliminary development phase for various vehicle structures.

However, the limits of this method should be borne in mind. This applies primarily to the dummy, which must be examined in particular detail. It is for example currently impossible to analyze an FE dummy secured by a seat belt, as the relevant model is still being compiled. Since the seat belt influences the pattern of dummy movement, it is possible that the calculated load values will also change slightly.

Friction between dummy and door trim not only influences loads on the dummy but also the reverse effect exerted by the dummy on the structure. This effect still has to be investigated in detail. In the same way, the dummy is supported by the seat through its foam upholstery, but the simulation does not take this into account. As the above report indicates, special attention has to be paid to the contact areas. For example, detailed modeling is needed to take the snagging and interlocking of various elements into account. This is also true in this context of the behavior of welded and adhesive joints and of crack formation in sheet-metal panels.

However, it cannot be the task of calculation to simulate reality absolutely authentically, since even experimental results obtained in the same general conditions can suffer from quite considerable fluctuations on occasion. This is particularly true of experiments involving such complex measuring instruments as dummies. The great advantage of computer simulations is the posibility to conduct many optimizations with investigating only one parameter of the system.

With the aid of the method described here, the structures of various vehicles will be studied intensively in coming months, and parameters for effective partner-protection analyzed. In addition to the coordinated rigidity of the accident participants' front and side structures (Figure 5), detail solutions are those which ensure that vehicles behave in a compatible manner. For example, it is essential for a strong cross-structure to be retained at the front, in order to prevent aggressive behavior on the part of the side-members. Similarly, a coordinated deformation characteristic in the front structure is plausible, in order to ensure a high level of partner protection, but also good protection for vehicle occupants in a one-car accident. Precisely these in-depth investigations make computer simulation essential, since it permits each individual parameter to be investigated and optimized separately.

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Review of Occupant Protection In Light Commercial, Off Road and Forward Control Passenger Vehicles

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Paper Number 94-S4-W-26

ABSTRACT

Off-road passenger vehicles and light commercial vehicles are becoming an increasing proportion of the Australian passenger vehicle fleet.

Consequently, a review of the level of occupant protection provided by these vehicles was commenced in 1992. As part of the review, a further study was commissioned with the Monash University Accident Research Centre (MUARC) to examine the occurrence of injuries to occupants of these vehicles.

In parallel, a review group was set up with industry to explore ways to improve the level of occupant protection provided by these vehicles. This included a regulatory impact statement on the costs and benefits of new ADR requirements.

This paper provides the outcome of that review in general and specifically covers the FORS crash test program which was conducted as part of the standards development program to improve occupant protection provided by off-road passenger vehicles and light commercials.

The test program provides a general indication of the safety performance of these vehicles and in summary, supports the application of ADR 69/00 Full Frontal Occupant Protection to Off-road Passenger Vehicles and Light Goods Vehicles which would result in an improvement in the occupant protection levels. This would also shift the focus towards performance based testing of the vehicle occupant protection package as a whole and provide a level of occupant protection equal to that of passenger cars.

INTRODUCTION

In 1989, the Federal Office of Road Safety (FORS) commissioned a major study to determine the performance of the Australian Design Rules (ADRs) through examination of real life cases and to recommend the improvements that could be made.

The study was carried out by the Monash University Accident Research Centre (MUARC)[2] as part of a review of passenger car occupant protection and showed that despite the improvements in vehicle safety, occupants were still being injured by contact with parts of the passenger compartment.

The outcome of the review was the introduction of ADR 69/00 for full frontal occupant protection[3]. This saw the fitment of airbags in a number of passenger cars when the Design Rule was introduced during 1995.

In recognition that off-road passenger vehicles and light commercial vehicles were becoming an increasing proportion of the Australian passenger vehicle fleet, a review of the level of occupant protection provided by these vehicles was commenced in 1992. As part of the review, a further study was commissioned with MUARC to examine the occurrence of injuries to occupants of these vehicles.

In parallel, a review group was set up with industry to explore ways to improve the level of occupant protection provided by these vehicles. This included a regulatory impact statement on the costs and benefits of new ADR requirements.

This paper provides the outcome of that review in general and specifically covers the FORS crash test program which was conducted as part of the standards development program to improve occupant protection provided by off-road passenger vehicles and light commercials.

CRASHED VEHICLE STUDY

The objectives of the MUARC study were to examine the extent and patterns of injuries occurring to occupants of off-road passenger vehicles and light commercials and ways to address this trauma.

Examination of the mass database indicated that frontal impacts comprised the majority (55%) of crash types and were similar to earlier findings of 60% for passenger cars.

The majority of contacts causing injury were with the instrument panel, seatbelts and steering wheel. These components were also the most common contact sources reported for frontal crashes among passenger cars. As a result, the suggested countermeasures coming out of this study were those suggested from the previous study for passenger cars.

CRASH TEST PROGRAM

The FORS crash test program was conducted as part of the standards development program to improve occupant protection provided by off-road passenger vehicles and light commercials.

In this program, six vehicles were tested - three off-road passenger vehicles and three light commercial utility vehicles.

For these vehicles, crash testing required by the Australian Design Rules, and the regulations enforced in other countries except the USA, assesses rearward displacement of the steering column. In order to shift the focus from component based testing to assessment of the total occupant protection package performance, the series of tests conducted in this program examined the likelihood of injury to occupants using instrumented test dummies. The six tests were conducted according to the requirements of ADR 69/00.

Test Requirements

Injury Parameters - The injury parameters set out in ADR 69/00 were used, viz:

- **Head Injury Criterion (HIC)** measured by accelerometers in the test dummy's head. The value is the maximum cumulative integration of acceleration using the expression:

$$\left[\frac{1}{(t_2-t_1)}\int_{t_1}^{t_2} a\,dt\right]^{2.5}(t_2-t_1)$$

where a is the resultant acceleration expressed as a multiple of the acceleration due to gravity, and t_1 and t_2 are any two points in time during the crash which are separated by not more than 36 milliseconds. The limit specified in ADR 69/00 is 1000.

- **Chest Deceleration** measured by accelerometers in the test dummy's chest. The limit specified in ADR 69/00 is 60g except for intervals whose cumulative duration is not more than 3 milliseconds. In this expression, "g" is the acceleration due to gravity.
- **Compression Deflection of the Sternum** measured relative to the spine by a chest mounted rotary potentiometer. The deflection limit specified in ADR 69/00 is 76.2 mm.
- **Femur Load** measured by load cells as the force transmitted axially through the upper leg. The limit specified in ADR 69/00 is 10 kN.

These injury parameters provide an indication of the likelihood of serious or fatal injuries to vehicle occupants.

In addition to these injury parameters secondary instrumentation was installed to measure base data relating to neck and lower leg moments and forces. This instrumentation consisted of the following:

- One array of three uniaxial accelerometers fitted in the pelvic cavity of both driver and passenger side dummies to measure deceleration in his region;
- One six axis load cell fitted to the neck of both driver and passenger side dummies to measure neck forces and moments in and about the X, Y, and Z axis;
- Two strain gauge load transducers fitted to each driver's knee clevis assembly measuring axial loading in each component;
- One twin axis upper tibia load cell fitted to each driver's lower leg assembly measuring moments about the X and Y axis;
- One tri axis lower tibia load cell fitted to each driver's lower leg assembly measuring axial force in the Z axis, shear force in the Y axis and the moment about the X axis.

Location of Additional Transducers and Load Cells - The following additional measuring equipment was used.

Seatbelt Assembly: Two load cells were fitted to each front outboard seatbelt assemblies to measure belt loadings during impact. These were positioned on the webbing near the outer lap anchorage and the upper sash point.

Care was taken to position the transducers so they did not affect test dummy trajectory, especially in the area of the lower ribs and iliac crest.

Vehicle Structure: A three axis accelerometer was fitted to the vehicle to measure the deceleration pulse of the occupant space. It was placed on the vehicle body at the base of the 'B-Pillar". In the latter of the six tests, an additional single axis accelerometer oriented to measure acceleration in the X-axis was mounted to the chassis rail directly below the body B-Pillar. This was used to assess the difference in longitudinal acceleration between the vehicle body and chassis.

Impact Velocity - The impact velocity requirements of ADR 69/00 are that the test vehicle strike the barrier at 48 km/h. In accordance with ADR 69/00 Clause 5.1.1, vehicles impacted at velocities above 48 km/h are deemed to comply with the rule provided all injury criteria are met.

Test Vehicles - Listed below are the vehicles tested and their corresponding test mass and indication of their drive configuration. They were selected as representative of popular vehicles in these categories.

Test No.	Vehicle Make / Model	Test Mass	Drive
B4026	Mitsubishi Pajero	2158.0 kg	4WD
B4025	Suzuki Vitara	1458.0 kg	4WD
B4029	Toyota Landcruiser	2521.7 kg	4WD

B4028	Holden Rodeo	1641.0 kg	RWD
B4027	Mitsubishi Triton	1584.2 kg	RWD
B4030	Toyota Hilux	1580.9 kg	RWD

RESULTS

Head Injury Criteria (HIC)

The HIC36 values ranged from 709 to 1167 for the driver's side and 471 to 1379 for the passenger's side. (Figure 1.)

Generally the HIC value was lower for the passenger than for the driver. This reflected driver head contact with the steering assembly whereas passenger head contact was minimal in most cases.

Contrary to this trend were the HIC values for B4025. For this test the recorded passenger side HIC was considerably higher than for the driver's side. This was due the test dummy's head striking its lower thigh.

The HIC15 values ranged from 569 to 1150 for the driver's side and 212 to 641 for the passenger's side.

The significant difference observed on comparison of the HIC 36 to HIC 15 figures is the reduction in passenger HIC where either no contact was made or contact was with the dummy's thigh flesh. The passenger HIC 15 value for B4025, for example, is 54% lower than the corresponding HIC 36 while the driver value for the same vehicle is 8% lower.

Chest Deceleration

The chest deceleration values ranged from 43 to 69g for the driver's side and 37 to 54g for the passenger's side. (Figure 2.)

For all but one vehicle, the chest deceleration was greater for the driver than for the passenger. This was generally due to driver contact with the steering wheel.

Compression Deflection of the Sternum

The measurements of compression deflection of the sternum ranged from 39 to 57 mm for the driver's side and 33 to 46 mm for the passenger's side. (Figure 3.)

In all cases, the deflection measurements for the driver were greater than those for the passenger. This is also attributed to driver contact with the steering wheel.

Femur Loads

All measured femur axial loads fell below 6 kN (Figure 4.). There seemed to be no distinct trend indicating whether the drivers side or passenger side exhibited higher average loadings.

In the two vehicles recording the highest loads it was found that the dummy's knees had contacted areas of the dashboard which were supported by mounting bars of some form.

In all cases, however, the measured loads are well below the ADR 69/00 limit of 10 kN indicating that a serious femur injury was unlikely.

Lower Leg Loads

Using Hybrid III instrumented lower legs, axial loading on both sides of each knee clevis assembly, tibia shaft upper and lower moments M_x and M_y and tibia shaft axial loading measurements were made.

Clevis tensile loads ranged from 300 N to 700 N while compressive loads ranged from 600 N to 1900 N. No unusually high loadings were observed.

Tibia shaft maximum moments ranged from 60 Nm to 356 Nm. Vehicle B4025 exhibited maximum moments at the upper end of this range for both left and right lower legs. This observation is coincident with the higher driver side femur loads of the same vehicle discussed above.

Compressive loads on the tibia shaft ranged from 0.8 Nm to 3.8 Nm. Again no unusually high loadings were observed while the magnitude of each load roughly correlated to the clevis loadings as would be expected.

Neck Loads and Moments

The Hybrid III instrumented necks fitted to driver and passenger side dummies enabled the gauging of loads through the neck during the vehicle impact. The primary measurements taken were flexion and extension moments about the Y axis, tensile and compressive axial loadings and fore and aft shear forces along the X axis.

Neck extension moments ranged from 19 Nm to 265 Nm with the maximum figure measured for the driver of vehicle B4025. Neck flexion moments ranged from 13 Nm to 139 Nm with the maximum figure measured for the passenger of vehicle B4025. In most cases the passenger moments for flexion and extension were higher than those of the driver. It was found that there was no substantial alignment between the neck moments and the incidence of high HIC readings.

Neck tensile and compressive forces ranged from 1.9 kN to 4.3 kN and 0.015 kN to 2.0 kN respectively. The incidence of above average tensile neck loading appeared to coincide with an above average (higher) HIC reading though no correlation of HIC data to neck compressive force data was evident.

Neck fore and aft shear forces ranged from 0.1 kN to 0.7 kN and 0.6 kN to 4.8 kN respectively. In all cases, the neck shear forces in the aft direction were greater than in the fore direction as expected.

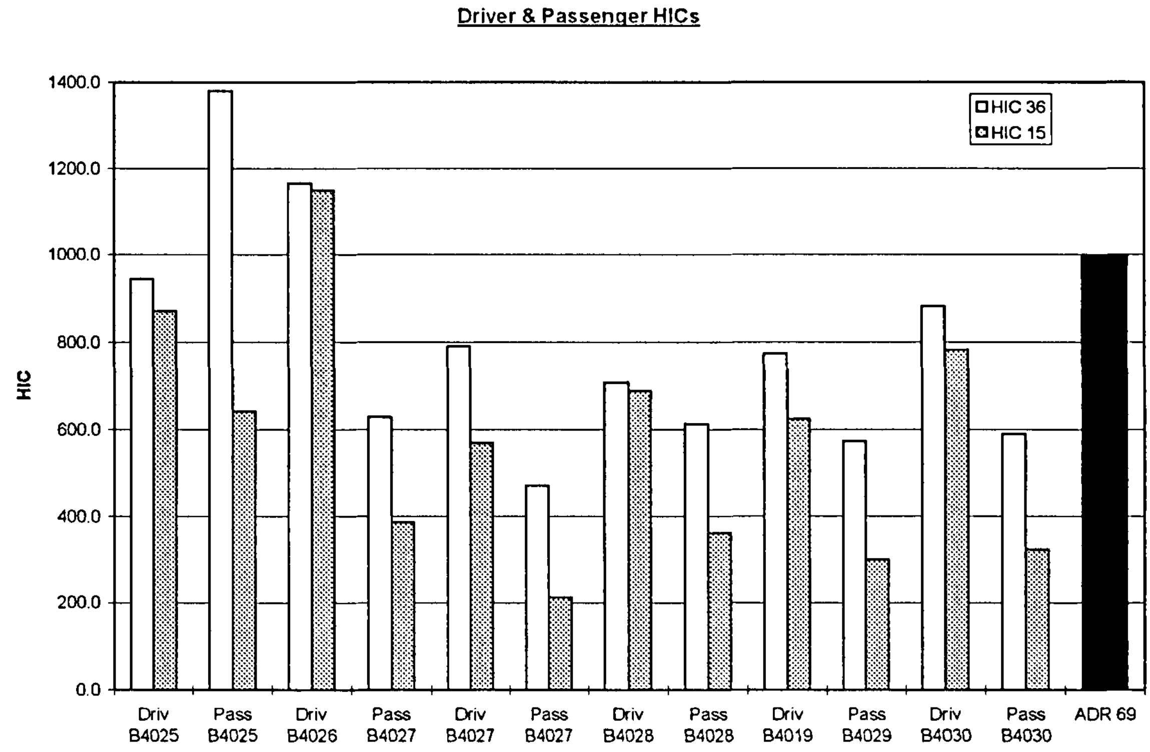

Figure 1. Driver & Passenger HICs.

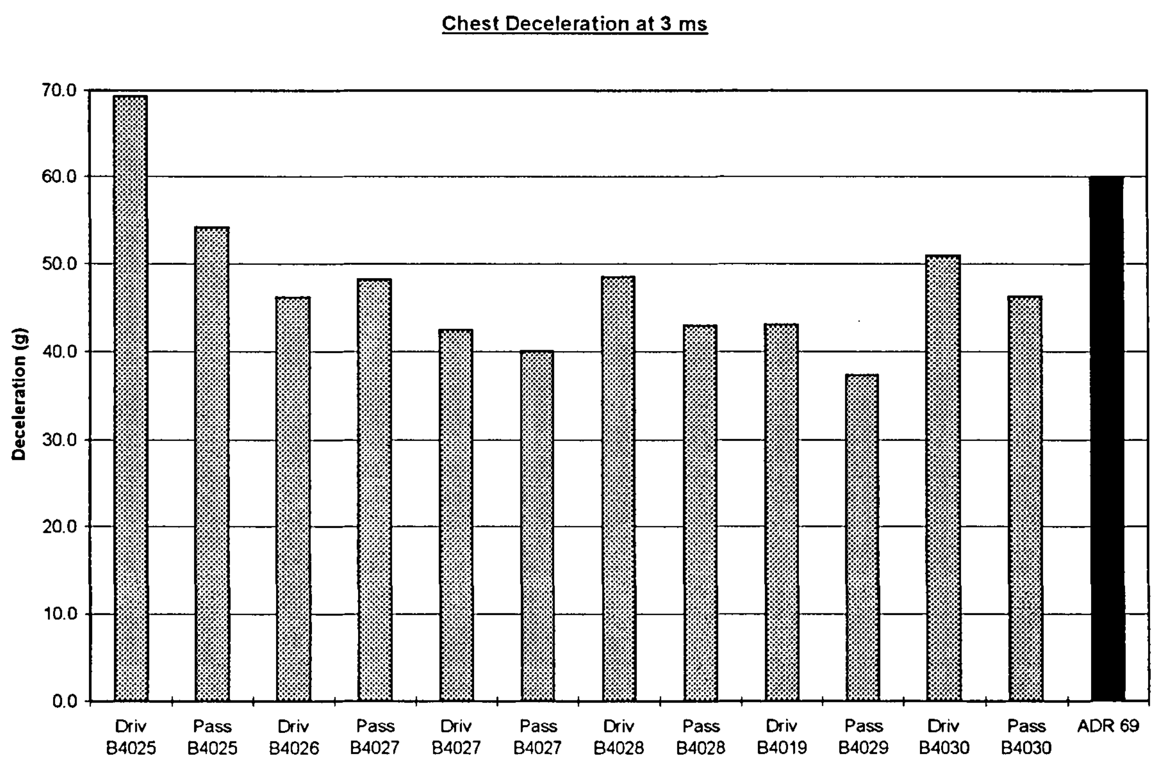

Figure 2. Chest Deceleration.

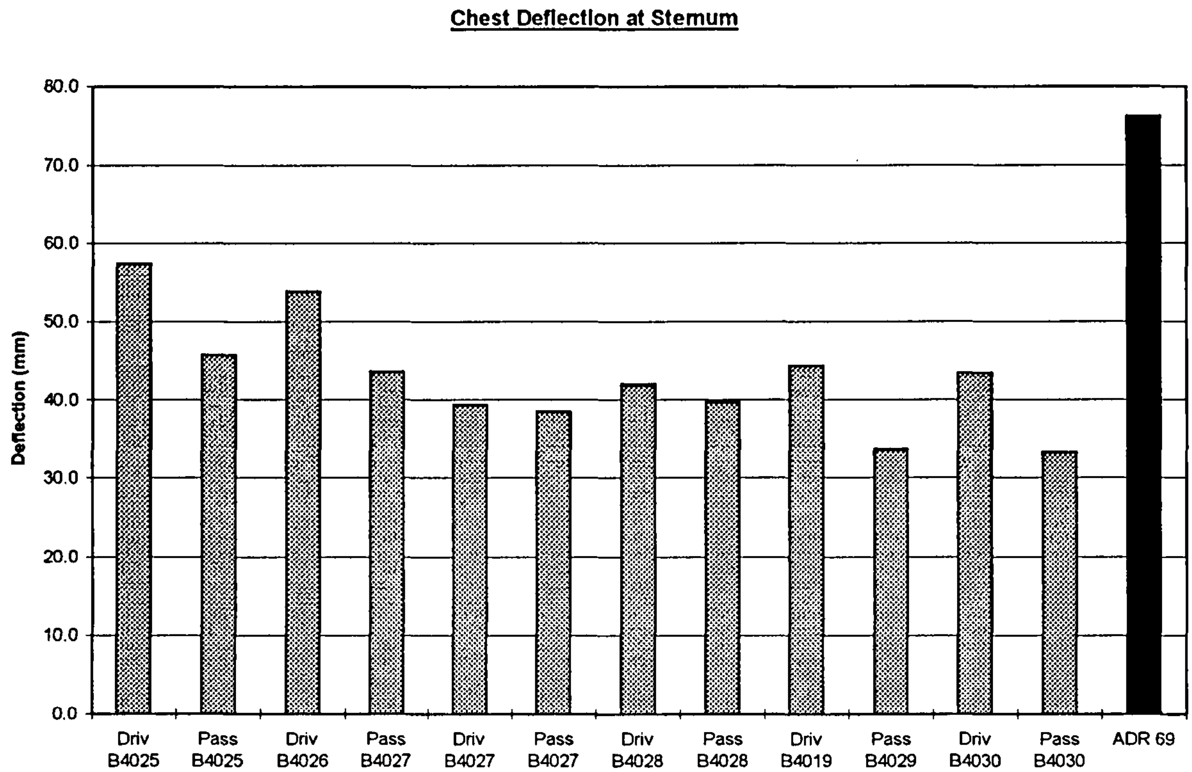

Figure 3. Chest Deflection at Sternum.

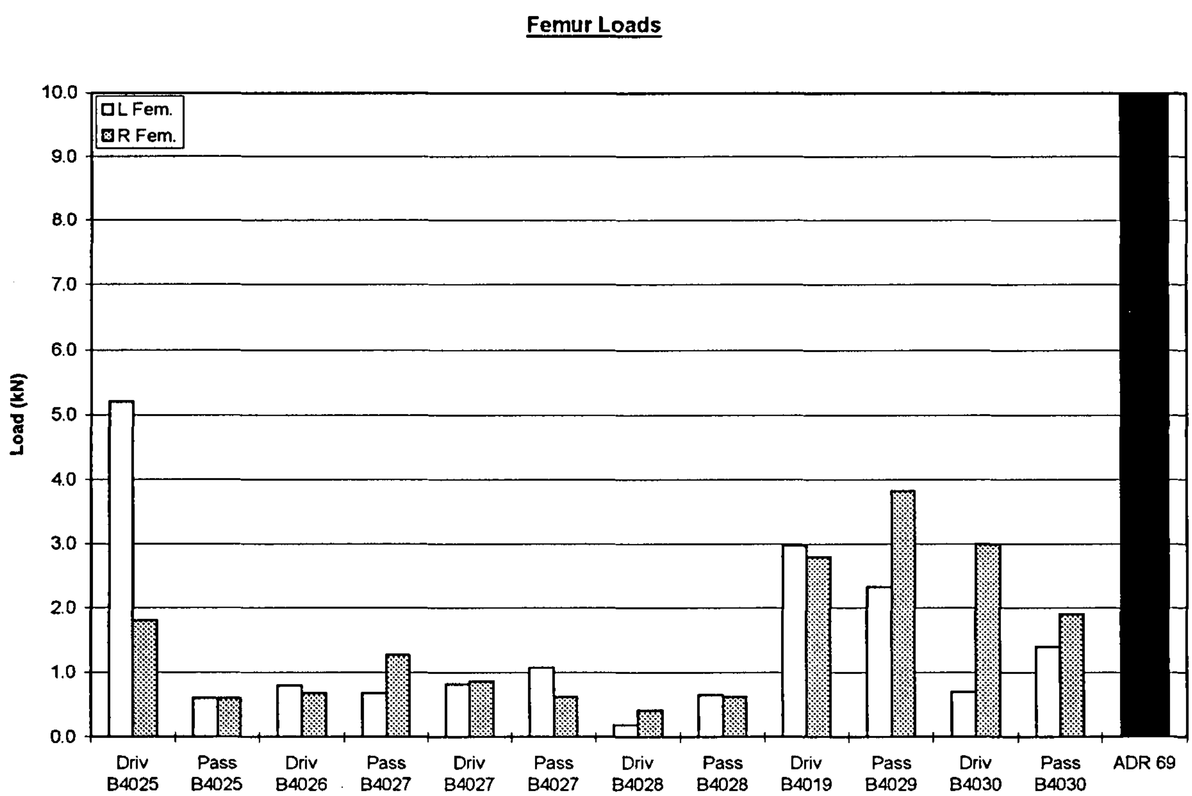

Figure 4. Femur Loads.

Seatbelt Loads

Seatbelt loads were measured at points on both the lap and sash portions of the seatbelt. In summary, peak lap belt loads ranged from 4.4 kN to 8.4 kN for the driver's side and from 4.3 kN to 9.0 kN for the passenger's side. Sash belt peak loads ranged from 6.6 kN to 8.1 kN for the driver's side and from 6.6 kN to 8.3 kN for the passenger's side. Depending on the vehicle seating package configuration, the incidence of low lap belt loadings in some cases coincided with above average femur loads.

During the conduct of the test no seatbelt failures were observed. All seatbelt retractors were observed to have locked during impact while post test inspections revealed all seatbelt release buckles were operative without excessive force.

Seatbelt anchorages in most cases showed loading deformation. However no failures were observed.

Examination of Results

Injury Threshold - In order to gauge the probability of occupant injury, the 'Injury Assessment Values' developed by Mertz[4] and specified in FMVSS 208[5] and ADR 69/00 are used. According to Mertz, dummy response measurements falling below certain developed limits indicate that corresponding occupant injuries are considered unlikely. The limits developed by Mertz and specified in the above mentioned regulations are as follows:

- HIC $\leq$ 1000
- Chest/spine acceleration not greater than 60g for more than 3 ms
- Chest compression $\leq$ 50 mm for sash loading, and $\leq$ 75 mm for distributed frontal chest loading
- Axial compressive femur loads not exceeding that described by the time dependant injury assessment criterion for distributed knee loading. This criterion has been specified in FMVSS 208 and ADR 69/00 such that the maximum femur loading shall be no greater then 10 kN.

By studying the resultant data of each vehicle and applying the 'Injury Assessment Values', the following observations can be made:

- A "significant" head injury is unlikely to occur to the driver and front outboard passenger of any of the vehicles tested except for the driver in B4026 and the front outboard passenger of vehicle B4025. However, note the high HIC figure for B4025 has occurred due to the dummy head contact with its right thigh and not the vehicle structure. The HIC values for the driver are approaching the threshold for vehicles B4025 and B4030 while for the remaining vehicles, both driver and front outboard passenger HICs fall 20% or greater below the threshold.
- A "significant" thoracic organ injury due to gross chest/spine acceleration is unlikely to occur to the driver and front outboard passenger of any of the vehicles tested except for the driver in vehicle B4025 while the value for the front outboard passenger of the same vehicle is approaching the threshold.
- A "significant" thoracic organ injury due to chest compression from the sash belt is unlikely to occur to the driver and front outboard passenger of any of the vehicles tested. All values recorded fell 20% or greater below the threshold.
- A "significant" liver and/or spleen injury due to shoulder belt loading of the lower lateral part of the rib cage is unlikely to occur to the driver and front outboard passenger of any of the vehicles tested.
- A "significant" leg injury is unlikely to occur to the driver and front outboard passenger of any of the vehicles tested. All values recorded fell 20% or greater below a 10 kN threshold.

As specified by Mertz and related to the AIS (Abbreviated Injury Scale), a "significant" injury in this context includes:

- Serious injuries (AIS = 3)
- Reversible brain concussion
- Bone fractures
- Major injuries
  - life threatening injuries (AIS > 3)
    brain damage
    thoracic and abdominal organ damage
  - permanent impairment injuries (AIS $\geq$ 2)
    spinal cord damage
    knee joint damage

In addition to the injury assessment values described above, Mertz developed values for the assessment of neck and lower leg injuries. Although these values are not specified in either FMVSS 208 or ADR 69/00 they are of merit for indicating the probability of neck and lower leg injuries. For occupant injuries to be considered unlikely, bounds for the injury assessment values are specified as follows:

- Neck flexion moment less than 190 Nm
- Neck extension moment less than 57 Nm
- Axial neck tensile loadings for all durations to fall below a curve described by the time dependant injury assessment criterion.
- Axial neck compressive loadings for all durations to fall below a curve described by the time dependant injury assessment criterion.

- Fore and aft neck shear forces for all durations to fall below a curve described by the time dependant injury assessment criterion.
- Combined bending and axial compressive loading of the leg, defined by the following equation, not to

$$\frac{M}{M_c} + \frac{P}{P_c} = 1$$

exceed 1:

Where Mc = 225 Nm
Pc = 35.9 kN
M is the resultant bending moment
P is the corresponding axial compressive force

- Medial and lateral tibial plateau compressive forces to be less than 4000 N

By relating these additional criterion to the measured data of each vehicle and applying the 'Injury Assessment Values', the following observations can be made:

- A "significant" neck injury due to neck flexion is unlikely to occur to the driver or front outboard passenger of any of the vehicles tested. The neck flexion moments measured were all 20% or greater below the threshold.
- A "significant" neck injury due to neck extension is unlikely to occur to the driver or front outboard passenger of vehicles B4026, B4028 and B4029. Neck extension moments measured for both the driver and front outboard passenger of vehicles B4025 and B4030 and the front outboard passenger of vehicle B4027 indicated potential for a "significant" neck injury.
- A "significant" neck injury due to tensile or compressive forces is unlikely to occur to the driver or front outboard passenger of any vehicle except for the drivers in vehicles B4025, B4026 and B4030. For these vehicles neck tensile forces exceeded the upper limit of 3.3 kN indicating potential for a "significant" neck injury to the driver. All other measured loads were found to fall below the force/duration curve. The neck compressive forces measured for the driver and front outboard passenger for all vehicles were well below the of 4 kN threshold while no force measurements were found to fall above the force/duration curve and sustained for the corresponding duration.
- A "significant" neck injury due to neck fore or aft shear forces is unlikely to occur to the driver or front outboard passenger of any vehicle except for the driver of vehicle B4025. For this vehicle the neck shear force in the aft direction exceeded 3.1 kN indicating potential for a "significant" neck injury to the driver. The neck shear forces measured in the fore direction for the driver and front outboard passenger of all vehicles tested were well below the 3.1 kN threshold while no force measurements were found to fall above the force/duration curve and sustained for the corresponding duration.
- A "significant" lower leg injury due to combined bending and axial compressive loading is unlikely to occur to the driver in any of the vehicles tested except for vehicle B4025. In this vehicle the value for the expression given above exceeded 1 for both the left and right legs indicating potential for a "significant" leg injury.
- A "significant" leg injury at the medial and lateral tibial plateau due to compressive loading is unlikely to occur to the driver in any of the vehicles tested. In all cases the measured loads were below the 4 kN threshold.

Further to this discussion it must be noted that the use of the Injury Assessment Values and Injury Threshold Levels developed by Mertz have their limitations. Not all types of significant injuries that an occupant may experience are included while the dummy is only instrumented to measure limited data. In addition, the data collected corresponds only to the collision specified in the test procedure and therefore can not be applied to collisions of differing severities or crash modes. Occupants of different ages and physical condition will also have varying injury tolerances.

Consequently it cannot be stated that an occupant will not experience a significant injury in a vehicle where a measured dummy response fell below the injury threshold. Equally, it cannot be stated that an occupant will experience a significant injury in a vehicle where a measured dummy response fell above the injury threshold. Relating the measured dummy responses to the Injury Assessment Values and Injury Threshold Levels is therefore intended only as a guide for assessing the potential for occurrence of significant injuries to occupants.

Non-Contact Head Injury Criteria - Figure 1 provides the HIC values calculated using both 15 ms (HIC15) and 36 ms (HIC36) integration periods.

Analysis of data by the US National Highway Traffic Safety Administration (NHTSA) has indicated that there was no risk of belted occupants in a frontal crash suffering serious head injury in non-contact crashes, but they might be subject to a risk of neck injuries.

Hybrid II or Hybrid III can currently be used for certification to ADR 69/00. Hybrid III is a more biofidelic test device and would generally result in a better designed restraint system.

However the mere fact that Hybrid III has a more biofidelic neck, which Hybrid II does not, can cause misleading HIC figures in non-contact crashes. In non-contact crashes, the dummy head trajectory is such that the triaxial head accelerometer can record a high overall deceleration. This can be explained as follows:

In the initial ride down phase when the belt engages, the head moves forward and starts to decelerate with a high x-axis (longitudinal) deceleration. As the head rotates further and the dummy's face is pointing toward the ground, the z-axis is now pointing in the longitudinal direction and records a high deceleration along this axis as the dummy's motion is arrested by the restraint. The high resultant deceleration gives a high HIC36 figure even though a hard impact, high level deceleration event such as contact with the dashboard or steering wheel has not occurred.

For this reason a proposal to provide three alternatives to measuring HIC in non-contact events when using the Hybrid III was considered for inclusion in ADR 69/00. These three alternatives were* :

a) Where a Hybrid III test dummy is used, a neck injury criteria which measures the neck tensile force in the inferior-superior (z) axis (vertical) with an injury threshold limit of 3300N.
b) Where either a Hybrid III or Hybrid II test dummy is used, a HIC15 limit of 700 which is currently being considered by Transport Canada. Research has shown that when a hard head impact occurs, the HIC number is the same or similar whether it is calculated over a 36 ms or 15 ms time interval.
c) Where a Hybrid II test dummy is used, an inferior-superior resultant head acceleration limit of 75 g, except for intervals whose cumulative duration is not more than 3 ms.

Using alternatives a and b when comparing the HIC15 data in figure 1 with the measured neck tensile forces (Z) for both driver and passenger, it is seen that HIC15 appears to be a good predictor of neck injury in both contact and non-contact events.

Head to Knee Contact - As mentioned above, the Hybrid III's head and neck are more representative of human response than that of the Hybrid II. However, the Hybrid III head/neck assembly does not totally replicate human head/neck trajectory in a frontal crash situation. Volunteer testing at low level decelerations has shown that the initial motion of a human head is translational in the longitudinal direction which is then followed by rotation as the neck flexes forward. The Hybrid III dummy's head starts to rotate immediately without this initial forward translation.

This difference in response ultimately affects the trajectory of the head. As part of the advanced dummy research, a new head/neck assembly that is better capable of reproducing this translational response is being developed.

The construction of the Hybrid dummies is such that the area of the thigh near the knee joint is much more solid than that of a human. Therefore, impacts in this area produce non-biofidelic responses when compared to a human.

For these two reasons, interpretation of HIC numbers resulting from head to knee strikes should be treated with great caution and must be backed up with proper analysis of the high speed film and other crash data.

DISCUSSION

Motor vehicle crash testing in itself is complex by nature. Not only are there many interrelated variables affecting the test outcome but the relationships between these variables are often unknown. Due to this complexity, test results from vehicles of similar structure often vary considerably. Test to test variability can often be in the order of plus or minus 20%. For this reason, the test results from this crash test program do not form a basis for drawing a sustainable comparison of the safety performance of each vehicle.

Bearing these issues in mind, the following comments can be provided. Of the six vehicles tested, two were found to have exceeded the injury criteria limits imposed on Passenger Cars through ADR 69/00. These two vehicles, B4025 and B4026, exceeded the HIC limit of 1000 for passenger and driver respectively. The chest deceleration limit of 60g was also exceeded for the driver of vehicle B4025.

All of the six vehicles tested had a separate chassis as opposed to monocoque construction. A review of the crash pulses measured at the B-pillar of the vehicle's body show a much earlier onset of crash forces when compared with passenger cars together with a higher peak deceleration. This is the result of the lack of specific crumple zones in the vehicle structure. On the last vehicle tested, B4030, an extra accelerometer was mounted on the chassis to examine whether the mounting system of the cab to the chassis introduced any attenuation of the crash pulse. It was found that the crash pulse was attenuated by some 64% and indicated that the

* All three alternatives were subsequently included in ADR 69/00 from December 1995.

effect provides manufacturers with some opportunity of tuning the dummy kinematics by using the cabin/chassis mounting system.

Generally the results demonstrated that there is scope for improvement of the occupant protection offered in the vehicles tested. The tests also indicated that it is possible to design this type of vehicle to meet the ADR 69/00 criteria. If the ADR 69/00 requirements were to be imposed on these vehicles, further development work would be required in order for manufacturers to gain confidence that all production vehicles met the standard.

CONCLUSION

The crash test program was carried out to evaluate the possible extension of Australian Design Rule 69/00 Full Frontal Occupant Protection applicability to Off-road Passenger vehicles and Light Goods vehicles. The vehicle models tested in the program were built to comply with the current Australian Design Rules for vehicle safety which offer comparable levels of safety to the requirements in force in Europe and Japan.

During the conduct of the tests no body structural or seatbelt failures occurred. In addition, no unexplainably high measures of injury criteria greatly exceeding the ADR 69/00 limits were observed.

Highlighted by the injury criteria results measured, this crash test program has indicated that the application of ADR 69/00 to these vehicles would result in a significant improvement in occupant protection levels provided by Off-road Passenger vehicles and Light Goods vehicles. Although not included in this crash test program, it is envisaged that this would also hold for Forward Control passenger vehicles.

Currently the primary vehicle safety features relating to occupant protection for these vehicle categories is specified through individual performance requirements for individual components.

The application of ADR 69/00 to these vehicle categories would shift the focus toward performance based testing of the vehicle safety system as a whole and in summary would therefore bring the level of occupant protection equal to that of passenger cars.

OUTCOME OF REVIEW

The outcome of the review was to gradually introduce cost-effective improvements to occupant protection in four wheel drive passenger vehicles, light commercial vehicles and forward control passenger vans.

The first stage of improvements will be introduced progressively from 1996 and bring the requirements for these vehicles up to the level currently applied to passenger cars.

The second stage will introduce the performance based requirements of ADR 69/00 for full frontal crash protection to these vehicles

The final improvements will be the introduction of dynamic side and offset frontal impact performance requirements to these vehicles once these standards are finalised in the international arena.

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LOAD RETENTION AND CARGO BARRIERS

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Paper Number 96-S4-W-27

ABSTRACT

" Second Collision " - a collision between cargo carried inside the vehicle occupants compartment and vehicle occupants or vehicle interior structures is not well documented or widely publicised. There are no reliable statistics to quantify magnitude of the problem, although several cases of serious injuries and deaths cased by the unrestrained cargo are reported by the public media every year.

This paper will provide the latest research results from Milford Testing Laboratories after ten years of continuous development of cargo restraining systems. The paper will also provide current status of national standards covering this problem and a special analysis of a comparison between two alternatives for better cargo retention:

- **Stronger vehicle seats**

or

- **Installation of a cargo barrier**

INTRODUCTION

A basic vehicle is a sedan type. Design intent of that vehicle is to be used for transport of 4 - 5 people from one place to another. The amount of cargo that can be carried in this type of vehicle is relatively small (limited with restricted space in the the vehicle boot). From this - basic people mover several other vehicle derivatives are being developed, such as:

	Cargo mass
- Hatchback -	150kg
- Station wagon -	250kg
- Panel van -	750kg
- Forward control van -	1000kg
- Off road vehicle -	450kg

All of above vehicles are designed to carry not more passengers than the sedan, but more or a lot more cargo! This cargo share same space with the vehicle occupants and it is located right behind rear or front seats and in many cases the cargo is higher than the top of the seats.

At present, automotive companies do not provide crash tested cargo restraints for those vehicles, at the same time currently fitted vehicle seats are not capable of restraining significant amounts of cargo.

SECOND COLLISION

Since 1985, Milford Industries and Telecom Australia (now Telstra) have been studying this phenomenon and found that inertial forces of the unrestrained load during the frontal vehicle collision can exceed 2 to 3 times inertial forces experienced on the bodyshell! This depends of:

- Type of the load carried - soft or hard.
- Distance from the cargo to the impacted structure (or human body).
- Stiffness of the impacted structure (or part of the human body; head, back, etc).

A summary of the results is given in Fig. 1,

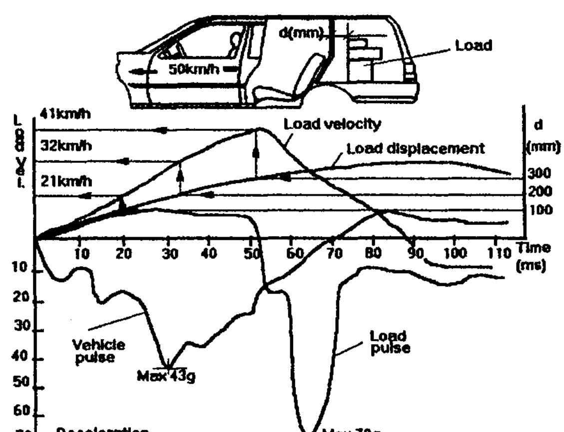

Figure 1. Cargo impact severity

This diagram shows importance of keeping load pushed up against the cargo barrier. The further away from the barrier that the cargo is positioned the greater additional relative velocity is is going to be generated.. The most severe impact will occur when the cargo is positioned far enough that will impact the cargo barrier when the vehicle rebounds rearwards during the frontal collision. During that time the cargo deceleration can reach over 70 "g".

Therefore, an item of cargo weighing:

100kg	becomes	appro.	70,000.N	force
200kg	-II-	-II-	140,000.N	-II-
500kg	-II-	-II-	350,000.N	-II-
1000kg	-II-	-II-	700,000.N	-II-

No vehicle seats can withstand this level of inertial forces. Milford Testing Laboratories crash tests have provided information that cargo of 100kg is capable of dislodging rear and front seats from their mountings (majority of tested vehicles) and crushing them against the steering wheel and the instrument panel.

From the reported real life accidents, the smallest object that killed a person was 2kg pot of honey, which killed a woman in Queensland in 1990. This illustrates seriousness of the problem facing the vehicle occupants from the unrestrained load.

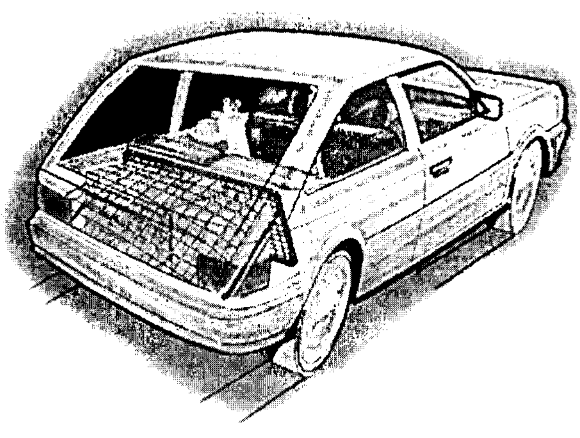

Figure 2. A hatchback partial cargo barrier. The cargo net prevents loose items of cargo to escape over the top of the rear seats and injure vehicle occupants.

FREQUENCY OF ACCIDENTS

There are no reliable accident statistics to quantify magnitude of the problem - bodily injuries caused by the unrestrained cargo. In Australia there are 10 to 20 media reported cases every year.

Indirectly, the number of accidents can be estimated based on the probability that a vehicle carry significant amount of cargo (over 10kg) during the accident. A survey conducted by Milford Testing Laboratories shows that the following percentage of vehicles carry 10kg or more of cargo all the times:

- Hatchbacks	10%
- Station wagons	10%
- Panel vans	25%
- Forward control vans	20%
- Off road vehicles	25%

At the other hand, the vehicles which do not carry " significant " amount of cargo all the time are used in average once a week for weekly shopping (5 - 20kg of groceries) and once or twice a year for long holiday trip (loaded with 20 - - 100kg). The conclusion that can be drawn from above figures is that; every fifth or sixth vehicle might have " significant " amount of cargo during the accident. Therefore, from 3,000 death every year in Australia from the vehicle accidents, in 500 - 600 cases the unrestrained cargo might have contributed to the more severe bodily injuries, and from 40,000. hospitalised people every year, the unrestrained cargo might have contributed to more severe bodily injuries in 6667 - 8000 cases.

In order to draw a parallel between the cargo barriers and some other safety devices, it is interesting to mention that in Australia occupancy of the rear seats is approximately 10% (nine of ten vehicles do not have rear seat occupants). In spite of this relatively low rear seat occupancy, rear seat belts are fitted to every vehicle!

Milford Testing Laboratories have approached traffic accident authorities to include additional information about the unrestrained load, during the accidents investigation, but so far this has not happened yet.

Figure 3. Cargo - 120kg dislodged rear and front seats, during 48km/h frontal crash test.

STRONGER SEATS OR CARGO BARRIERS

There are two ways to solve the problem of cargo retention in motor vehicles designed to carry significant amount of cargo:

1. Redesign vehicle seats
2. Install a cargo barrier

1.Stronger Seats in general would eliminate need for cargo barriers. This solution would require much stronger seat locks, stronger pivoting mechanisms and solid rear seat panel to prevent individual objects; toolbox, pipe, screw driver,etc from passing through the seat foam. In addition, the seat anchorage points would need to be upgraded to meet the current legislation due to heavier seats and the additional load from the cargo impact.

- Firstly, these modifications have to be done to all vehicle seats - front and rear.
- Secondly, how much the seats would be allowed to deform, before they are considered unsafe?
- Thirdly, the cargo still can fly over the top of the seats and injure the vehicle occupants. The collision kinematics forces rear end of the vehicle to <u>move vertically up and throw the cargo over the seats.</u>

2. Cargo Barrier is a more cost effective and more complete solution as a cargo restraint. It covers cargo area from roof to floor and can be fitted on a controlled distance from the seats, to either prevent damage to the seats or minimise damage to the seats to an acceptable level.

Other advantages of the cargo barriers are:

- They can be used in rear and front positions.
- No seat modifications required.
- They can be used for existing vehicles as well as for the new ones.
- They provide protection in roll over accidents.
- They prevent the vehicle occupants from being thrown back towards rear of the vehicle in event of the rear end collision.

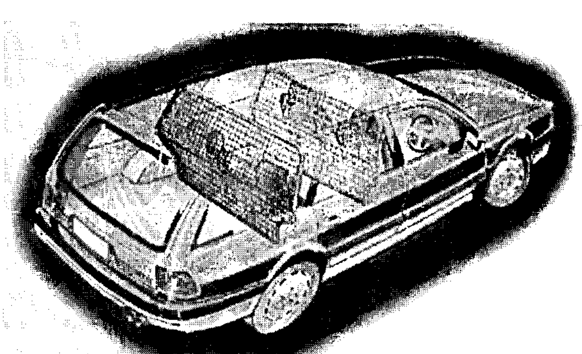

Figure 4. Two position cargo barrier

NEW DESIGN POLYMER BARRIER

The current design steel cargo barrier has been well received by the commercial market (tradesman vehicles, delivery vans, etc.) and has been sold in tens of thousands in past ten years. However new motor vehicles require more aesthetically pleasing design cargo barrier.

During past five years Milford Testing Laboratories have been involved in search for new, clear impact plastic material and the technologies to produce a high tech light cargo barrier for the other end of the automotive market. This market is concerned not just with the performance of the cargo barrier, but also with the emphasis on light design, ease of use and automotive quality finish, see Fig 5.

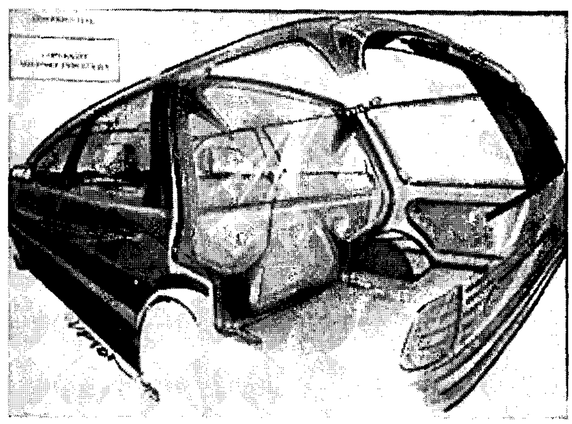

Figure 5. Polymer cargo barrier

This new design cargo barrier has required extensive testing on different ambient temperatures, air flow around the barrier, fogging up and several other technical problems relevant to plastic products.

The polymer barrier, commercially called " Ultrashield " will be released soon on the market as an addition to the steel cargo barrier.

AUSTRALIAN STANDARDS

Current Australian and New Zealand standard AS/NZS 4034:1992 - " Cargo Retention in Passenger Cars Under Accident Condition" is the most comprehensive standard on cargo retention in passenger vehicles. It is regularly updated to include new materials, design concepts and test methods.

It provides design parameters, reference to other Standards and Design Rules. AS/NZS 4034 :1992 it specifies tests methods and the latest edition will have a sled test method as an alternative performance test to the current drop test.

The ME48/2 Australian Standards Subcommittee has just completed new Draft of Standard - Cargo Restraints in Light Vehicles, which addresses other restraining systems than cargo barriers. The cargo barrier is capable to restrain 60kg single mass cargo and approximately 200kg of loose cargo if the items are mass 5 -10 kg. Any additional cargo - larger than 60kg or heavier than 200kg needs to be separately restrained. This Draft of Standard specifies cargo tie down anchorage points to be installed inside the vehicle and used to lash down any additional cargo.

The DR 95214 Ambulance Restraint Systems is a Draft Standard published on 15th of May 1995 and covers specialised medical vehicles. This Draft specifies performance criteria for equipment restraining systems including patient restraining devices. It provides a specification for performance criteria and pass / fail criteria.

After four years as a voluntary standard AS/NZS 4034: 1992 - Motor Vehicles - Cargo Barriers for Occupant Protection the Standard has been greatly accepted by all sides of the automotive market. Large fleet owners and automotive companies are using the Standard as an important set of guidelines to select performing products. Milford Testing Laboratories, Milford Industries (main cargo barrier production company) and several government department and the large fleet owners believe that is right time to push for mandatory standard or inclusion of the cargo barriers in ADR 69 - (Australian Design Rule 69 - Full Frontal Impact Occupants Protection).

CONCLUSION

The cargo barriers are as equally important vehicle safety device as seat belts or air bags. The seat belts and air bags protect vehicle occupants from being injured by impacting into the instrument panel or steering wheel. The cargo barriers protect vehicle occupants from the unrestrained load!

In comparison with the alternative solution - stronger vehicle seats, the cargo barriers are more cost effective and more complete restraining systems.

ACKNOWLEDGMENTS

The author of the paper would like to take this opportunity to thank Milford Industries Management Team for vital support during the development of the product and the National Standard. Special thanks to the General Manager, Mark Giumelli for his assistance in designing the product and assisting in Standard preparation, to John Rees and Rees family (owners of the company) for being great believers of this product and for supporting this project financially. Special thanks to Jim Coogan, Andy Baldock, Paul Statham, Gary Young, George Damjanovic, Jon Shaw, Annie Ryan and Wally Tylar.

Special thanks to Telstra staff: Nicholas Dalinkiewicz, Engineering Manager for many years of support and encouragement to continue with the research and for technical support during Standard preparation and product development, and my former Superivsing Engineer Alan Whittle and mechanic Vince Self for their significant contributions.

Also thanks to very enthusiastic Standard Committee ME/48/2 members; Brian Wells, Jim Eldsen,Barrie Vials, John Goff and all other committee members.

Thanks to Dr. Essie Bidhendi from the University of South Australia for providing electronic equipment to measure deceleration and forces and Frosty Gorostaga from Split Images for providing the high speed photography and Industrial Designer - Mike Upton for unique design concepts .

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Technical Session 5

Vehicle Rollover and Occupant Protection (Crashworthiness and Crash Avoidance)
Chairperson: Kaneo Hiramatsu, Japan

CURRENT RESEARCH IN ROLLOVER AND OCCUPANT RETENTION

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ABSTRACT

This paper provides an overview of NHTSA's rollover crashworthiness research program. This program emphasizes reducing ejections through door openings and side windows. Opportunities to reduce occupant to vehicle interior contacts are investigated through improved restraints, reduced roof crush, and improved upper interior head protection (Federal Motor Vehicle Safety Standard Number 201 upgrade, FMVSS 201).

RESEARCH BACKGROUND

The National Highway Traffic Safety Administration Authorization Act of 1991 mandated that the agency initiate rulemaking on rollover protection. Subsequently, a rulemaking plan titled "Planning Document for Rollover Prevention and Injury Mitigation" was published for public review (Docket 91-68 No. 2, Sept 1992). This paper discusses the current status of the crashworthiness research program that was outlined in this planning document.

On average, 7,797 rollover involved fatalities were reported by the Fatal Accident Reporting System, FARS, between 1988 and 1994. There were also between 43,000 and 58,000 annual rollover involved incapacitating injuries for the same time period, as reported by NASS GES. Approximately 59 percent of the rollover fatalities come from the 10 percent of the rollover involved occupants who are ejected, or partially ejected from the vehicle. Significant safety improvement can be established by reducing ejection alone. Research is currently underway to study methods for reducing ejection through door openings and side windows. For occupants that are not ejected, the research is focused on limiting the frequency and severity of occupant interior contacts during rollover. This research emphasizes the importance of the vehicle structure in providing a safe zone for the occupant, and the restraint systems in providing methods for keeping the occupant within this zone. Limiting roof crush is the most visible aspect of providing a safe zone for occupants. NHTSA is also investigating ways to prevent interior impacts by using seatbelt pretensioners, adjustable anchor points, inflatable restraints, and seats with integrated restraint systems. The performance requirements of FMVSS 201 will also play a role in reducing the severity of occupant-to-upper-interior impacts.

Table 1: Ejection Routes for Rollover Involved Occupants
1988-1994 NASS CDS Annual Averages

Ejection Route	% Fatalities	% AIS 3+
Side Window (front and rear)	56.0	49.4
Side Door (front and rear)	13.9	13.3
Roof Opening	9.0	7.1
Windshield	7.8	9.1
Rear Window	6.3	5.7
Rear Door	0.6	0.7
Other	6.4	14.7

EJECTION MITIGATION RESEARCH

Ejection is largely a problem of belt nonuse. From 1988 through 1994 FARS reports that 92 percent of all rollover ejection fatalities are unrestrained. This includes 93 percent of the ejection rollover fatalities and 73 percent of the partial ejection rollover fatalities. This research program recognizes the great importance of restraint usage but it also investigates additional measures that can be utilized to prevent ejection.

The ejection routes for occupants ejected fatally in rollover accidents are shown in Table 1. The ejection portals for seriously injured (AIS 3+), ejected, rollover involved occupants are also shown. The primary ejection routes are through side windows and door openings.

Ejection Resistant Glazing

An average of 7,492 people are killed and 9,211 people are seriously injured each year in passenger car, light trucks and vans because of partial or complete ejection through side windows (NASS CDS 1988-1994). NHTSA is investigating the safety potential for using ejection-resistant glazing materials such as bilaminate, trilaminate, and rigid plastic. These materials must be encapsulated and retained in the door frame to withstand and retain an occupant impacting from the interior of the vehicle. Based on accident studies, full scale and sled testing, NHTSA has developed a component test that can evaluate a glazing system's ability to retain an occupant[1]. This component test consists of a 18 kg guided impactor, shown in Figure 1, that can impact the interior of a side window from 16 to 24 kmph. A final impact speed has not yet been determined. Prototype glazing systems were developed in conjunction with several industry suppliers. They use bilaminate, trilaminate, and rigid plastic materials that were encapsulated with a polyurethane edge along two sides. These windows were installed in a modified Ford LTD door and tested with the retention impactor. Almost all of the prototype systems retained the 18 kg impacting mass at 24 kmph. Development work is continuing on both the glazing systems and the refinement of the test procedure. Additional testing using the FMVSS 201 free motion impactor is being conducted to evaluate the head injury potential for passengers that may impact these stronger glazing systems.

A cost-benefit study was also conducted[1,2] of the effectiveness of these prototype systems. The cost estimates for the side glazing systems ranged from $96 to $159 additional cost per vehicle. The potential benefits were estimated by evaluating hardcopy accident cases to estimate whether the prototype glazing systems could have prevented the ejection. The vehicle intrusion codes, as reported by NASS, were then used to develop an effectiveness estimate. For example, for an accident with 8 to 15 cm of intrusion near the ejected occupant, an alternative glazing system would prevent 75 percent of the ejections. A matched pair analysis, which compared driver versus front seat passenger injury levels in crashes in which the driver was ejected but the passenger was not, and vice versa, was conducted[3]. This analysis estimated the increased risk of fatality and serious (incapacitating) injury from being ejected. These estimates were then applied to the estimates of the number of ejections that would be prevented by ejection resistant glazing to estimate the overall safety benefits.. This analysis concluded that each year, an estimated 1,313 fatalities and 1,297 serious injuries could be prevented by the use of ejection-resistant front side windows.

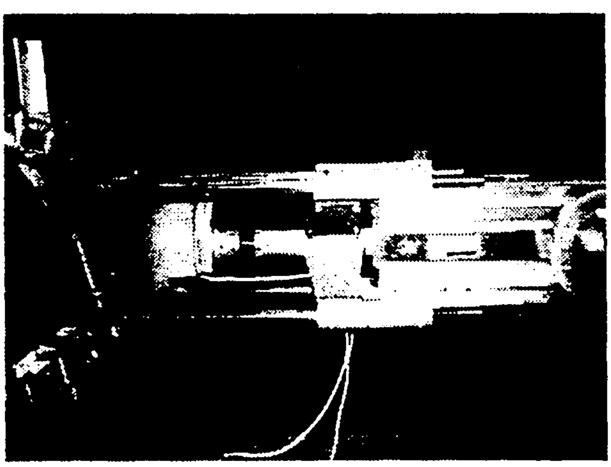

Figure 1. Glazing Retention Impactor.

Door Latch Research

There are an average of 2,974 ejections each year through door openings (NASS CDS 1988-1994). The intent of this research program is to study the causes for door latch openings and to investigate the possible benefits of upgrading FMVSS 206. NHTSA has conducted an investigation of NASS accident cases to evaluate the causes of door latch failure[4]. About half of the NASS cases showed indications of latch activation while the rest of the door openings appeared to be because of structural failure. It was noted that for more than half of the reviewed cases the latches were subject to compressive loadings. Laboratory research procedures were developed to test door latches in failure modes that were more representative of the loading conditions observed in the accident cases. Test procedures for full-door longitudinal strength, full-door lateral strength, detent lever - fork bolt bypass, linkage activation, horizontal rotational loading (GM procedure), and inertial loading were implemented. Almost all of these test procedures produced some door latch failures. However, there was no single test procedure that would clearly identify the "weak link" in all types of latches. One additional test procedure is currently being evaluated. This is a "reverse static 214" test procedure, where the cylindrical loading device is pressed into the interior of the door to evaluate the door mounting system's overall strength. Preliminary research on correlating NASS investigated door openings with the above mentioned laboratory test procedures did not identify a strong relationship with the results of any single test procedure.

REDUCING OCCUPANT INJURY DURING ROLLOVER

This research is intended to study the occupant/vehicle kinematics involved in different rollover accidents and to evaluate the potential for reducing injury for occupants that remain inside the vehicle during rollover. NASS GES indicates that the annual number of rollover-involved non-ejected occupants ranges from 330,000 to 385.000 between 1988 and 1994. The annual number of incapacitating injuries ranges between 39,000 and 50,000 for rollover-involved non-ejected occupants. The majority of non-ejected rollover .involved occupants, 87 percent, receive AIS 1 injuries.

Vehicle intrusion, particularly the roof, has always been a major concern in rollovers. NHTSA is evaluating the potential safety benefits of upgrading the current FMVSS 216 roof crush standard. Additionally, the recently upgraded FMVSS 201 requires a minimally compliant vehicle upper interior that may reduce interior contact injuries in rollovers. The potential safety implications for both these regulations are being investigated in this research program.

Several full scale rollover tests of light pickup trucks have raised concern about the roof strength of these vehicles and a research program was conducted to evaluate the feasibility of reducing it.

GM researchers demonstrated that reducing roof crush alone does not prevent an occupant from contacting the roof, or vehicle interior[5]. Not only must the vehicle structure provide a safe zone for the occupant, the restraint system must keep the occupant within the zone. NHTSA has recently begun research to evaluate the performance of various restraint alternatives in rollover situations.

Advanced Roof Crush Testing

NHTSA is conducting research to compare the performance of vehicles in the FMVSS 216 tests against NASS-investigated accident cases[6]. The deformed roof shapes seen in the FMVSS 216 tests were considerably less severe than those observed in the NASS cases. A series of extended FMVSS 216 tests were conducted where a vehicle was successively crushed to four levels: the FMVSS 216 required load levels, 13, 25, and finally 38 cm of crush. The deformation profile of the roof was recorded at each of these load levels. These tests produced roof crush patterns similar to those seen in the NASS investigated accidents.

Vehicle drop tests were then conducted at energy levels equivalent to the energy absorbed in the static tests. Each vehicle was supported with its roof toward the floor and positioned so that the roof would be initially loaded at the same angles as in the static tests. The vehicle was then dropped onto a flat surface. Successive, or multiple drop tests, were conducted to obtain vehicle deformation for comparison with the static tests. Figures 2 through 4 compare the deformed roof shape for a Nissan pickup after a full scale rollover test against both an extended FMVSS 216 test and a dynamic drop test. All three vehicles show a similar deformed profile. One shortcoming to this type of comparison is the

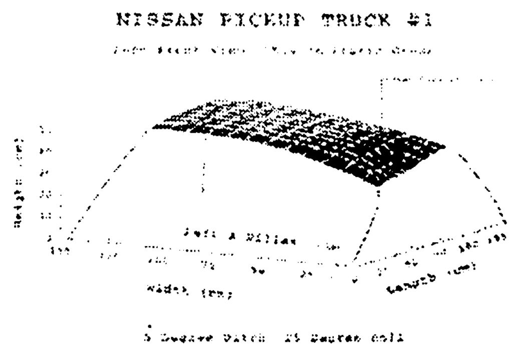

Figure 2. Deformed Roof Profile for Extended FMVSS 216 Procedure.

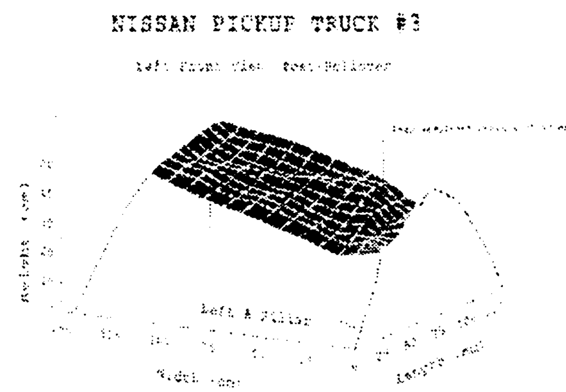

Figure 3. Deformed Roof Profile for Dynamic Rollover Test.

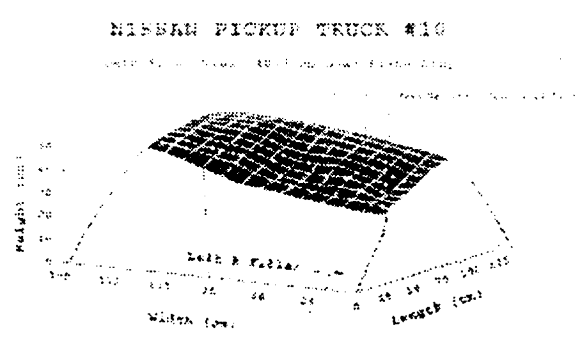

Figure 4. Deformed Roof Profile for Roof Drop Test.

lack of control in the rollover test and the significant amount of roof spring back experienced in the static tests. When the load is removed in static tests, several centimeters of spring back may occur, and post test measurements do not reflect the full severity of the damage to the vehicle. Conversely, during rollover tests, the vehicles are prone to bouncing and incur multiple impacts with the ground. These multiple impacts affect the measured profile of the deformed roof. To avoid these complications, further drop tests were conducted onto a load plate to obtain dynamic force/deflection characteristics (see Figures 5 and 6). Note that the force/deflection profiles from the dynamic tests are much higher than those from the static tests, as shown in Figure 7.

Research is continuing to see if the static test results can be correlated to the dynamic results on an energy basis. It is advantageous to utilize a static test procedure for roof crush evaluation for several reasons. These include that the current FMVSS 216 requires a static test, thus hardware and procedures are already in existence, the ease of setup and testing, and repeatability. However, the static test results are far more useful if they can be used to qualitatively predict the dynamic performance of the roof strength. Once a test procedure has been developed, it will be necessary to compare baseline testing data against the NASS investigated accident

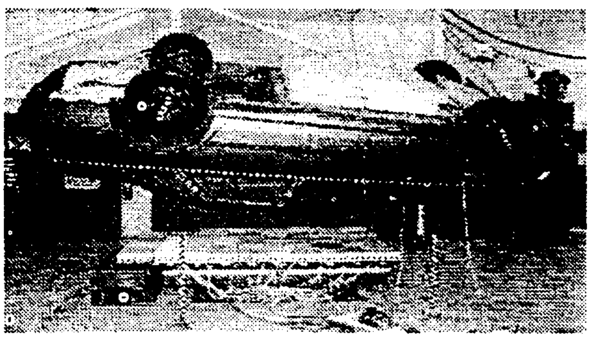

Figure 5. Load Plate Drop Test - Pre-Test.

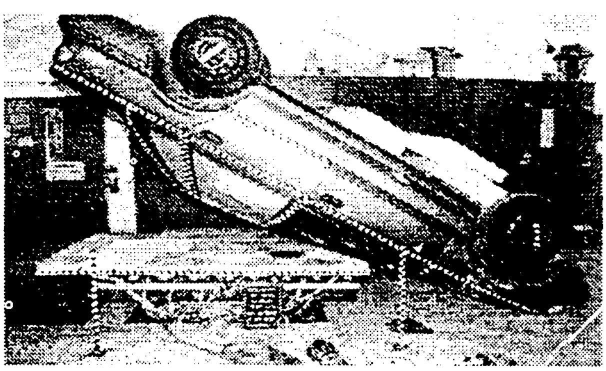

Figure 6. Load Plate Drop Test - Post Test.

data in order to evaluate the effectiveness of the test procedure.

Vehicle Modifications to Reduce Roof Crush

Full scale rollover testing of light pickup trucks displayed significantly higher roof crush than for the other vehicle

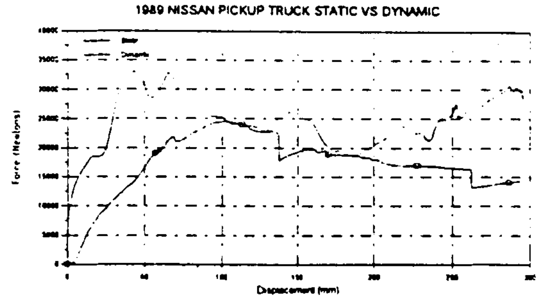

Figure 7. Comparison of Static and Dynamic Force/Deflection Profiles.

categories tested. The rollover test was designed to simulate a severe rollover accident and to evaluate a vehicle's ability to resist roof crush[7]. In this test, a vehicle is traveling at 48 kmph and is tossed from a height of 1.2 m with sufficient rotation to land the vehicle on its roof. Large roof crush was noted for all the light pickup trucks that were tested. In order to evaluate potential countermeasures, a contract was awarded to Pioneer Engineering with follow up work awarded to EASi Engineering[8]. The task was to design and modify the roof structure of a Nissan pickup truck to reduce its roof crush during full scale testing. All vehicle modifications were intended to be consistent with high volume manufacturing techniques. A finite element model of the vehicle was developed and analyzed to evaluate multiple configurations.

Figure 8. Unmodified Nissan Pickup, NHTSA Test 1394.

A series of four vehicles were sequentially modified and tested in this program. The modifications included up-gauging the inner panels of the B pillars and rear roof headers, injecting a structural foam into the B pillars and roof headers and adding some additional gussets and flange material. Each of these vehicles was tested on the NHTSA rollover test cart at 48

Figure 9. Modified Nissan Pickup, NHTSA Test 2270.


Figure 10. Restraint Systems Rollover Test Device

kmph in a full lateral roll. Figures 8 and 9 show similar views of a post test crush for an unmodified and modified vehicle. These photos display a significant reduction in roof crush. However, considering the extensive vehicle modifications that were used, this program demonstrated the difficulty of eliminating roof crush in severe rollovers.

Rollover Testing of Advanced Restraint Systems

NHTSA has recently initiated research to evaluate the safety potential for advanced restraint systems in rollover accidents. Previous studies[5] have shown that in a rollover test, a belted occupant can contact the roof structure prior to any roof deformation. Additional studies with NASS hardcopy cases of restrained occupants involved in rollover accidents[6] showed that even when roof deformation is not apparent, head injury from roof contact still can occur. This program is intended to evaluate the performance of restraint systems in reducing the occupant motion during a controlled rollover. A test device is being developed to provide a controlled roll rate for a seated occupant. The prototype test device is shown in Figure 10. The occupant will be restrained in the seat and a drop weight will roll the seat over. The seat base will then impact an energy absorbing barrier, simulating roof to ground impact. The test device will simulate roll rates from approximately 0.5 to 1.5 rev/s. Several restraint system designs will be evaluated on the basis of their ability to restrict the occupants motion in the vertical and lateral directions under a variety of simulated roll rates and directions. Pretensioning systems will be initiated at different times during the seat rollover to establish criteria for rollover sensing systems. Different occupant sizes will also be examined and belt loads will be measured, when applicable.

Additional Interior Contact Countermeasures

NHTSA is also evaluating the safety benefits that may stem from FMVSS 201 upper interior head protection in rollover accidents. Researchers at Wright Patterson Air Force Base are developing an ATB computer simulation of a NASS-investigated accident where the driver received a severe head injury from contact with the roof. This simulation will be used with models of several different types and thicknesses of padding to evaluate the potential for reducing head and neck injuries because of roof contact. Additional lumped parameter computer models are also planned for use in conjunction with the restraints research. Simulations of the rollover test will be used to develop accurate restraint models for evaluation of occupant protection in these simulated rollover accidents. Also, models of inflatable head restraints are being evaluated for their potential application in rollover accidents for reducing ejection and mitigating interior contact injuries.

FUTURE RESEARCH

Rollover is a widely varying accident type that is difficult to characterize and it is difficult to determine with any certainty which injury mitigation approaches would provide the greatest safety benefits. One of the long term research goals is the potential for developing a statistical characterization of the rollover accident environment. It is hoped that some of the newer NASS variables, such as tip-over and trip-over, can be used to develop a characterization of the rollover environment. This would enable the development of computer simulations for a finite number of accident types and vehicle types that could be used to estimate the safety impact of rollover countermeasures. These estimates could then be used to help guide rollover research towards countermeasures that could provide the greatest safety benefit. Work is currently underway, but it could be several years before there are enough accident cases to support a valid characterization.

NHTSA intends to further investigate opportunities for using rollover countermeasures to provide a safe zone for the occupant as well as restraining the occupant within this zone. Both of these approaches must be combined to achieve significant safety improvement in rollovers.

ACKNOWLEDGMENTS

We would like to thank Ms. Catherine McCullough of NHTSA and Mr. Dan Elliot of Abacus for preparing the statistical references in this paper.

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THE SAFETY OF CONVERTIBLES IN REALISTIC ROLLOVER CRASH TESTS

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Paper Number 96-S5-O-02

ABSTRACT

The US regulation FMVSS 208 defines a rollover as when a vehicle with no longitudinal velocity starts to roll over unaided. However, in real-life accidents rollovers usually take place with some longitudinal velocity component. This results in very different loading conditions for the passenger compartment and windscreen frames. Rollover safety is particularly important for convertibles because of the high injury potential. Although this accident mechanism is relatively rare, realistic rollover testing of convertibles is nevertheless meaningful since the number of registered convertibles in Germany, for instance, is continually increasing.

TÜV Bayern has, therefore, developed a new procedure in which vehicles are rolled over with a specified longitudinal speed. This is made possible using the TÜV Bayern ECV Crash System, which controls the vehicle in driving along a special one-sided ramp.

This paper deals with a series of convertible rollover tests. The weak points discovered in the vehicles included the failure of a rollover bar and insufficient windscreen frame stiffness. Results from a sedan rollover test are shown for comparison.

INTRODUCTION

Airbags, belts and stable passenger compartments are making cars increasingly safer. This is true for the majority of accident mechanisms such as frontal, side and rear impacts and equally for rollover. In general, rollover has played a more subordinate role in relation to injury potential. But what about rollover in an convertible car?

It can be fundamentally stated that the proportion of rollovers in total accident occurrences is relatively small. Data collection gives the proportion as generally not more than 5%. Of these, convertibles comprise a below-average proportion of total accident occurrences.

Because convertible cars so seldom have accidents, there are no representative numbers for accident mechanism or injury risk. One thing is nevertheless certain: during a rollover in a convertible car the head, neck and arms are the body parts worst injured due to the rotation of the vehicle. Even when the bodywork is undeformed, they can far too easily end up outside the vehicle contour, thereby giving rise to a high, incalculable injury risk. Even when the front windscreen frame does not collapse, severe head, neck and arm injuries can be expected. This accident mechanism has therefore an importance which can no longer be neglected.

In order to clarify these question, TÜV Bayern has developed a new test method in which rollovers can be carried out extremely realistically.

TEST METHODS

Legally prescribed rollover tests exist only in part. There is no such test in Europe. In the USA there is a load resistance test but convertible cars are exempt so long as they are fitted with three-point belts.

Despite this, the American tests are also modified for use with convertibles, so that the roof strength can be tested according to US regulation FMVSS 216. In this test, the roof corners, beginning at the A pillars, are loaded with 1.5 times the vehicle weight and may not deflect more than 127 mm (Fig. 1).

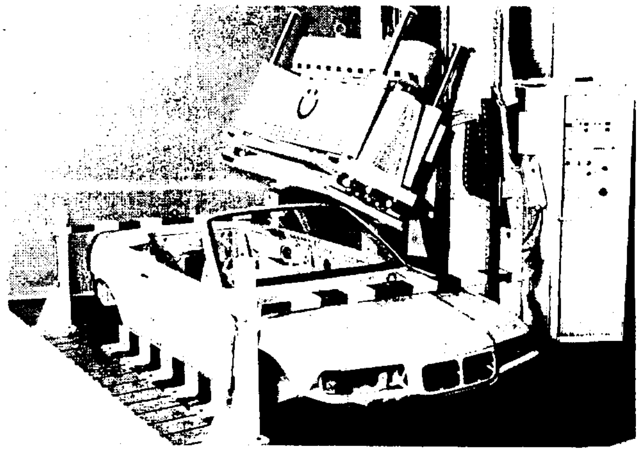


Fig. 1 Roof test acc. to FMVSS 216

Fig. 2 Rollovertest acc. to FMVSS 208

Fig. 3 Screw rollover test

A more realistic rollover test is set out on US regulation FMVSS 208 (Fig. 2). In this dynamic test, the crash candidate is held tilted at an angle of 23° in a carriage and slid in a transverse direction along the test track. The carriage speed is 30 mph (48 km/h). The regulation is satisfied when no occupant body part moves outside the vehicle contour during the rollover. Naturally, the dummies used can be instrumented to give better information on the occupant loadings.

Even more realistic, but harder to achieve, is the newly developed screw-rollover test, in which the vehicle is driven over a 1.1 m high one-sided ramp at around 70 km/h (Fig. 3). In contrast to the launch test, the screw-rollover test includes a relatively high speed in the vehicle longitudinal direction, which leads to a higher loading of the windscreen frame. This test is further described in the next section.

THE SCREW ROLLOVER TEST

In contrast to the US rollover test to FMVSS 208, the novel screw rollover test has a velocity component along the vehicle's longitudinal axis. This velocity component is necessary for a realistic reproduction of rollover, since real-world rollovers usually begin with the vehicle moving longitudinally.

Particularly in combination with the windscreens commonly used today, which are strongly angled to reduce drag coefficient, this test procedure gives a higher but more realistic loading on the windscreen frame.

In order to start the screw rollover the car is driven at about 70 km/h against a 1.1 m high one-sided ramp. The roll impulse imparted is so high that the car is launched completely into the air and will theoretically turn through 180° about the longitudinal axis with only moderately reduced velocity.

Depending on their mass and mass distribution, most test candidates reach 120° to 150° before touching the ground with the right rear wing or boot lid. The impact of the windscreen frame follows, and the vehicle then skids for about 30 m to a halt.

This test simulates an extremely severe rollover which allows conclusions to be drawn about the strength of the windscreen frame, the effectiveness of the rollover bar and the calculation of the injury risk.

In order to allow driverless control, the TÜV Bayern ECV (Electronically controlled vehicle) system is used for the test.

The particular advantage of this test method is that the vehicle can be driven under its own power. The ECV system can further be distinguished from other current test methods as follows:

- high longitudinal and transverse precision through electronic steering and control system;
- complete transportability, i.e. the ECV system is universally applicable to enclosed test tracks;
- the greatest possible flexibility with respect to crash configuration and vehicle combination include the collision of two moving vehicles;
- adjustable for different passenger cars and trucks.

The required speed is entered into the on-board vehicle controller before the test. The vehicle is constantly and steadily accelerated to the desired speed by an actuator. The actual speed signal comes from a tacho hub, normally mounted on a non-driven wheel. System accuracy is ensured by means of control measurements on comparison trips with the help of an external speed measurement system. An automated gear shifter can be incorporated to cover large speed ranges or for use in heavy trucks. Passenger cars are normally driven in second gear.

Path control is achieved by driving the vehicle along an electrical pilot cable. The cable is an emitter with a 10 kHz A/C signal, generating a concentric electromagnetic field. An antenna mounted on the front of the vehicle detects lateral deviations by measuring the field intensity and communicates these to the onboard computer. The required steering corrections are automatically calculated and transmitted to the steering shaft by an electric motor. Figure 4 shows a functional schematic of the ECV Crash System. Detailed information about this technology can be gained from the fourth reference.

Since, with screw-rollover as a criterion, survival space reduction must be taken into consideration, as well as for reasons of damage risk, dummies such as Hybrid III were not used for convertible tests. Moltopren plastic dummies were used to simulate occupants in convertibles. Measurement data desired can be recorded from the vehicle. In general, these are accelerations, forces or dynamic deformations.

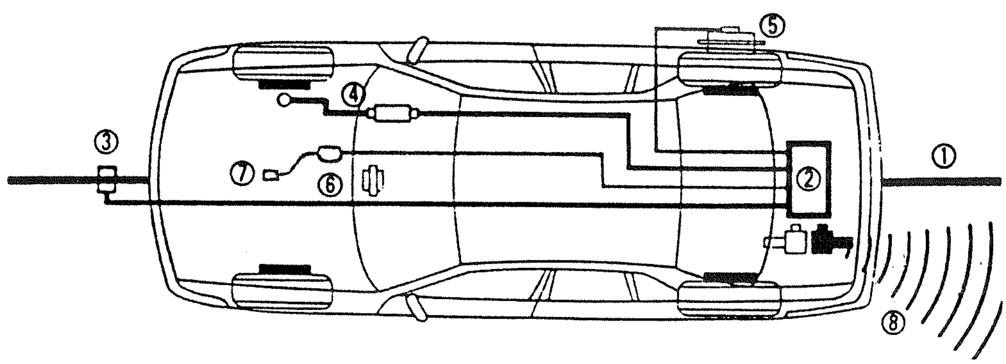
TEST RESULTS

Several tests were carried out according to the conditions described in the previous section. The series of photographs shown as Figure 5 illustrate the typical progress of a screw rollover test.

The German magazine „auto motor und sport" describes the results of screw-rollover tests on 4 modern convertibles distributed in Europe. For two vehicles, the front windscreen frames did not withstand the impact and folded down on both sides to the beltline (Figs. 6 and 7).

The test candidates shown in Figure 8 had the A-pillars strengthened by a steel tube. Nevertheless, the windscreen frame also gave way here. The left A-pillar broke away first, the right A-pillar stayed in place but deformed to an angle of 17°. Therefore, in the case of such a severe impact a high to life-threatening injury risk must be reckoned with for this vehicle just as with the first two. All test dummies displayed deep scratch marks in the area of the face, shoulders, upper arms and chest. Cracks in the dummy necks and two complete decapitations imply a poor prospect for living occupants.

The effectiveness of a fixed standing rollover bar in the case of test car 4 was also impressively tested by screw rollover. Although both A-pillars were somewhat buckled on this convertible, the bar hindered the collapse of the windscreen frame and insured the survival space by remaining in position at the rear (Fig. 9). Two occupants (left front and left rear) briefly had contact with the ground. The test dummies displayed no obvious scratching or scraping.



1 pilot cable
2 control unit
3 antenna
4 electric motor for steering
5 speedometer hub
6 actuator
7 throttle valve
8 remote control for braking

Fig. 4 **Functional schematic of the ECV crash system**

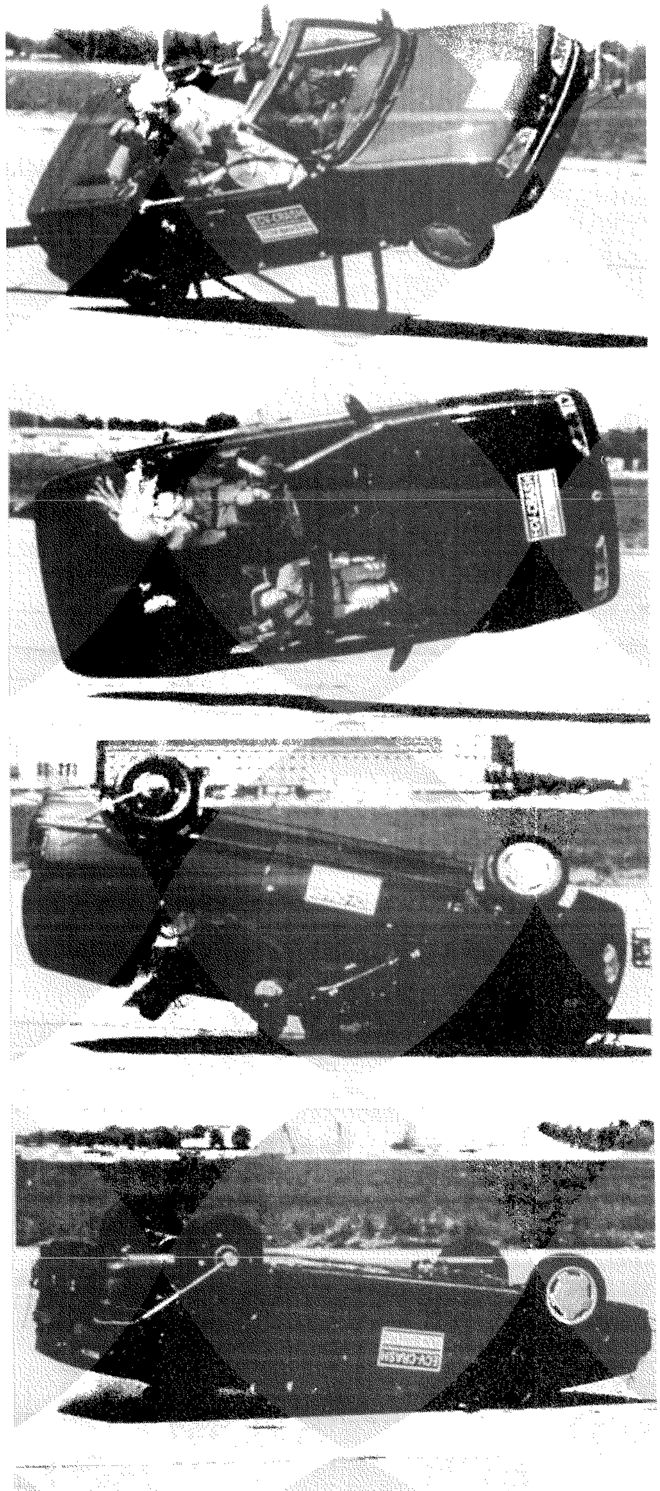


Fig. 5 **Progress of a screw rollovertest**

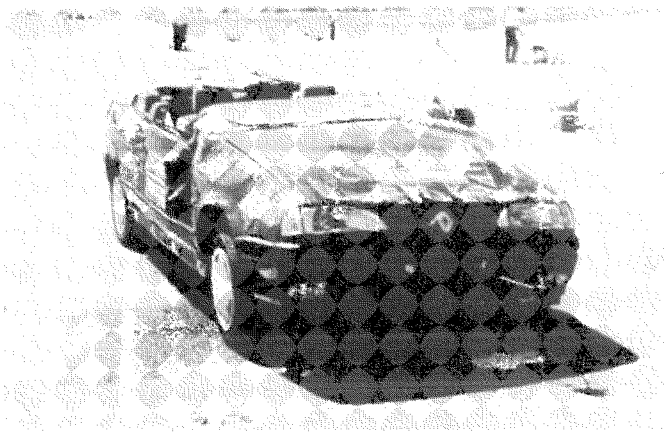


Fig. 6 **Test vehicle 1**

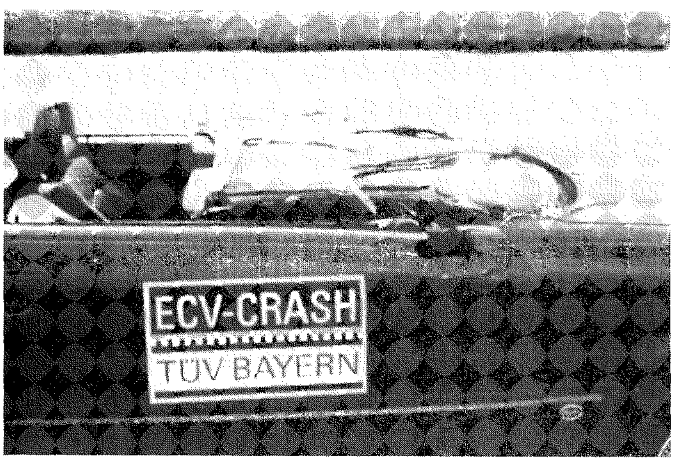


Fig. 7 **Test vehicle 2**

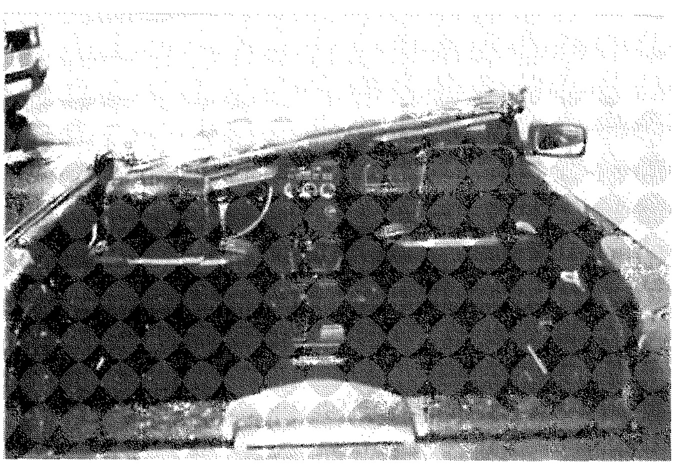


Fig. 8 **Test Vehicle 3**

CONCLUSION

Because of the special injury mechanism in rollover, the conservation of survival space must be an essential goal of convertible design. A simple test criterion is therefore suggested for the screw rollover test in order to assess survival space.

As shown in Fig. 10, a measuring bar is placed on the deformed convertible along the sagittal plane of the driver or front seat passenger. For convertibles without a rollover bar, this line usually runs from the windscreen frame to the boot lid corner. In addition, the distance from the measuring bar and the R-point is calculated. In accordance with the definition of the headrest height, this measurement must be greater than 750 mm in order to withstand the test.

In order to pass the test, as the results in the previous sections have illustrated, protective measures such as a rollover bar are necessary. Fundamentally, it becomes more effective the closer it is placed to the occupants to be protected and the higher it is. The strong solution of placing it at the B-pillar position, as shown for test vehicle 4, is effective but not particularly aesthetic. For this reason, there is a range of other protective systems such as automatically deploying rollover bars or integrating rollover bars into the seat. Together with a sufficiently rigid windscreen frame, such systems offer adequate protection in a severe rollover.

Finally, in comparison with the results from convertibles, a screw rollover test with a sedan car is shown. Figures 11 and 12 demonstrate that this test can be withstood by an enclosed car without modification, even with an older model. The survival space exhibits practically no reduction in this test.

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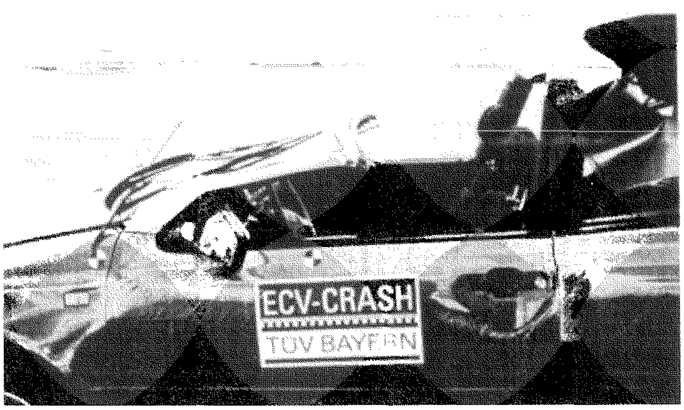

Fig. 9 **Test vehicle 4**

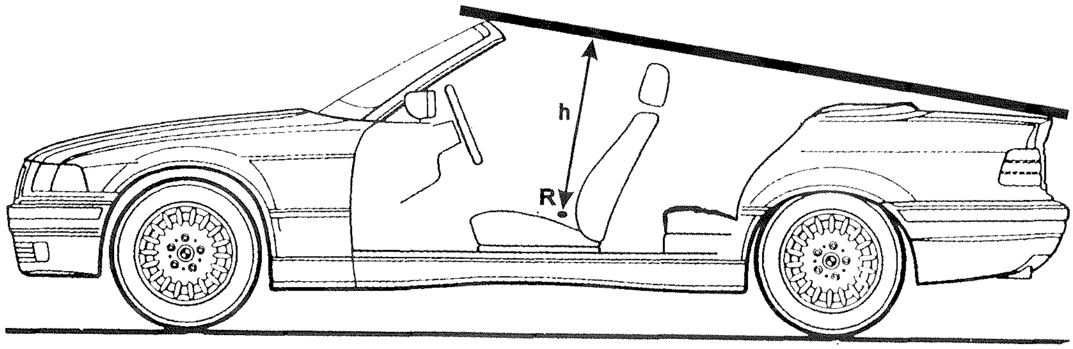

Fig. 10 Evaluation of the survival space

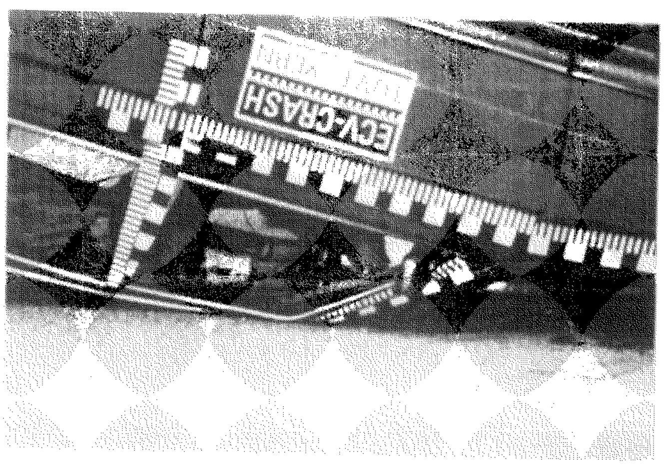

Fig. 11 Screw rollovertest with a sedan

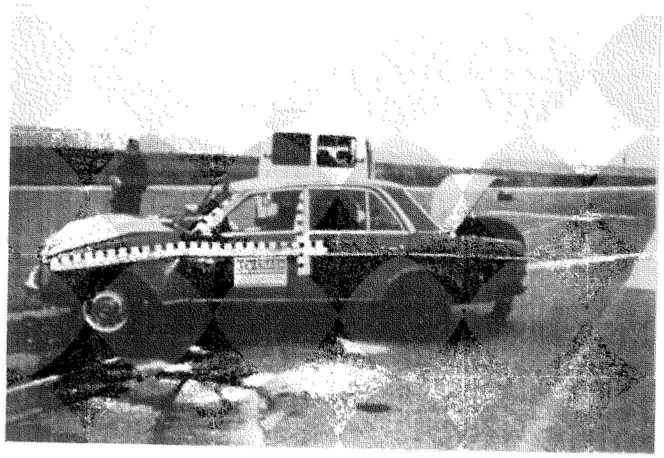

Fig. 12 Screw rollover test with a sedan

ROLLOVER PROPENSITY OF VARIOUS CATEGORIES OF AUSTRALIAN VEHICLES

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Paper Number 96-S5-O-03

ABSTRACT

Certain US 4WD utility vehicles now require labelling indicating risk of rollover. This labelling followed field observations that 4WD vehicles were much more likely to fall over. Quite a few of these vehicles had a narrow wheel track and high centre of gravity. From this and associated litigation we have observed a fairly quick widening of wheel base on imported 4WD vehicles, however windscreen labelling on Australian 4WD vehicles is not required, and we have no assessments of their overall rollover propensity. However, in analysis of field data it has been observed that 4WD vehicles are significantly overinvolved in rollover crashes.

This project conducted static rollover tests on a tilt table on a variety of 4WD wagon and utility vehicles and passenger sedans on sale in Australia. Future direction is to review the involvement in rollover crashes of equivalent categories of 4WD vehicles and passenger sedans and to compare static rollover stability to equivalent US categories.

INTRODUCTION

The number of rollover accidents occurring on NSW roads is small compared to the US. NHTSA (1) reports that almost 10 000 people are fatally injured each year in rollover crashes. Utility vehicles and light duty pickups were identified as being least stable of the passenger vehicle types. The reason for the higher incidence of rollover of these types of vehicles is the poorer stability characteristics. High centres of gravity and narrow track widths make them more likely to rollover.

On average there are six 4WD rollover accidents (as the first impact) resulting in fatalities on NSW roads every year. Table 1, shows the number of rollover accidents involving 4WD's (non sedan type) with the varying degrees of injury.

The number of fatalities on NSW roads in 1994 involving rollover, as the first impact, was 5 in 4WD's and 24 in passenger cars (sedans/stationwagons).

Rollover statistics on 4WD's and passenger cars, gathered from NSW data, show that 4WD vehicles are over involved in rollover accidents. There are 35.5 times as many passenger cars on register as 4WD's, yet the fatality rate for rollovers involving 4WD's is 7.5 times higher than for passenger cars. If we take into consideration the total number of rollover accidents, regardless of degree of injury, the 4WD rollover rate is 1.7 times higher than the rate for passenger cars.

Presently there is no mandatory standard for evaluating a vehicle's rollover or stability characteristic in Australia.

Many studies have attempted to correlate the rollover propensity of a vehicle with measurable vehicle characteristics (1,2,3). The three most commonly used indicators are:

1. Static Stability Factor
2. Tilt-Table Ratio
3. Side-Pull Ratio

When experimental difficulty is assessed for the three methods, the ease and accuracy of the tilt table test makes it most useful. The vehicles centre of gravity and weight does not need to be known, reducing the possibility of errors. The tilt table test and side pull test are more representative of a vehicle negotiating a turning manoeuvre since the vehicle's suspension movements and tire deflections are taken into account (1).

Table 1. 4WD Rollover Accidents

	Degree of Accident				
Year	Fatal	Serious Injury	Other Injury	Non-casualty	Total
1992	6	12	20	41	79
1993	6	11	18	41	76
1994	5	16	29	55	105
September 1995	3	12	21	34	70

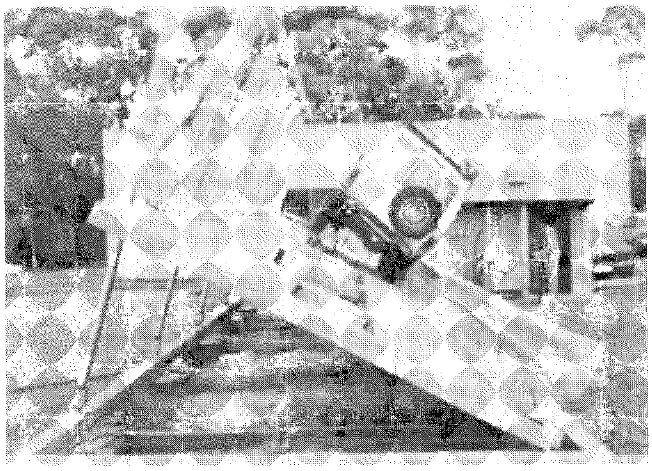

Figure 1 - Tilt Table

TILT TABLE TESTS

The Australian Road Research Board (ARRB) tilt deck facility, Figure 1, was used to test and rank vehicles based on a measure of their static rollover stability. The ARRB tilt deck has been mostly used to establish the stability limits of heavy articulated vehicles.

The tilt deck approximates the lateral acceleration acting to roll a vehicle in a steady turn, by inclining the vehicle in the roll direction. As the deck angle changes it causes the force of gravity to act in the lateral as well as in the vertical direction. The test also simulates, more accurately, a vehicle negotiating an embankment or side slope.

The lateral and normal components on the vehicle are:

$$F_{lateral} = m.g.sin\phi$$

$$F_{normal} = m.g.cos\phi$$

The Tilt Table Ratio (TTR) is defined by dividing the lateral force by the normal force.

$$\text{TTR} = \frac{F_{lateral}}{F_{normal}} = tan\phi$$

where, ϕ is the angle at which the vehicle's wheel (at the high side) looses contact with the deck.

TEST VEHICLES

Fourteen vehicles were selected so as to span a broad range of vehicle types, comprising of three passenger cars (sedans), seven 4WD wagons and four 4WD utilities. Table 2 lists the vehicles tested.

All vehicles were manufacturers standard with exceptions of the Mazda B2600 (non standard tyres and gas conversion), Toyota Landcruiser 4500 GXL and Nissan Patrol Ti (tow assembly at rear), and the Toyota Troupe Carrier. The Troupe Carrier had been extensively modified for its application of outback tours. Modifications included changes to suspension stiffness and fitment of after market shock absorbers.

TILT TESTING PROCEDURE

In preparation for the tilt tests, tyre pressures on each vehicle were set to that recommended by the manufacturer and the fuel tanks filled. The vehicles were to be tested in the laden condition with the equivalent of four 75 kg passengers, representing the 50th percentile male mass, and for the 4WD vehicles a further 100 kilograms of cargo located in the wagon or utility section. The passenger loading was achieved by placing sandbags at the designated seating position. The load was secured to the seat by the seat belt and secured in the wagon to prevent the load from moving excessively during the test.

Table 2. Test Vehicles

Vehicle Type	Year	Description
Passenger Cars		
Mitsubishi Magna	1993	5 seater sedan
Ford Laser	1994	5 seater sedan
Holden Commodore	1993	5 seater sedan
4WD Wagons		
Jeep Cherokee	1995	5 seater
Suzuki Vitara	1995	5 seater
Toyota Landcruiser 4500 GXL	1995	7 seater
Nissan Patrol Ti	1995	7 seater
Toyota RAV4	1995	5 seater
Toyota Landcruiser	1994	Troupe Carrier
Mitsubishi Pajero	1993	7 seater
4WD Utilities		
Mazda B2600	1990	Dual Cab 5 seater
Holden Rodeo DLX	1995	Dual Cab 5 seater
Mitsubishi Triton	1995	Dual Cab 5 seater
Toyota Hilux	1993	Dual Cab 5 seater

The vehicle was tilted with the driver's side at the low side of the deck. The wheels on the driver's side were chocked to prevent lateral movement of the vehicle. A steel section 50 mm high was placed against the tyre wall of the 4WD's and 25 mm high section for the passenger sedans (referred to as a trip rail). The reason for the increase in trip rail height for the 4WD vehicles is that the tyre excessively bulges over the top of the 25 mm high trip rail. Safety chains were loosely placed around the axles to secure the vehicle to the platform yet not to influence the suspension movements of the vehicle.

A trial tilt was conducted to ensure the chains were not influencing the vehicle's movements and to obtain an approximate tilt angle. Following the trial tilt, two tilts were conducted measuring and recording the load shifts and angles of the tilt deck and vehicle body. The repeat test was conducted to provide data on experimental variation.

The tilt deck is raised at approximately 6 degrees per minute. One side of the deck is supported and raised by hydraulic rams and the other is hinged. Accelerometers were clamped to the tilt deck and the vehicle body. This allowed accurate monitoring of the deck angle and the roll of the vehicle body on its suspension. A 1g accelerometer was attached to the deck and vehicle body with its active axis perpendicular to gravity. When the platform is raised the output voltage increases.

One side of the platform (high side) is supported by four load cells. This allows measurement of the load remaining on the deck on each axle during the tilting phase. Monitoring the remaining load establishes whenwheel lift has occurred. Data was recorded at a rate of two samples per second.

The maximum tilt deck angle of the facility was approximately 40 degrees. This was insufficient to reach wheel lift for some of the vehicles. Ramps were placed on one side of the deck to give the vehicle an initial nominal angle, which was determined by the track width and ramp height. Table 3 shows measurements which were taken from each vehicle.

TILT TABLE RESULTS

Table 4 contains the results from the tilt table tests. The vehicles are listed in order of decreasing roll threshold. The angle stated in the table is the maximum tilt angle which causes one of the wheels on the high side to loose contact with the deck. The values are average values calculated from the two tilt tests conducted per vehicle. With exceptions to the Mazda B2600 and Toyota Landcruiser Troupe, the tilt deck results were within one percent of each other. The Mazda and Toyota had a variation of up to three percent between tests.

Table 3. Vehicle Measurements

Vehicle Type	**Seat Height (mm)**		**Track Width (mm)**		**Roof Height (mm)**	**Wheel Base (mm)**	**Front Axle Mass (kg)**	**Rear Axle Mass (kg)**	**Total Test Mass (kg)**
Passenger Cars									
	Front	**Rear**	**Front**	**Rear**					
Mitsubishi Magna	420	470	1545	1525	1355	2720	1000	710	1700
Ford Laser	430	485	1460	1460	1400	2610	810	620	1440
Holden Commodore	490	500	1500	1495	1365	2720	880	810	1690
4WD Wagons									
Jeep Cherokee	735	780	1466	1472	1530	2590	950	1010	1960
Suzuki Vitara	670	760	1460	1460	1655	2495	770	910	1680
Toyota Landcruiser GXL	850	900	1610	1605	1755	2850	1190	1390	2580
Nissan Patrol Ti	810	795	1585	1585	1680	2970	1190	1320	2510
Toyota RAV4	650	710	1460	1460	1620	2420	820	870	1690
Toyota Landcruiser (Troupe)	930	x	1430	1405	1925	2955	1370	1720	3090
Mitsubishi Pajero	830	915	1430	1440	1745	2730	1030	1420	2450
4WD Utilities									
Mazda B2600	750	825	1455	1435	1645	3005	1060	1040	2100
Holden Rodeo DLX	705	735	1410	1405	1645	3030	1010	1020	2030
Mitsubishi Triton	775	840	1405	1420	1675	2960	1020	1100	2120
Toyota Hilux	-	-	1410	1430	1680	2850	1110	1130	2240

Note: the weighbridge rounds to the nearest 10 kilograms. Weighing of individual axles required repositioning of the vehicle, hence discrepancies in the order of 10 kilograms may be evident between the sum of the two axles and the gross mass.

- data not recorded x no rear seat

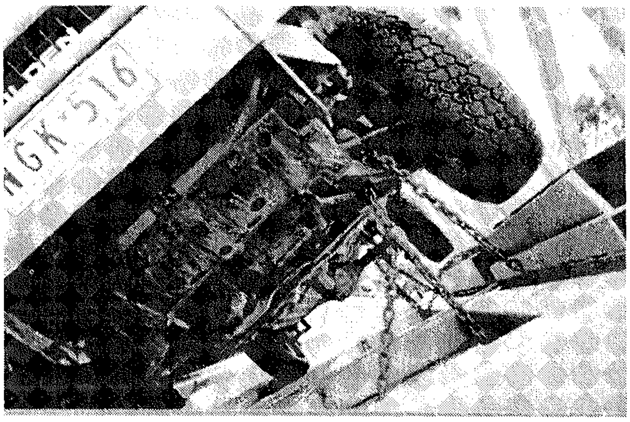

Figure 2 Wheel Lift - Holden Rodeo

Table 4. Tilt Angles

Vehicles	Angle (degrees)		Roll Threshold TTR
Passenger Cars	**Vehicle**	**Deck**[b]	
Mitsubishi Magna	54.29	49.64	1.18
Ford Laser	55.64	49.11	1.15
Holden Commodore	54.14	48.29	1.12
4WD Wagons and Utilities			
Jeep Cherokee	44.83	44.62	0.99
Mazda B2600	46.84	43.86	0.96
Suzuki Vitara	47.56	43.53	0.95
Holden Rodeo	49.07	42.64	0.92
Toyota Landcruiser	50.62	42.25	0.91
Mitsubishi Triton	51.03	42.10	0.90
Nissan Patrol	47.27	41.03	0.87
Toyota RAV4	46.34	39.64	0.83
Toyota Hilux	44.59	37.37	0.76
Toyota Land. Troupe	41.41	35.37	0.71
Mitsubishi Pajero	42.45	35.22	0.71

[b] deck angle is inclusive of the initial angle impart to the vehicle due to the ramp

Wheel lift occurred at the front wheel first for all vehicles, except the Toyota Hilux lifted the rear wheel first, and Holden Rodeo lifted both front and rear at the same time, Figure 2. This shows that the Holden Rodeo is a balanced vehicle ie. same split mass distribution per wheel. The reason for the Toyota Hilux lifting the rear wheel before the front in this instance would be due to two reasons: the rear suspension having less vertical travel than the front, and the split distribution of mass on each wheel may have been different.

FUTURE WORK

An investigation has commenced to determine if the vehicles which recorded a low static rollover threshold have a higher frequency of rollover accident involvement. Vehicle data is to be obtained for all rollover accidents which occurred from 1987 to 1994. This constitutes more than 5000 rollover accidents. In addition to this a comparison will be made to the equivalent US fleet.

DISCUSSION AND CONCLUSION

The passenger vehicles exhibited similar roll thresholds with 1.35 degree difference between this type of vehicle. However, the 4WD and utility range of vehicles showed a much larger variation in roll threshold or static stability. The variation between make and model being 9.4 degrees. Not surprisingly, the 4WD's and utilities rollover at significantly smaller tilt angles than passenger cars.

A less predictable result was that there is no clear difference in the rollover propensity between 4WD wagons and the range of utilities tested.

As expected the location of the vehicle's centre of gravity is an important factor in determining stability. The track widths varied by as much as 200 mm between vehicles, yet the Jeep Cherokee, which had the greatest resistance to roll, had a relatively small track width, implying a low centre of gravity.

If a comparison is made between the Jeep and the Pajero, the track widths are similar in dimension, however, the front and rear seat heights are 95 and 135 mm higher for the Pajero, together with a roof height 215 mm higher than the Jeep. This shows that the vehicle body is higher off the ground. In the laden condition this would significantly raise the centre of gravity of the Pajero, resulting in a lower roll threshold.

Table 4 shows that the 4WD vehicles with greater stability had less vehicle body roll on their suspension, that is a smaller difference between the deck and vehicle body angle.

4WD's have been an increasing popular family vehicle in Australia, in many cases because they can carry more passengers compared to sedans. Many buyers also consider them to be safer, yet the indications are they are not. The purpose of this project was to investigate and document one aspect of their relative safety compared to sedans. The longer term aims of the project are to investigate a link between rollover propensity and actual involvement in rollover crashes. Unfortunately, the available crash records are not easy to analyse in this regard. However, in time this will be completed. The eventual aim is to make this information available to consumers to assist in their purchase of a safe family vehicle.

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INFLUENCE OF ABS ON ROLLOVER ACCIDENTS

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Paper Number 96-S5-O-04

ABSTRACT

In the present study, we analyzed human behavior in driving a vehicle, identified the causes of increased rollover accidents involving ABS-equipped vehicles, and sought solutions to this problem.

Analysis of ordinary driver's behavior in a panic situation shows that many of them: (a) press the brake pedal, but not hard enough to activate the ABS system, or (b) never press the pedal, concentrating too much on operating the steering wheel, or (c) press the pedal but release it part of the way. Under such operating conditions, most vehicles run off the course, and may even spin.

These analysis results suggest that increase in rollover accidents involving ABS-equipped vehicles is not attributed to the characteristics of the ABS, but mainly to drivers who become aggressive in their driving method, relying too much on the ABS, or who cannot activate the ABS properly when necessary.

Therefore, to prevent the number of rollover accidents from increasing it is essential to ensure that each driver understands the proper operation of the ABS, and to install appropriate mechanical systems to assist drivers who cannot operate the ABS properly.

INTRODUCTION

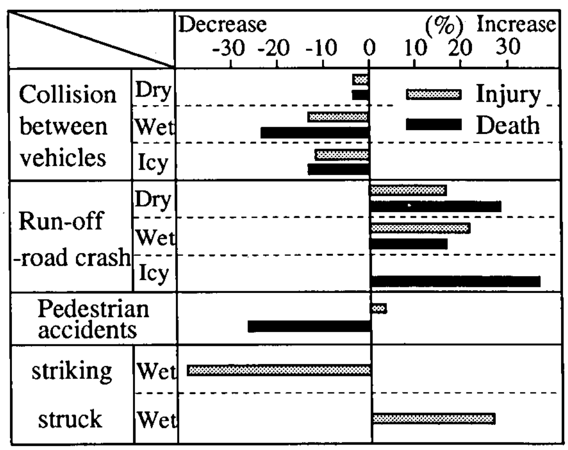
(1) Current Effect of ABS

The antilock brake system (ABS), a preventive safety device based on state-of-the-art technologies, has prevailed as it is considered to be an innovative measure to improve the ability of vehicles to avoid accidents.

As for the effect, however, the Insurance Institute for Highway Safety (IIHS) has reported that vehicles with and without ABS are the same in terms of insurance claim rates, as shown in Table 1. [Ref.1] A NHTSA technical report compares the accident rate between vehicles with and without ABS, using the same model. The comparison reveals, as shown in Fig 1, that those with ABS have been involved in a smaller number of collisions between vehicles and between vehicles and pedestrians, but a larger number of off-road crashes, so that the total number of accidents has not been decreased. [Ref.2] According to GM's report vehicles with ABS have caused a smaller number of pedestrian accidents, but a larger number of rollover accidents, as shown in Fig 2. GM's survey result (Fig 2) shows that since the number of accidents on wet roads has decreased, the total number of accidents involving ABS-equipped vehicles has decreased by 3%. [Ref.3]

Table 1. Insurance Claim Rate

Vehicle Name	Vehicle with ABS	Vehicle without ABS
Chevrolet Cavalier	96	95
Chevrolet Corsica	100	99
Chevrolet Lumina	66	69



Dry:Dry road Wet:Wet road Icy:Icy road

Fig 1. NHTSA's Survey Result

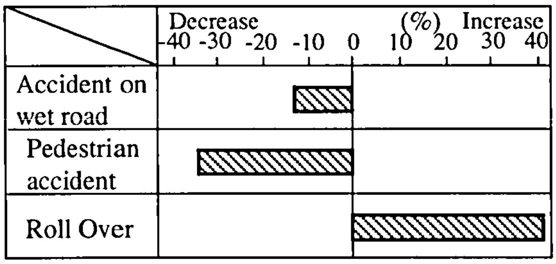


Fig 2. GM's Survey Result

(2) Reason for Ineffectiveness of ABS in Decreasing Accidents

To decrease the number of accidents, it is necessary to investigate the causes of accidents and take appropriate countermeasures.

Fig 3 shows the breakdown of the run-off-road crashes from Fig 1. We speculated about the causes of each type of run-off-road crash. Rollover and side impact accidents look different because the obstacles are different. However, the causes of these accidents are considered to be essentially the same: in both accidents, driver's handling in avoiding an obstacle possibly caused the vehicle to spin, because of high vehicle speed, as well as failure in activating the ABS. In most frontal impact accidents, the driver pressed the brake pedal until the ABS functioned, but steered the vehicle off the course because the steering remained effective; this situation, again coupled with high-speed driving, possibly caused the accident.

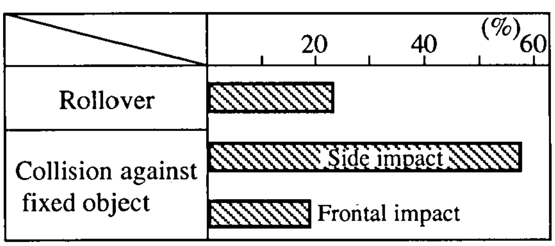


Fig 3. Breakdown of Run-off-road Crashes

Major possible causes of these accidents can be summarized as follows:

① The driver runs the vehicle at speed faster than normal, mistaking the ABS for a "more efficient and powerful brake."

② When avoiding an obstacle, the driver concentrates on steering and does not press the brake pedal until the ABS works.

③ Steering is effective even with full braking—the characteristic of the ABS.

Cause ① is supported by the aforementioned IIHS report describing the driving speed of vehicles with ABS as higher by 2 to 5 miles/h compared to those without ABS.

To confirm causes ② and ③, we experimentally studied ordinary drivers' behavior in operating vehicles.

ORDINARY DRIVERS' OPERATION

Ordinary drivers' behavior in operating vehicles was studied regarding three tasks: single and double lane changing on a straight course, and rounding a curve.

(1) Emergency Single Lane Changing to Avoid Obstacle

The experiment followed the method used by Tsutsui et al.[Ref.4] Specifically, while a vehicle is being driven along a two-lane straight course, an obstacle is suddenly thrust in front of the vehicle, producing an emergency situation that requires the driver to avoid the obstacle.(Fig 4) Drivers are not informed of this in advance. For the experiment, the vehicle speed was 80 km/h, because our analysis of actual traffic accidents shows that many serious accidents occur at 70 to 90 km/h. The obstacle was thrust in front of the vehicle leaving 1 - 2 s for reaction. Thirty-eight male and female ordinary drivers (non-professionals) aged between 22 and 56 took part in this experiment as subjects.

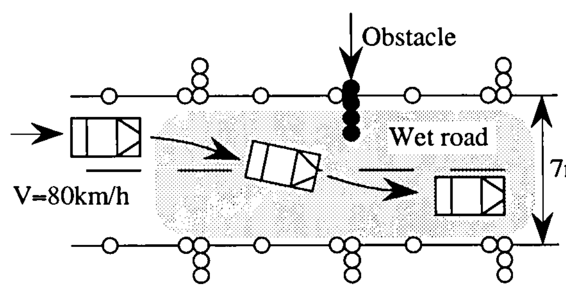


Fig 4. Single Lane Changing

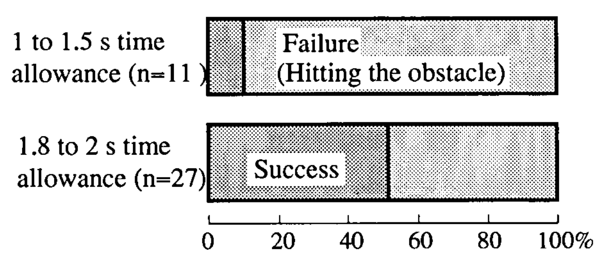


Fig 5. Percentage of Avoiding Obstacle

Fig 5 shows the result of the experiment. When the obstacle was thrust with 1 and 1.5 s time allowance, 91% of the subjects failed to avoid the obstacle. With a time allowance of 1.8 to 2 s, although the percentage of drivers successfully avoiding the obstacle increased, 48% failed. Each driver behaved differently to avoid the obstacle. Drivers' behavior can roughly be divided into "only braking," "braking and steering," "only steering," and "virtually no action," as shown in Fig 6, along with the percentage of each behavior. Fig 7 is the corresponding chart indicating the result of experiments by Tsutsui et al. conducted at 50 km/h. Comparison between these two sets of results reveals that, in our experiment conducted at 80 km/h, more drivers used steering alone to avoid the obstacle, and some took almost no action. Fifty percent of drivers did not activate the ABS. These results imply that drivers tend to resort more to steering as the vehicle speed is increased. On the other hand, some concentrated only on braking.

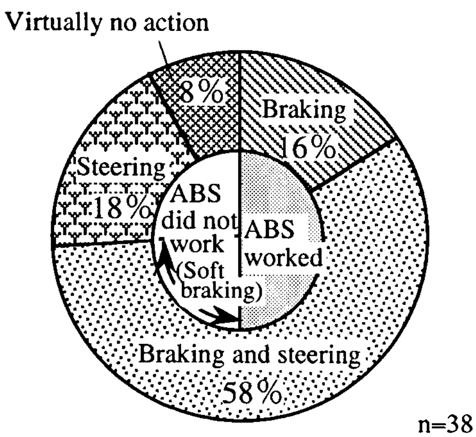


Fig 6. Driver's Behavior (at 80 km/h)

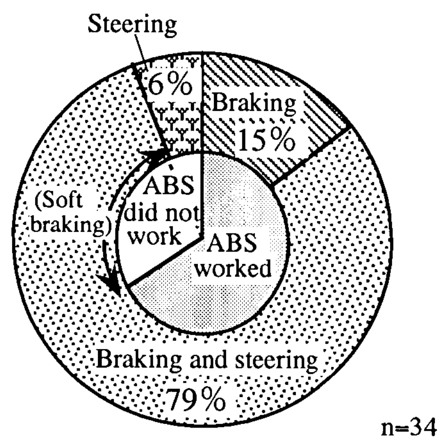


Fig 7. Driver's Behavior (at 50 km/h)

In the above experiment, when the obstacle was thrust with short allowance time, only few drivers could swerve the vehicles quickly enough to avoid the obstacle. Therefore, no run-off-road crash occurred.

To study the vehicle behavior, we conducted a similar experiment using skillful drivers. The drivers were asked to simulate ordinary drivers' behaviors in different combinations.

The experiment used the same test course as shown in Fig 4. The steering wheel angle in the initial steering was between 90° and 160° . Table 2 gives various cases of operating conditions for this experiment as well as for the experiment using ordinary drivers mentioned above.

Fig 8 shows the vehicle behavior and final vehicle position under each case of operating conditions, with the steering wheel angle at 90° . The broken line represents the boundary of 7 m wide two-lane course plus 3 m wide shoulder. A vehicle moving beyond this line on the right side was counted as a run-off-road crash.

Table 2. Operating Conditions

	Braking Simultaneous with Steering	Braking after Steering
Without ABS Full braking	Case1	Case2
With ABS Full braking	Case3	Case4
ABS does not work Soft braking	Case5	Case6

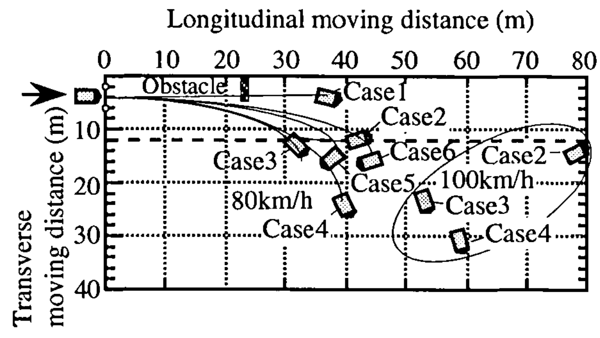


Fig 8. Vehicle's Behavior under Full Braking and Full Steering Conditions

Under case 1 conditions, the wheels were locked, making steering ineffective, so that the vehicle continued straight ahead and collided with the obstacle. Under case 3 conditions which correspond to ③ under possible causes of accidents given in Section 2-(2), the driver could avoid the obstacle owing to the characteristic of the ABS which allows steering to remain effective even under full braking. However, the vehicle moved beyond the boundary line, and was counted as a run-off-road crash. Under case 2 or 4 conditions, the vehicle could avoid the obstacle because steering was effective until brake pedal was pressed. Under case 2 conditions, however, the wheels were locked under full braking, so that the vehicle slid under inertial force in the same direction it was going before it had been steered to avoid the obstacle . Under case 4 conditions, since steering remained effective as under case 3 conditions, the vehicle was steered stably toward the boundary line. Since the brake pedal was pressed later, the vehicle went further beyond the boundary than under case 3 conditions. Under case 5 or 6 conditions, which correspond to ② of possible causes of accidents, although the ABS did not work due to soft braking, steering was effective to some extent. Therefore, the vehicle was directed toward the boundary. Since the ABS did not work, the vehicle became directionally unstable, and finally spun. Under all operating conditions, the vehicle instability and the stopping distance become larger at higher vehicle speed, as represented by cases 2, 3 and 4 in Fig 8. In addition, as the steering wheel angle was increased in the range from 90° to 160° , the vehicle was oriented more to the right, resulting in larger transverse moving distance. However, the general tendency of vehicle behavior under operating conditions of cases 1 through 6 was the same regardless of the steering wheel angle.

These results show that off-road crashes occur when the ABS works or when the brake pedal is not at all pressed or is pressed too softly. When the ABS works, the vehicle is directionally stable, and runs off the road front first. When the ABS does not work, on the other hand, the vehicle becomes unstable, and runs off the road sideways. From this vehicle behavior and the fact that drivers tend to press the brake pedal less firmly as the vehicle speed is increased, it can be said that most rollover or side impact accidents are caused not by the ABS function itself but by drivers' failure to properly activate the ABS. While a vehicle with ABS is driven at around 80 km/h, if the brake pedal is pressed to the full simultaneously with steering as in case 3, the ABS will work, causing the vehicle to run off the road. However, since the vehicle speed in running off the road is sufficiently low, frontal impact etc. would not have fatal consequences. Thus, frontal impact accidents, which account for about 20% of all traffic accidents, are caused primarily by high-speed driving or by the operating conditions of case 4, in which braking comes after steering so that activation of the ABS is delayed.

(2) Emergency Double Lane Changing to Avoid Obstacle

The test course for double lane changing is basically the same as that used for single lane changing, except that a stationary obstacle is installed on the passing lane, as shown in Fig 9. In this test course, vehicles changing lanes to avoid the thrust obstacle must return to the original lane to avoid the stationary obstacle. The vehicle speed was 70 km/h. The obstacle was thrust with 2 s time allowance. The road surface was slippery with a friction coefficient of 0.45. Thirty-one unskilled ordinary drivers took part in this experiment.

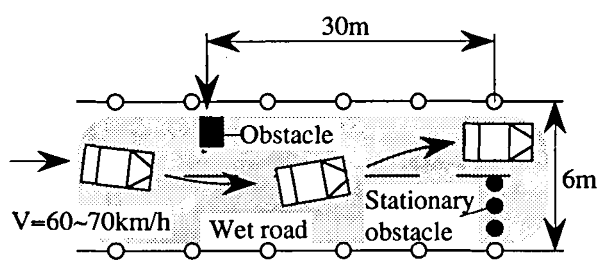


Fig 9. Double Lane Changing

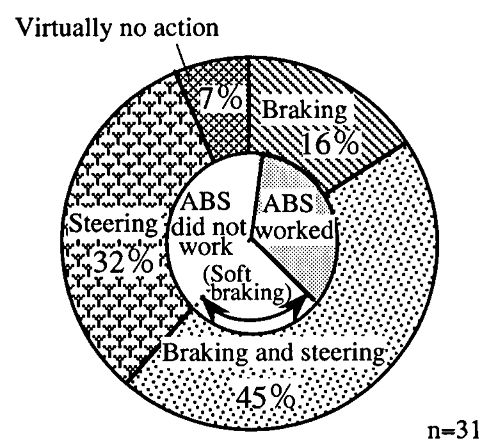


Fig 10. Driver's Behavior

The result was compiled in the same form as that for the single lane changing experiment, as shown in Fig 10. As compared with the results shown in Figs 6 and 7, the percentage of steering only and that of failing to activate the ABS are larger. This is because the drivers who used steering before braking to avoid the thrust obstacle also used steering first to avoid the stationary obstacle on the passing lane, and had no time to press the brake pedal or could not press it strongly enough to activate the ABS. Not only the vehicle speed but also the task complication is increased, drivers tend to resort more to steering.

The percentages of "virtually no action" and " braking only" are almost the same as those in Fig 6; it is natural that those who could take no action or only braked the vehicles without steering to avoid the thrust obstacle are not influenced by the stationary obstacle.

Fig 11 shows the experiment results concerning success or failure in avoiding the obstacles, as classified in three groups: "successful" which means the driver successfully avoided the obstacles; "run-off-road" which means the vehicle was steered to avoid the thrust obstacle, but spun or drifted out of the course ; and " hitting" which means the vehicle hit the thrust obstacle. Driver's behavior for each group is also shown in Fig 11. Only about 1/3 of the vehicles successfully avoided the obstacles, and about 1/3 ran off the road. Both " successful" and "run-off-road" cases resulted from driver's behavior of "steering" only or "steering and braking." "Hitting" vehicles resulted from driver's behavior of "no action" or "braking" only.

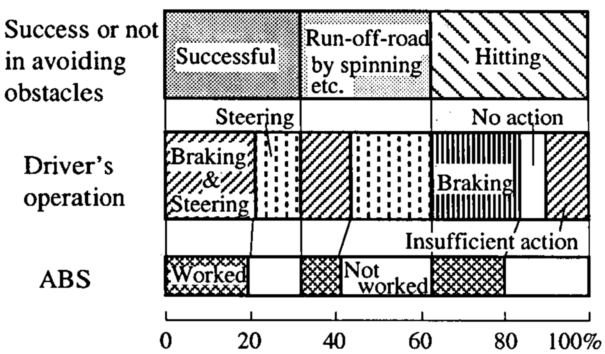


Fig 11. Success or Failure in Avoidance and Driver's Operation

Why could some drivers successfully avoid the obstacles while others ran the vehicles off the road, using the same operating technique? One reason is the vehicle speed. The percentage of "steering" alone is higher for the "run-off-road" vehicles than for the " successful" vehicles, which means that the drivers of the former vehicles did not reduce the vehicle speed sufficiently to suit their driving skill. Investigation revealed that some drivers pressed the brake pedal too softly to activate the ABS, causing the vehicles to spin. Some vehicles ran off the road although the ABS worked. This vehicle behavior resulted from driver's insufficient braking operation; they pressed the brake pedal but released it part of the way. (Fig 12, 13)

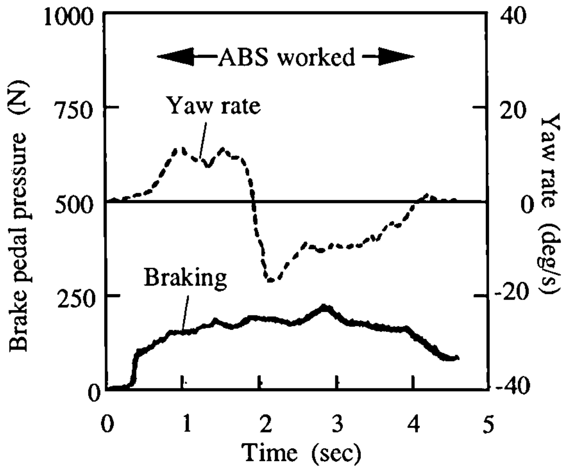


Fig 12. Driver's Operation and Vehicle Behavior (Successful in Avoiding Obstacle)

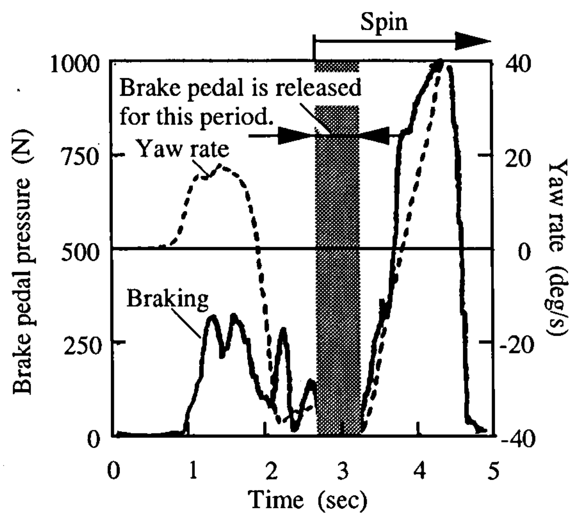


Fig 13. Driver's Operation and Vehicle Behavior (Run-off-road)

Therefore, the cause of "run-off-road" cases can be attributed, in part, to the fact that ABS allows steering to remain effective even with full braking, but are mostly attributed to driver's behavior. Ordinary drivers tend to run the vehicles at a speed beyond their driving abilities, and many of them only operate the steering wheel and do not or cannot press the brake pedal to avoid an obstacle. Or if they press the brake pedal, they do so too softly to activate the ABS, or release the pedal part of the way. As a result, they cannot take full advantage of the ABS function as in the case of the aforementioned single lane changing.

(3) Rounding a Slippery Curve

The test course was a right-hand curve 60 m in radius, leading to a straight. The road surface was slippery. Forty drivers were given the task of running vehicles along this test course at about 60 km/h, and we studied driver's behavior and vehicle behavior.

Fig 14 shows the result. Forty-five percent of the drivers entered the curve at a speed beyond their driving ability, and could not stabilize the vehicle posture. As a result, the vehicles spun or drifted out. Including those who successfully traced the course, most drivers resorted to steering to stabilize the vehicle posture; only few drivers pressed the brake pedal. Therefore, most drivers could not take advantage of the ABS function. Vehicle behavior in running off the road can be classified into four patterns as shown in Fig 15. Our survey on actual traffic accidents reveals that patterns 1, 2 and 3 are often followed by vehicles that cause single-vehicle accidents on a curve—this fact verifies the appropriateness of this experiment. Pattern 4 is followed by vehicles when the driver presses the accelerator pedal in mistake for the brake pedal. This mistake is not extraordinary, accounting for several percent of our various experiments, and can not be ignored.

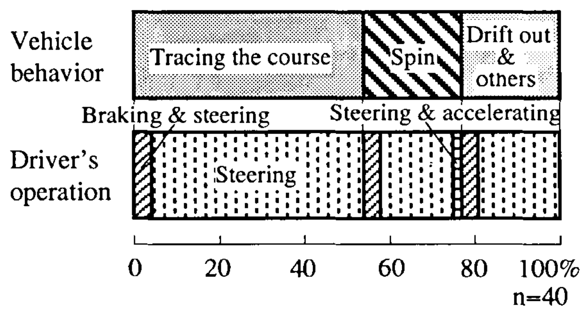


Fig 14. Vehicle Behavior and Driver's Operation

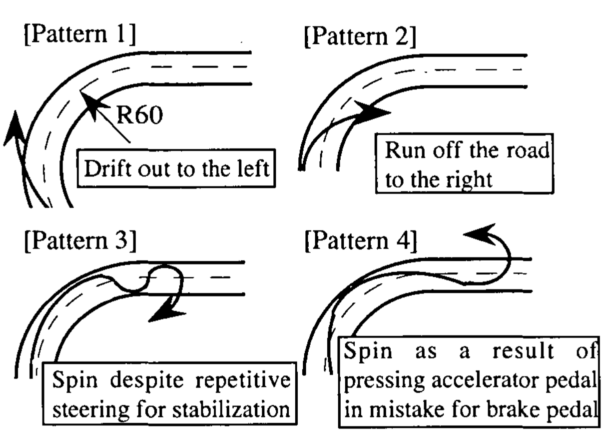


Fig 15. Vehicle Behavior on a Curve

CAUSES OF INCREASED ROLLOVER ACCIDENTS OF ABS-EQUIPPED VEHICLES AND COUNTERMEASURES

The experiments of Section 3-(1), -(2) and -(3) revealed the characteristic behaviors of ordinary drivers in a panic, as summarized below:

(a) Drivers do not press the brake pedal simultaneously with steering.

(b) At higher vehicle speed, and given a more complicated task, drivers concentrate more on steering and less on braking.

(c) Drivers make mistakes in operation.

Many ordinary drivers have such characteristics. So when they need the ABS assistance, they do not take full advantage of the ABS function. However, they tend to increase vehicle speed when the vehicle is equipped with the ABS, believing that the ABS is a more efficient brake. Such driving behavior is the primary cause of increase in rollover accidents and other run-off-road crashes involving ABS-equipped vehicles. We asked the drivers subject to our experiments, about how they usually press the brake pedal of ABS-equipped vehicles in an emergency. Sixty percent answered that they pump the pedal as they do in vehicles without ABS. They misunderstand the ABS function.

Therefore, there is a very strong possibility that the ABS can reduce the number of run-off-road crashes of vehicles. For this purpose, it is the most important that drivers operate ABS-equipped vehicles with accurate understanding of the characteristics of the ABS. First, drivers should not operate ABS-equipped vehicles at higher speeds than they would operate vehicles without ABS. Second, full braking is necessary for the ABS to work properly. Third, unlike vehicles without ABS, those with ABS can be steered more stably with full braking than with pumping the brake pedal. Drivers should approach the vehicle with full understanding of these characteristics of the ABS. Nevertheless, even given proper knowledge of the ABS, some drivers may not be able to bring out the effect of the ABS. To assist such drivers, vehicles should be provided with an assistive mechanism that operates the brake for the driver.

TECHNOLOGY FOR IMPROVING TURNING STABILITY

One example of such assistive mechanism is the Vehicle Stability Control system (hereinafter referred to as "VSC"), which mechanically prevents the vehicle from spinning, a cause of rollover and side impact accidents. The following paragraphs describe the effectiveness of the VSC.[Ref.5]

(1) Function of VSC

If the front wheels drift on a right-hand curve, the vehicle cannot trace the curve, drifting out of the course, as shown in Fig 16. In such an occasion, if the vehicle is equipped with VSC, the VSC applies an appropriate braking load to each wheel while suppressing the engine power, thereby decelerating the vehicle. As a result, the tires recover the design traction, enabling the vehicle to successfully round the curve. If the rear wheels drift, the VSC applies a braking load to the front left wheel, to produce a counter-clockwise moment, thereby preventing right-hand spin.

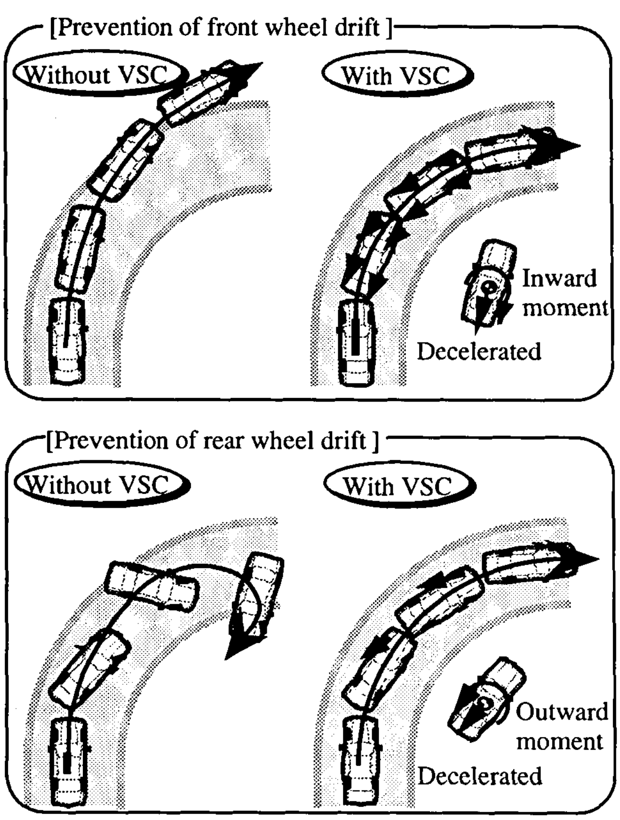


Fig 16. Operation and Effect of VSC

(2) Suitability to Ordinary Drivers

In developing the VSC system, we studied the operating behaviors of ordinary drivers in detail, to confirm the suitability of the system to them.

Ordinary drivers were given the same tasks as in double lane changing and rounding a curve, and we studied and compared the vehicle behavior between vehicles with and without VSC.

Fig 17 shows the vehicle behavior in double lane changing. No vehicle with VSC spun out of the course. Naturally, the VSC was useless in the case of drivers who steered the vehicle improperly and hit the thrust obstacle. These drivers should be assisted by an automatic braking/steering system, which is a future technical challenge for us.

Fig 18 shows the vehicle behavior in rounding a curve. No vehicle with VSC ran off the course by spinning, and only few vehicles with VSC drifted out.

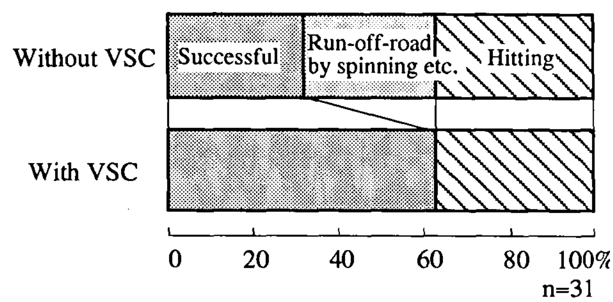


Fig 17. Vehicle Behavior in Double Lane Changing

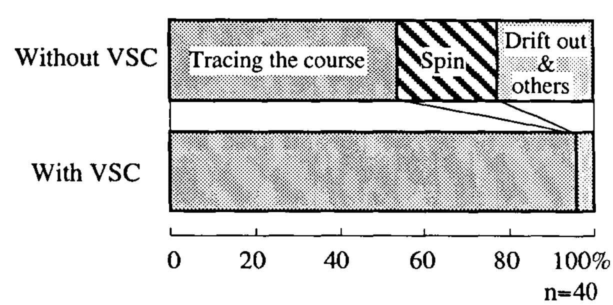


Fig 18. Vehicle Behavior in Rounding a Curve

Thus, the VSC is obviously effective for ordinary drivers, particularly in preventing vehicles from spinning, which can result in rollover or side impact accidents. This effectiveness is attributed to the characteristic of the VSC that it is turned ON/OFF not by the driver but by the G sensor and yaw rate sensor, to compensate for any difference in the driver's skill. That is, the machine compensates for human inabilities.

Even with this system, however, 5% of the vehicles drifted out of the course as shown in Fig 18, as a result of being driven too fast. However closely the machine can compensate for the driver's inabilities, no preventive safety device can reduce traffic accidents unless the driver understands that an accident will occur when the vehicle is operated improperly; the vehicle is not accident-proof.

CONCLUSIONS

Run-off-road crashes, involving rollover and side impact, of ABS-equipped vehicles have increased in number. Experimental analysis of human and vehicle behaviors in emergency situations revealed that the cause of the increase can be attributed to people driving vehicles at higher speed than usual, with overconfidence in the ABS function, and failure to activate the ABS due to improper operation.

To reduce run-off-road crashes, it is essential that drivers understand the proper methods of driving ABS-equipped vehicles. That is, they must drive the vehicles at moderate speed, and use full braking even while turning the steering wheel. In this regard, our activities towards educating ordinary drivers are important. The increase in run-off-road crashes of ABS-equipped vehicles can also be countered by installing the VSC.

Our future task is to develop mechanical systems capable of compensating for possible human error in driving vehicles.

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THE ROLE OF CALCULATION IN THE DEVELOPMENT AND TYPE APPROVAL OF COACH STRUCTURES FOR ROLLOVER SAFETY

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Paper No 96-S5-O-05

ABSTRACT

The paper reviews the experience with calculation in the development and Type Approval of coach structures according to the ECE Regulation 66 on rollover safety. The five main application areas considered are : structural development, Type Approval based on component tests and calculation of the safety structure, identification of the 'worst cases' and 'representative bays' for Approval and assessment of similarity when considering extensions of existing Approvals. Ability to predict beam collapse properties and optimise sections is also presented as a major development aid. Various quasi-static, 'quasi-dynamic' and full dynamic simulation approaches are also discussed and illustrated by practical examples. All the results are extensively supported by experimental evidence. Calculation-related interpretation of the Regulation 66 in UK is summarised. Finally, the paper reviews the options of achieving weight efficient designs by combining the calculations for rollover safety and normal service loading conditions.

INTRODUCTION

The ECE Regulation 66 on Rollover safety of Passenger Service Vehicles (PSV), developed in the late seventies [e.g. References 1 to 4], has recently been adopted in several European countries and Australia (ADR 59/00), with a strong likelihood of spreading across most of the European markets. Type Approvals can be granted on the basis of full scale rollover tests, bay rollover or pendulum tests and calculation combined with component tests (introduced largely on the basis of [5, 6]). Each method has advantages and limitations [7], but they all concentrate on preservation of sufficient residual space around the occupants.

The Regulation assumes *unbelted* occupants. If occupants are belted to prevent ejection, as recently suggested in Europe and Australia [8, 9], the safety structure will be exposed to a higher energy input, which may, perhaps, lead some day to amendments.

Rollover safety calls for a fresh approach to both the overall concept and detail body *design*. The safety 'superstructure' may comprise the complete body, two rollbars only, or any 'intermediate' solution, each option having advantages and drawbacks [7]. Design of *beams* and *joints* also has to address new issues of impact strength and energy absorbing capacity.

Calculation can cut time and cost of development of reliable and safe, but weight and cost efficient safety structures. To this end one has to identify priority issues, choose appropriate analysis tools, procedures and manpower and ensure that the results are available on time for the designers in a simultaneous engineering manner. In return, the better the design, the more controllable the collapse behaviour and the easier and more reliable the calculation.

The possible *analysis tools* span from hand calculations, over the 'general purpose' finite element software with some non-linear capabilities, simplified dynamics mass-spring models and dedicated quasi-static simulation programs, to the advanced explicit finite element codes.

Supporting *hardware* also ranges from hand calculators over personal computers and workstations to the powerful super-computers. Each approach also implies different profile (and cost) of *manpower* and management support issues. The choice should look at short, but but also at mid- and long-term effects (in practice, this is easier said than done in view of the 'environmental' conditions under which the industry works).

In the field of PSV rollover safety calculation concerns the following main areas, discussed below :

- structural development,
- Approval based on component tests and calculation of the complete safety structure,
- identification of 'worst cases' for Type Approval,
- identification of 'representative bays' for Type Approval tests,
- assessment of similarity when considering extensions of existing Approvals.

DEVELOPMENT OF SAFETY STRUCTURES

The safety performance of a bus body depends on both the overall collapse mode and on the behaviour of beams and joints. Following this principle, a 'hybrid development approach', where components are studied independently from the whole structure has distinct advantages.

Beam / joint collapse properties

Prediction of the elasto-plastic and deep collapse bending moment-rotation curves and energies absorbed by plastic hinges in prismatic thin walled beams, on the basis of section geometry and elastic/plastic material properties, is a routine practice with the aid of the CIC-SIM family of programs [10], i.e. :

(a) WEST (Figure 1(a)) concerns rectangular section tubes and also includes torsion collapse and automatic optimisation in terms of weight, cost, material, etc. for given maximum strength, energy absorbed and remaining strength after a specified hinge rotation angle ;

(b) SOUTH (Figure 1(b)) treats prismatic single cell thin walled tubes of arbitrary convex shapes with spot-weld connections ;

(c) TESTBASE processes test data bases on beams and joints, with selection based on criteria related to the vehicle, hinge location and load direction and various collapse properties ;

(d) XYPLOTS provides piece-wise 'smoothing' of test curves, fitting by analytical functions, or curve 'design from scratch'.

All programs have been validated by very extensive experimental work. They instantly produce the moment-rotation curves which are used as input data for collapse analysis of the complete structure.

Alternatively, a general purpose dynamics analysis program such as DYNA3D (discussed below) can also be used on components (Figure 1 (c)). Extraction of the moment-rotation curves for subsequent analysis of the whole structure calls for special treatment [11]. This approach is not really competitive in bus and coach structures and CIC uses it only on complex car body joints.

Quasi static collapse analysis of the complete structure with 'quasi-dynamic' effects

Quasi-static approach is adequate to describe the main effects in a rollover test due to :

(a) similarity of the static and dynamic 'sound' collapse modes, caused by low impact speed, small mass of the deforming structure in comparison with the whole vehicle and bending as the dominant collapse mode :

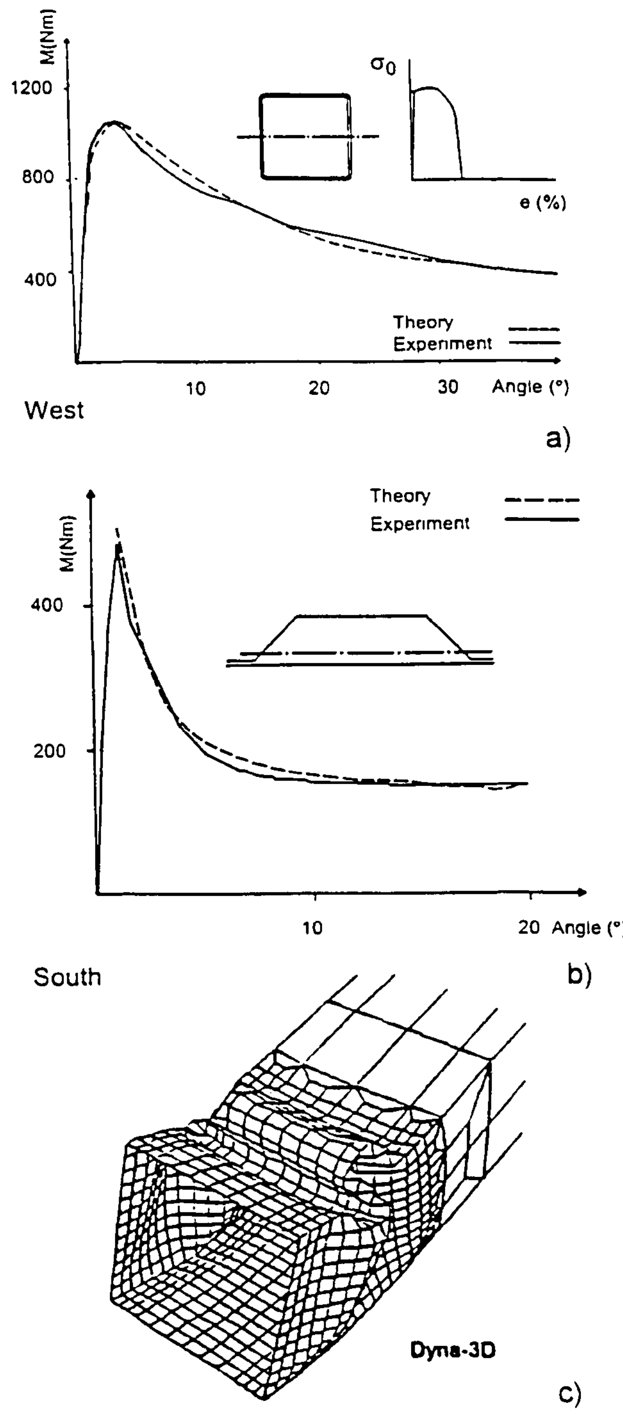

Figure 1 a, b, c - respectively : Theoretical and experimental moment-rotation curves of a rectangular section (WEST), spot-welded trapezoidal section beam (SOUTH) and deformed shape during bending collapse study of a rectangular section (DYNA3D)

(b) well established impact energies to be absorbed by the structure ;

(c) preservation of the survival space being the only safety criterion.

The CIC-SIM program SIMSTAT (updated CRASH-D) deals with quasi-static deep collapse of two and three-dimensional frameworks, allowing for the non-linear effects of large deformations, occurrence of plastic hinges with varying moment rotation curves in the elastic/plastic and deep collapse range and with some forms of axial collapse. The beam elements are composed of the elastic length and three non-linear rotary springs at each end (two for bending and one for torsion) whose stiffness is updated during each step of the analysis. Analytical treatment of the multiaxial collapse at a beam end has been developed [10, 12, 13] with a strong experimental support.

For example, the model of a complete bus body in Figure 2 comprises all the structural elements that are potential candidates for the safety structure. It does not include the chassis longitudinals (far enough from the deforming segments), windows, seats, panels and other small components. The moment-rotation curves are entered as input data and apply to any material or design, as long as they accurately reflect how the rest of the structure 'feels' the effect of the appropriate hinge. For example, they can represent the experimentally checked effect of panels, multiple cell beams simulated by one element, etc. In 'sound' steel structures, static moments can also be multiplied by a 'dynamic factor' (approximately 1.2) to allow for the increased energy absorbing capacity under higher strain-rate conditions.

The *boundary conditions* normally include translation constraints at the attachments to chassis and middle floor longitudinals (floor providing a strong and stiff restraint of relative movement in the lateral direction). As these are approximate, variations can be investigated, normally revealing a low sensitivity as long as the constraints are far enough from the floor/ side joints.

As regards the *load application*, the evidence from the accident investigations and full scale rollover tests indicate that roof collapse (and occupant ejection) are the prime causes of death. The full scale tests in UK, Belgium and Australia indicated that waistrail contact occurred *after* the maximum deformation of the roof, particularly in reinforced structures. The load is therefore applied to the cant rail only as a displacement generating a reaction force at 25 to 30^{o} to the floor. This complies with Approval by pendulum test and represents an approximate average value for most structures that pass the test. This angle can be updated if the analysis is split into phases, but the final results are not sensitive within the range of angles of interest.

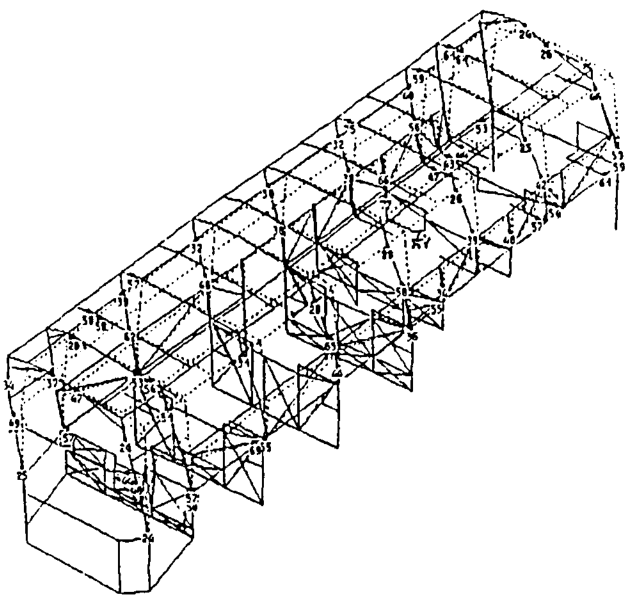

Figure 2 Coach model for quasi-static collapse analysis - original and deformed shapes with hinge location

If the structure is to be Type Approved by *component tests with calculation* or *pendulum impact*, the *energy* E* *to be absorbed* before intrusion into the residual space is taken from Regulation 66. This energy is a function of vehicle mass and the drop of the centre of gravity during rollover test from a 800 mm high platform (this drop is also influenced by the size and shape of the vehicle cross section). The roll moment of inertia was excluded from the E* formula after it was demonstrated [2] that the ratio of the mass and roll moment of inertia was very similar over the range of vehicles tested. The E* figure is also affected by the multiplier (0.75 in Regulation 66 and 0.62 in the ADR 59/00), allowing for the fact that not all of the vehicle kinetic energy is absorbed by the safety structure.

If the Approval is to be based on *rollover tests* (full scale or bay), the author's experience indicates that :

(a) the energy E* can be reduced to the level specified in the ADR 59/00,

(b) the energy distribution along the vehicle ought to broadly follow the general mass distribution,

(c) individual body segments should be designed to take their share of total energy by deforming to a similar distance from the residual space along the whole vehicle.

The calculation cycle at which the deforming structure contacts the residual space is established from the deformed geometry plots (Figure 3(a)) and then the energy absorbed is read from the energy vs. cycle diagram (Figure 3(b)).

The results also include hinge locations, internal load histories, energies absorbed by individual rings and many design and Approval oriented data.

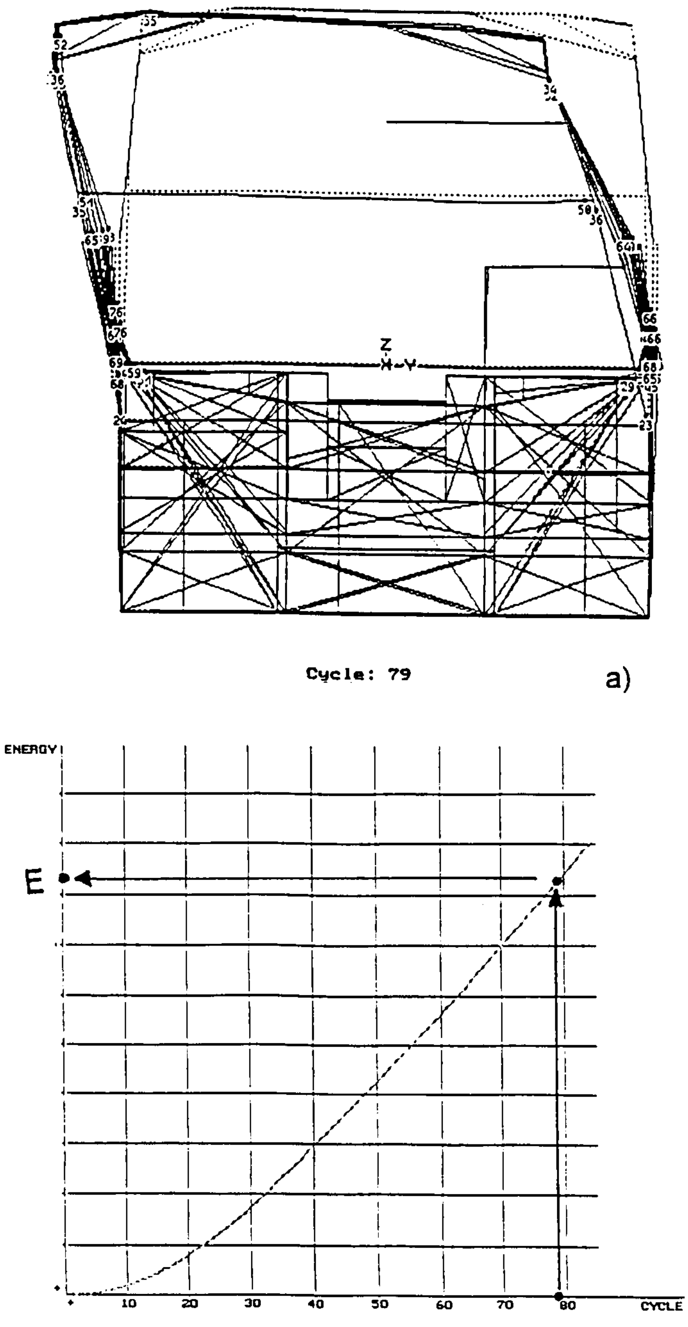

Figure 3 Contact between the pillar centreline and the (expanded) residual space (a) ; the appropriate calculation step is then used to determine the 'quasi-dynamic' energy (Ed) absorbed by the structure (b)

Dynamic simulation of the full scale rollover test

Dynamic simulation of the full scale rollover test presented here was carried out using the explicit formulation, general purpose finite element code DYNA3D (Lawrence Livermore Laboratory, or Oasys commercial version). The beam elements also consist of an elastic length with three non-linear rotational springs at each end. An improved version of that element has been developed at CIC [13] for multiaxial deep collapse of thin-walled beams and joints, together with a procedure to extract the component moment-rotation curves from the DYNA3D results.

The model for dynamic rollover simulation in Figures 4 and 5 has the same basic geometry and beam finite elements as the quasi-static model in Fig. 2 (the very 'weak' thin plates are included here for visual presentation only). The hinge moment-rotation curves are entered from the component analysis programs or tests. The whole superstructure of the coach model is mounted on a 'rigid chassis' at points which correspond to the constrained points in the SIMSTAT model. Rigid 'wheels', also attached to the chassis, define the contact points with the platform about which the coach rotates.

Dynamics analysis also requires the information on the *mass* and *moments of inertia* of the vehicle as a whole (about the longitudinal and lateral axes through the centre of gravity), but also of all the deforming parts and equipment attached to them. Small lumped masses were also added at various points in the structure, so that the overall mass and centre of gravity location were as intended and the inertia of the 'chassis' was adjusted to achieve the total rotational inertia close to the value assumed on the basis of vehicle mass [2]. Sensitivity studies may be required to investigate the effect of uncertain parameters.

The initial impact conditions are determined by 'rolling' the model from an inclined position of marginal stability where the vehicle centre of gravity is positioned slightly 'off-centre' of the edge of the lower pair of wheels (Figure 4 (a)). To save analysis time and cost, the whole model is assumed to be 'rigid' until very close to the first cant-rail contact with the ground (Figure 4 (b)), when the run is stopped and the model switched to the elastic/ plastic behaviour.

Reliability of results was very sensitive to the simulation of the *contact between the wheels* and *platform edge* (the design or performance of the platform edge is unfortunately not described in the Regulation). Various modelling options were tried with a final decision to connect the wheels to the platform with kinematics joints 'holding well' until being deleted completely at a restart time chosen by the user. The model then behaved

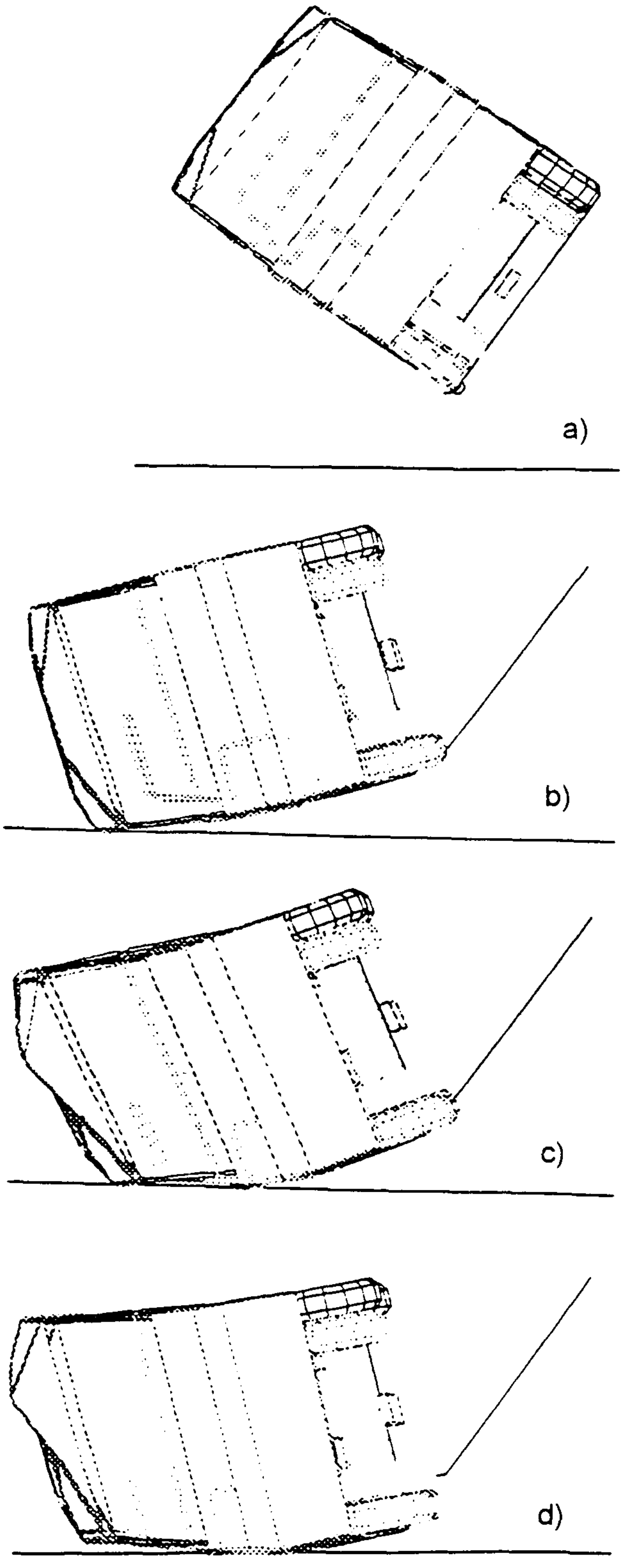

Figure 4 Dynamics rollover simulation of a full scale test - initial position (a), first contact with ground (b), maximum roof deformation (c) and sliding on the side with spring back (d)

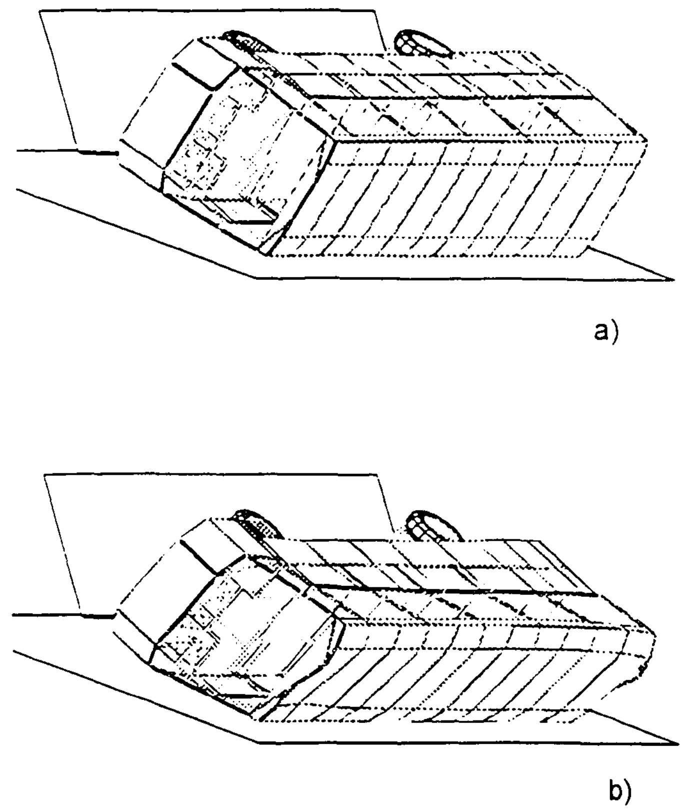

Figure 5 Dynamics rollover simulation of a full scale test - oblique view - first contact with ground (a) and maximum roof deformation (b)

in a similar manner as in the available full scale test, but also indicated that interaction between the wheels and platform deserves further study, both in the areas of platform design and numerical simulation.

The *contact between the cant rail and ground* also deserves attention in terms of contact friction and stiffness in the vertical direction (influenced by both the ground and local deformation of the roof corner).

After some adjustments, the roll kinematics and deformations in Figures 4 and 5 were in good agreement with the full scale test. The maximum deformation is shown in Figure 4 (c) and 5 (b) at around 90 to 100 ms after the ground contact, while Figure 4 (d) shows a later stage of the test with the body sliding on its side and with some elastic spring back.

The main interest here was again whether the residual space (defined by the half-trapezoidal shape fixed to the rigid chassis model) is intruded upon by the deforming side or roof structure.

Although the energy absorbed is not directly required for Type Approval purposes in this case, it is of interest to compare it with the energy E* required for the quasi-static analysis. The graph in Figure 6 shows the kinetic energy and the energy absorbed by the structure during the second phase of the analysis as a function of time. The kinetic energy reduces to a minimum close to the moment of maximum deformation, then starts rising again due to the elastic spring back of the structure. In the case presented, the maximum energy absorbed by the structure was around 80% of the kinetic energy at impact, which is compatible with factor 0.75 in the formula for E* in the Regulation 66. However, one should also bear in mind that the model does not include a number of items which also 'shake, break and slide' thus taking further part of the kinetic energy away from the beams and joints included in the model (hence the earlier preference of the energy E* as defined in the ADR 59/00).

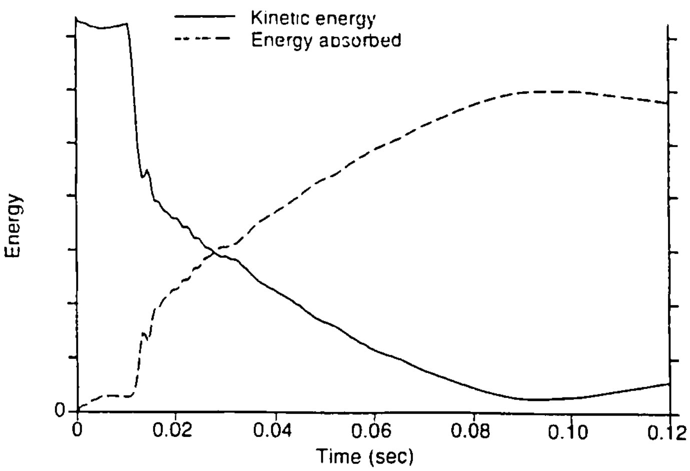

Figure 6 Change of kinetic and absorbed energies in time following the cantrail impact at time 0.01 s.

Dynamic or Quasi-Static Analysis ?

Both static and dynamic analyses can be used well to develop efficient rollover safety structures.

The main *advantages* of the dynamic over quasi-static analysis are :

(a1) the loading conditions better reflect the full scale rollover test, particularly as regards the distribution of loads and energies over individual safety rings where they have very different stiffness and strength and/or very uneven distance from the vehicle centre of gravity; in such cases there may be a significant difference in the roof deflections along the vehicle (this realistic scenario should, however be avoided by design) ;

(a2) the impact energy is calculated directly from the vehicle mass and inertia properties, hence no reference to the E* energy in R66 or ADR 59/00 is necessary (apart from checking the analysis results),

(a3) the waist rail contact is simpler to deal with (without restarts), which may be useful in cross section geometries with an excessive 'tapering' of the body cross section between the floor and roof .

The relative *drawbacks* of the dynamic analysis concern :

(d1) the 'speculative' content in the model is rather higher, with the results sensitive to some of these 'difficult' simulation parameters,

(d2) the costs of analysis are *very much higher*, not only in terms of investment in software, hardware and man-power, but also as regards direct costs of analysis.

A typical rollover analysis of a complete vehicle such as the one in Figure 2 takes : on a 486 PC with 33 MHz between 60 and 90 minutes and 20 to 30 minutes on workstations for 50 to 100 cycles. This time depends only on model size and number of steps.

A single-pass simulation using the model in Figures 4 and 5, over 145 ms after the cantrail impact, took 3.5 hours of CPU time on a CRAY J90 over 152000 cycles (strongly affected by the time taken by sound to travel over the shortest deformable element). The difference from the SIMSTAT conditions is particularly significant in view of the iterative nature of developing the optimum structures and their numerical analysis models.

A Typical Development Procedure

The general development process is discussed in some detail in [7,14]. It typically starts with identification of the 'worst case(s)' and assessment of the existing structure(s) at the component and overall level, using the experience, programs and procedures above. The general concept of the superstructure is then discussed by examining alternatives and bearing in mind production and overall cost reasons. Experience can often resolve the broad issues, but numerical analysis can be essential to *quantify* important design parameters. The chosen safety structure is then optimised starting from the analysis model(s) that include sufficient surrounding structure to envelope the correct collapse mechanism. Several iterative loops may be necessary using the component prediction and optimisation programs and using their moment-rotation curves as input data for the analysis of the whole structure. In CIC projects with steel structures, the only tests now concern confirmation of material properties and the official Type Approval tests. The requirements are usually met in the first attempt and with only a small safety margin (i.e. structures were not over-designed).

TYPE APPROVAL BASED ON CALCULATION (WITH COMPONENT TESTS)

Approval by calculation has the following advantages :

(a) it saves time and cost by combining development and Type Approval information,

(b) it clearly shows the performance and contribution of each component,

(c) failures require redesign and testing of the faulty components only,

(d) approved parts can be used in various structures (particularly useful when building on different chassis, or having a wide variation of designs while using the whole body as the safety system),

(e) helps with many ambiguities associated with problems discussed under the headings below.

However, the method also requires clarification and interpretation of the Regulation 66 itself. The level of expertise required is still beyond many of the bus manufacturers and Type Approval Authorities and the approach (at least as adopted in UK) is somewhat more conservative than rollover testing.

In contrast to *development* approach (which is entirely up to the manufacturer), the acceptance of a calculation method and procedures for *Type Approval* is of an international importance. This issue is not clarified well in the Regulation and the matter deserves further attention [15]. The UK experience from background research and cooperation between the Vehicle Certification Agency and its Technical Service on practical Approvals indicated the following main points [7] :

The 'right' *software* for Type Approval should closely reflect the overall and component behaviour within the whole range of allowed deformations. It ought to be *experimentally validated* and fully *'transparent'* in terms of : *input data* (with clear correlation to the associated static and dynamic tests and design drawings, component by component), *simulation procedure* and *outputs* (including internal load histories at all beam ends, deformations, energy absorption etc., *warning* and *error messages* and continuing *support* and *development*. These were the guidelines when developing the CIC-SIM family of software which is formally accepted.

The analysis *procedure* can be described as *'Type Approval by numerical simulation of the overall collapse behaviour of the complete safety structure ('superstructure') on the basis of the static and dynamic tests of all the relevant beams and joints'*. For reasons discussed above, the Approvals are supported, for the time being, by quasi-static analysis, with experimentally proven 'dynamic factors' applied to static hinge moment-rotation test curves. These dynamic factors are determined by comparative static and dynamic tests on nominally identical test specimens. The permanent hinge angle (θd) is measured after dynamic test with a known impact energy 'Ed'. Using the static moment-rotation curve and θd, it is possible to calculate the static energy 'Est' required to produce the same maximum static deformation as during the dynamic test. The dynamic factor 'Kd' is then defined as Kd = Ed/Es, assuming that plastic deformations are much higher than elastic ones

The model must simulate separately all the elements of the superstructure and has to extend far enough from the deforming region to demonstrate diffusion of internal loads into the surrounding structure without overloading it and without imposing too stiff boundary conditions on the deforming segments. All the calculation input data describing the elastic/plastic and deep collapse behaviour of the components must be determined by static and dynamic *component* (i.e. *not material*) tests. Components reliably qualified as *non-failing* (with a margin of at least 20 %) may be described by the moment-rotation curves determined by program WEST or another suitable method. The loading direction must reflect the collapse of the whole structure when rolling on either side and the deformation energy absorbed prior to the intrusion into the residual space must be greater than the E* energy from the Regulation 66.

SELECTION OF THE WORST CASE SCENARIOS

Most manufacturers offer a range of models, all of which may have to meet rollover safety requirements. In order to rationalise the process and standardise the design, it is often appropriate to select the 'worst case' scenario(s) representative of a range of vehicle models and develop a common safety system. Worst case selection should allow for a combination of :

(a) the energy to be safely absorbed (to this effect the E* data can be useful for comparative purposes even in Approvals by rollover tests),

(b) the allowed deformation (residual space along the vehicle, allowing for roof racks, etc.),

(c) structural variations of the body, i.e. floor (and roof) height, chassis mounts, location and size of doors and emergency exits, location of window pillars, special equipment such as toilets, air conditioning, etc., joint and beam designs (e.g. continuous pillars vs. continuous waist rails), etc.

(d) mass distribution and position of the centre of gravity in the longitudinal direction,

(e) concept of the superstructure,

(f) economy of applying the same superstructure to a range of models (a single worst case may be too heavy for some models, not convenient in others, etc.).

Both designer and Approval Authority can indeed be presented with a wide and difficult choice. The scope can usually be narrowed by experience and engineering judgement, but there are frequent puzzles where parameter combinations need to be quantified beyond intuition. Calculation can often resolve such ambiguities in a far more effectively than testing.

One extreme case is when the whole structure is used for safety and there is a wide variety of designs. Some manufacturers have opted to develop calculation models for each and every design (most of which are only variations on the common 'theme'), based on Approved components. Development rationale has thus helped the Approval process too.

The other extreme is to go for two rollbars only (that may even be of identical design) to cover all cases, but such a solution is not likely to be most effective in the long run unless all vehicles are very similar in terms of weight and size.

The 'intermediate' cases are the most frequent and often indicate that different local reinforcements may be due in the vicinity of strengthened rings, depending on the model. For example, if the rearmost ring is strengthened into a 'rollbar', quite different waist and floor rails may be appropriate to bodies with or without emergency exits (that break the waistrail) and other still for rear doors (that break the floor rail). A single 'worst' case may not even be possible, let alone rational and calculation complying with Approval criteria can and has been used to clarify such issues.

SELECTION OF REPRESENTATIVE BAYS FOR APPROVAL TESTS

In order to save costs, Type Approval can also be based on rollover or pendulum tests of 'representative bays', i.e. segments of the body structure. It is up to the manufacturer to propose and Approval Authority to agree or otherwise which bays can be regarded as representative of the complete vehicle, or a family covered by a common worst case. The choice is again aggravated by the same problems as discussed under the previous heading. And again, approved calculations similar to those above can help clarify these issues.

EXTENSION OF EXISTING APPROVALS

As time passes by, the manufacturers are often faced with the need to introduce new models after having Type Approved some earlier, similar ones. Depending on the original selection of the worst cases and, even more so, on the method of the original Type Approval, the new models may not be undoubtedly covered by the existing documentation. The important question then is to assess *how similar* the new structures are to the earlier ones and what may be required to extend the Type Approval.

The problem is again very similar to the 'worst case' selection discussed above, only the cost implications can be much more severe. CIC was involved in several exercises of this type and calculation again proved to be an essential aid to both the Approval Authorities and the designers themselves.

THE OVERALL WEIGHT AND COST EFFICIENCY OF THE BODY STRUCTURES

Introduction of rollover safety implies weight increase in most coach structures. At the same time, most coach structures are already overweight, because the design teams are involved in urgent problems of the day and cannot afford the time to take a longer-term look. There is also a partially justified perception that weight reduction implies risks that companies cannot afford. However, commercial pressures regarding coach equipment, safety of seats and axle load limitations are also encouraging manufacturers to consider the weight and cost efficiency. Calculation has helped here too.

For example, calculations allowing for the service loading conditions only and using the elastic analysis models as the one in Fig. 7 have enabled mass reduction of the steel skeleton of several integral body coaches [16] of the order of 300 kg, and without any changes of the established production jigs. This creates ample reserve to introduce rollover safety features.

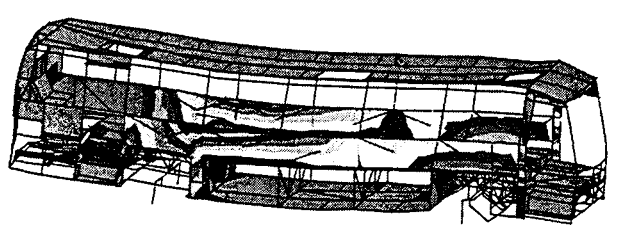

Figure 7- Coach analysis for weight reduction under normal service loading conditions (program NISA)

The practical problem, however, is that the overall weight and cost efficiency can only be achieved when both the *service loading and rollover safety are considered together*. The following main choices are at the designer's disposal (with other implications in [7]) :

(a) design the whole structure as the safety system - the strength is more evenly distributed and the general production process will be least affected (except for an increased number of local reinforcements) ; however, most of the beams and joints are designed to meet rollover safety rather than service load requirements which also increases the weight ;

(b) rely upon two rollbars designed (with the immediately surrounding structure) for rollover and setting the rest of the body free for production convenience, variation and weight reduction under service loads ; stronger (and more expensive) material can be used for rollbars only ; production difficulties may be met with heavier rollbars ;

(c) there is also a wide scope solutions falling between the extremes above.

CONCLUSIONS

Calculation can provide a very significant aid in development and Type Approval of bus and coach structures to the ECE Regulation 66 on rollover safety. The five main application areas are : structural development, Type Approval based on component tests and calculation of the complete structure, identification of the 'worst cases' and 'representative bays' and assessment of similarity when considering extensions of existing Approvals. The best weight and cost efficiency can be achieved when the structure is simultaneously designed and analysed to meet safety, service loads and production requirements.

Calculation concerns both components and complete structures and a range of software packages and analysis methods have been shown. Both quasi-static and full dynamics analysis of the rollover test can be used for successful development. However, quasi-static analysis still appears more reliable for Type Approval purposes, mainly due to the higher speculative content when selecting parameters to which the results may be sensitive.

Regulation 66 lacks specific minimum requirements that Approval by Calculation (and component tests) must meet. The technical and economic implications are serious. A coordinated international exchange of views and possible amendments to the Regulation would be justified.

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VEHICLE ROLLOVER PREVENTION, A BALANCED APPROACH TO A COMPLEX PROBLEM

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Paper Number 96-S5-O-07

ABSTRACT:

A variety of laboratory tests, both static and dynamic, have been employed by engineers with the intention of defining a measure to correlate with real world vehicle rollover propensity. Two static metrics, Title Table Angle (TTA) and Critical Sliding Velocity (CSV), have recently been put forth as measures that identify causal factors related to rollovers in a predictable manner. This paper explores the physical principles underlying TTA and CSV and their relationship to real world rollover performance. In addition the ability of TTA or CSV to accurately predict rollover propensity among various vehicles is examined.

INTRODUCTION:

Rollover crashes are severe events that involve every type of vehicle. The National Accident Sampling System, General Estimates System (NASS GES) data show that about 2.2 percent of all crashes are rollovers. Injuries occur more frequently in rollover crashes than in the average crash. **Figure 1** shows the distribution of injuries for rollover crashes. Injury was more frequent in this type of crash, with a fatality occurring in 2.3 percent of the crash-involved vehicles and an incapacitating injury in 17.1 percent of the vehicles. The injury data indicate that the rollover crashes are higher energy crashes than the average crash. The data strongly suggest that the rollover crashes involve higher speeds than the average crash.

The NASS CDS teams report the number of quarter turns involved in the lateral rollover crashes. As **Figure 2** shows, less than 15 percent of the crashes involve only a one-quarter turn. Over one-third of the crashes involve two-quarter turns. More than 42 percent of the crashes involve four or more quarter turns. These data, like the NASS GES injury data, also show that most rollover crashes are severe crashes that involve higher energy and usually higher speeds. The energy required to roll a vehicle four or more quarter turns is significant. Speeds in the range of thirty-five to fifty miles per hour or more are likely to be associated with rollover crashes. When an initial lower speed is involved, downslopes are usually also involved so that kinetic and potential energy are combined to provide the total energy necessary to produce multiple quarter turns of the vehicle.

Figure 3 shows the distribution of injuries using the Abbreviated Injury Scale (AIS) for the NASS CDS single-vehicle lateral rollover crashes. The injury reported is the maximum injury to any occupant in the vehicle. **Figure 4** presents the distribution of these injuries by quarter turns. As expected, injury severity increases with the number of quarter turns. About 70 percent of the major injuries (AIS 3-6) occur in four or more quarter turn rollovers.

Rollovers occur as a result of complex interactions among the driver, the vehicle, and the environment. Differences in risk due to how, when, where, and by whom a vehicle is used make it very difficult, if not impossible, to compare the rollover risk of different vehicles. In fact, the rollover risk of a given single make/model of vehicle varies depending on how, when, where, and by whom it is used.

DISCUSSION:

It should be noted that statistical correlation with rollover accident data are *not* the basis for maintaining that c.g. height, track width, suspension and tire movements, and roll moment of inertia are causal factors regarding rollover. The principles of physics reveal that these factors determine the amount of side to side load transfer in circumstances common to many rollover accidents (NHTSA Addendum to the Technical Assessment Paper: Relationship between Rollover and Vehicle Factors 1994, 11)).

In reality, static stability is only one of many vehicle factors that can potentially affect a vehicle's probability of rolling over. Other factors which have an effect on rollover crashes include handling and tire properties, suspension factors, vehicle geometry including wheelbase, the driver, and the crash environment. No one could reasonably deny that the physical characteristics of the vehicle are related to its rollover resistance. However, it has not shown that any measure of physical parameters such as tilt table angle (TTA) or critical sliding velocity (CSV) has any unique, overwhelming physical relationship with rollover resistance to real world rollover crashes. As noted above, rollover crashes are complex, nonrepeatable, dynamic events which are related to many factors. These factors influence the cause and final outcome of a rollover at all three stages of the crash: pre-rollover, rollover, and post-rollover. Let's take a closer look at both TTA and CSV and their ability to be a predictor of real world rollover performance.

Tilt Table Angle

The tilt table test involves placing the vehicle on a platform which is then tilted about an axis parallel to the vehicle's longitudinal axis and the vehicle's rollover stability is characterized by the angle at which the tires on the upper side of the platform lose contact with the platform and the vehicle begins to fall off the platform. **Figure 5** depicts the tilt table test.

Although the tilt table test results account somewhat for aspects of vehicle suspension, the tilt table test essentially predicts the minimum angle at which a vehicle will tip over onto its side as opposed to dropping back onto its wheels. It does not have any specific theoretical physical relationship with a rollover crash involving more than one-quarter turn. Other aspects of vehicle geometry enter when more than one-quarter turn is involved (e.g., vehicle size and shape).

There are many aspects of the tilt table procedure which influence the results in ways that are not indicative of real-world rollovers. For instance, the one inch high stop that is commonly located at the downhill side of the vehicle lowers the effective value of the center-of-gravity height (**h**) and induces a higher tilt table angle relative to the vehicle's T/2h. There are many other factors, however, which effectively reduce the tilt table angle. These include the raising of the center of gravity due to the unloading of the force normal to the tilt table as the tilt table angle increases and the effective reduction in track width due to suspension compliance. The magnitude of these factors (suspension loads are reduced in the range of 30 percent), which are artifacts of the tilt table test procedure, indicates that the test does not simulate a real world rollover. As stated earlier, the results obtained from this test predict only the minimum tilt angle required to roll a vehicle one-quarter turn.

Critical Sliding Velocity

According to the NHTSA, critical sliding velocity is defined as "a measure of the minimum lateral velocity required to initiate rollover, when the vehicle is in a tripping orientation" (NHTSA's Technical Assessment Paper, 1991, 4-35). The equation for CSV incorporates vehicle factors consisting of track width (**T**), center-of-gravity height (**h_{cg}**), roll moment of inertia (**I_{ox}**), and mass (**M**).

$$CSV = \sqrt{\frac{2gI_{oxx}}{Mh_{cg}^2}\left(\sqrt{\frac{T^2}{4} + h_{cg}^2} - h_{cg}\right)} \quad (1)$$

where

$$I_{oxx} = I_{xx} + M\left(\frac{T^2}{4} + h_{cg}^2\right) \quad (2)$$

The derivation of the equation which NHTSA relies upon is based upon the assumption that a vehicle can be accurately modeled as a rectangular, rigid block sliding laterally into a rigid stop. The total minimum energy required to rotate the block about the rigid stop to the point at which the center of gravity is directly above the impact corner of the block (i.e., the balance point) is used to calculate the critical sliding velocity. Upon impact with the rigid stop, some of the vehicle's initial energy is lost due to the impulse required to instantly change the direction of travel of the center of gravity. Momentum is conserved prior to and just after impact. The remaining kinetic energy in the system after the impact with the rigid stop is used to rotate the vehicle to its balance point.

The CSV is only an estimate, based on a rigid block model, of the velocity required to roll a vehicle one-quarter turn. **Figure 6** depicts the event that the NHTSA CSV model is intended to simulate.

Like the TTA, CSV is simply an estimator of one-quarter turn rollovers. It does not have a direct theoretical physical link with multiple quarter turn rollovers. In particular, it does not have a clear relationship with the four or more quarter turn rollovers in which most of the serious injuries occur.

Estimation of multiple quarter turn rollover velocities is much more complicated and requires additional factors (e.g., vehicle size and shape) and information pertaining to subsequent impacts and the environment. In multiple turn rollover crashes, energy is lost due to vehicle deformation and interactions with the environment. The velocities required to induce multiple quarter turns are much greater than those required to roll a vehicle one-quarter turn and are likely to be less sensitive to changes in vehicle parameters such as center-of-gravity height and track width.

Notwithstanding the inherent limitations noted above, can TTA and/or CSV provide useful measures of vehicle rollover performance? NHTSA's explanations suggest that TTA and CSV are very similar in their attributes and their ability to predict rollover experience. To examine the similarity of the two measures, the data from the 181 vehicles contained in NHTSA's rulemaking Docket 91-68 where both TTR and CSV were reported were evaluated. The values of TTR were converted to TTA and rounded to the nearest degree. Similarly, values for CSV were rounded to the nearest kilometer per hour (kph).

Next, a graph was produced plotting TTA versus CSV (**Figure 7**). The figure demonstrates that some vehicles have the same CSV but have widely varying TTAs. Each point on the graph represents vehicles with the same value of TTA and CSV. For example, there are sixty-nine vehicles that have a CSV of 17 kph. Their TTAs range from 44 degrees to 50 degrees so the sixty-nine vehicles are represented by seven points arrayed in a horizontal line. This range of 7 degrees is more than half the entire range (13 degrees) reported. Actually, except for two make/models, all the vehicles reported fall within a 9 degree range from 42 degrees to 50 degrees.

Similarly, there are twenty-nine vehicles that have a TTA of 46 degrees. The CSVs for these vehicles range from fifteen to nineteen kilometers per hour resulting in five points arrayed in a vertical line. The range for CSV values is 14 kph to 21 kph.

Figure 7 shows that vehicles with the same CSV have different TTAs and vice versa. The practical significance of this to the consumer is shown in the following figures. **Figure 8** presents five vehicles with a TTA of 46 degrees. The vehicles include a utility vehicle, cargo and passenger vans, and a small passenger car. A label showing TTA would indicate that these vehicles have the same rollover risk.

Figure 9 presents the same vehicles ranked by CSV. A measure showing TTA would rank the rollover risk of these vehicles as the same. A measure showing CSV would rank them differently.

Figure 10 presents the actual rollover experience for the five vehicles with the same TTA. The actual rollover experience varies widely and cannot be reliably inferred from either the TTA or the CSV for the five vehicles.

Figure 11 presents the seven vehicles with a CSV of 17 kph. The vehicles include pickup trucks, passenger cars, utility vehicles, and a van. The CSV value of seventeen is near the middle of the CSV range (14 to 21 kph) and includes a diverse set of vehicles. A measure based on CSV would rank the rollover risk of these vehicles as the same.

Figure 12 presents the same seven vehicles ranked by TTA. Clearly, when the vehicles are ranked by TTA a different result is obtained. **Figure 13** presents the actual rollover experience for the seven vehicles with a CSV of 17 kph. Again, the actual rollover experience varies widely and cannot be reliably inferred from either CSV or TTA.

These figures illustrate the practical limitations of TTA or CSV as a basis for predicting real world rollover performance. First, neither reliably predicts the rollover experience of an individual vehicle. Second, often TTA and CSV produce different comparative rankings of rollover risk for the same vehicle.

RECOMMENDATIONS FOR A BALANCED APPROACH TO REDUCE ROLLOVER CRASHES AND INJURY:

Many crashes, including rollover crashes, can be prevented, and safety belt use can reduce the risk of injury in crashes that do occur. Safety belt use is particularly important in rollover crashes due to the severity and randomness of occupant kinematics during the rollover event. Based on the discussion presented above, it does not appear that either TTA or CSV provides any meaningful measure of rollover performance.

A major flaw in the consideration of these two static metrics is the unreliability of TTA and CSV as predictors of rollover in the real world. The measures are, at best, estimates of the energy levels required to tip over a vehicle--a one-quarter turn rollover. They are not theoretically or empirically related to the serious, injury producing, high energy rollovers that involve four or more quarter turns.

Additionally, the two measures predict different results for the same vehicles. More importantly, the actual rollover crash experience varies widely for vehicles with the same TTA or CSV. This variation in the crash experience makes both measures a poor choice as the key element for assessing vehicle rollover performance.

In stark contrast, the effectiveness of safety belts in reducing injuries is clear. Increasing safety belt use must be a major element of any effort focused on reducing injury in rollover crashes.

Risk taking behaviors like speed too fast for conditions and alcohol-impaired driving also contribute to rollover and nonrollover crashes. These facts need to be understood by consumers as well and should be included, along with the need for safety belt use, in a comprehensive consumer education program.

In order to most effectively address the problem of rollover crash related injury, the following five point action plan is recommended:

Action Plan:

1. Continue to communicate the benefits of safety belt use. Specific information should be included emphasizing the value of belt use in preventing rollover injury.

2. Continue to communicate the risks associated with alcohol and other drug impairment as well as the risks of other unsafe driving actions.

3. As soon as possible, initiate research to define the information consumers need to prevent rollover and nonrollover crashes and the associated injury.

4. Concurrently investigate the best methods for effectively communicating the information to consumers.

5. As a basis for action is established by the research findings, develop and implement a comprehensive consumer education program.

ACKNOWLEDGMENTS:

This paper is based on the information developed by the American Automobile Manufacturers Association (AAMA) working group on vehicle stability and contained in their October, 1994, submission to the NHTSA Docket 91-68. This author would especially like to acknowledge the work and contributions of Kent B. Joscelyn, Attorney at Law; David R. Houston, Ford Motor Company; Philip R. Horton, General Motors; and Carl C. McConnell, AAMA. A listing of references used by the working group follows.

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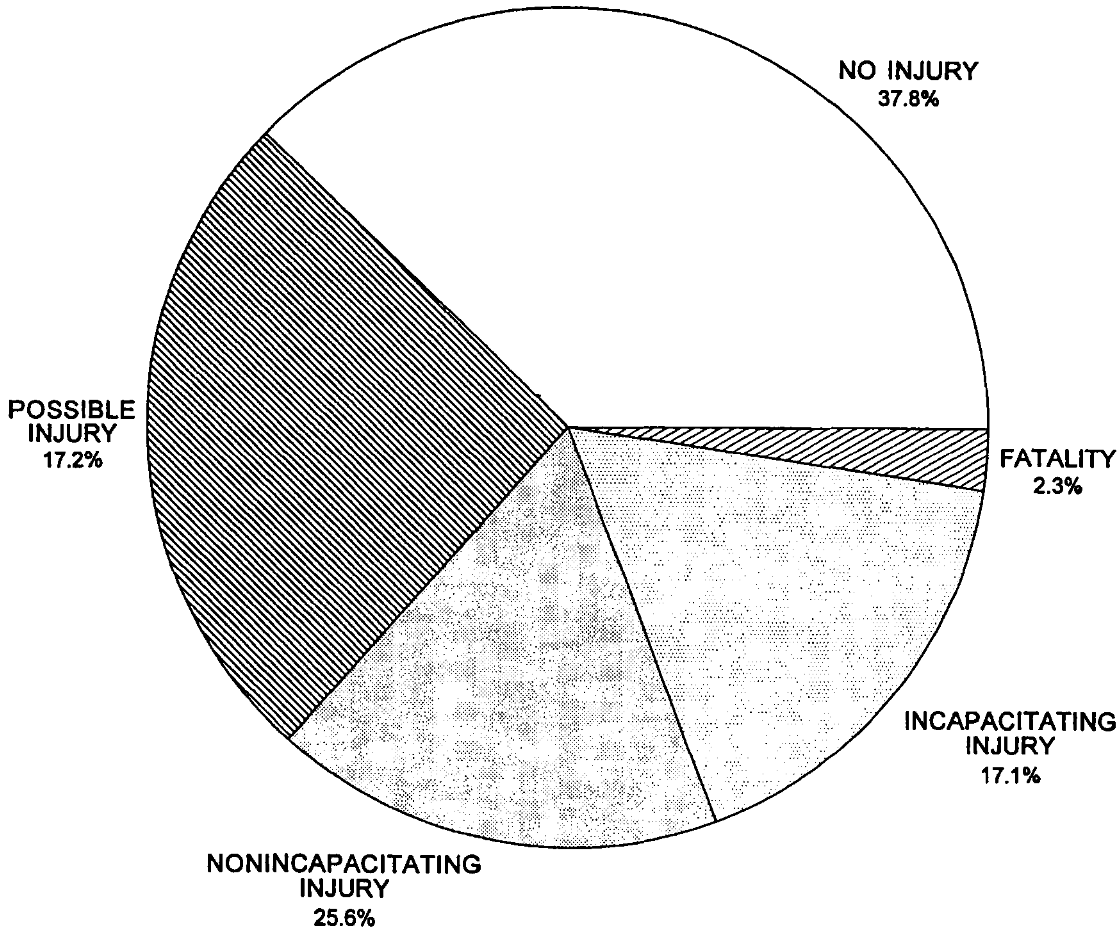
FIGURE 1

INJURIES OCCUR MORE FREQUENTLY IN ROLLOVER CRASHES THAN IN NONROLLOVER CRASHES

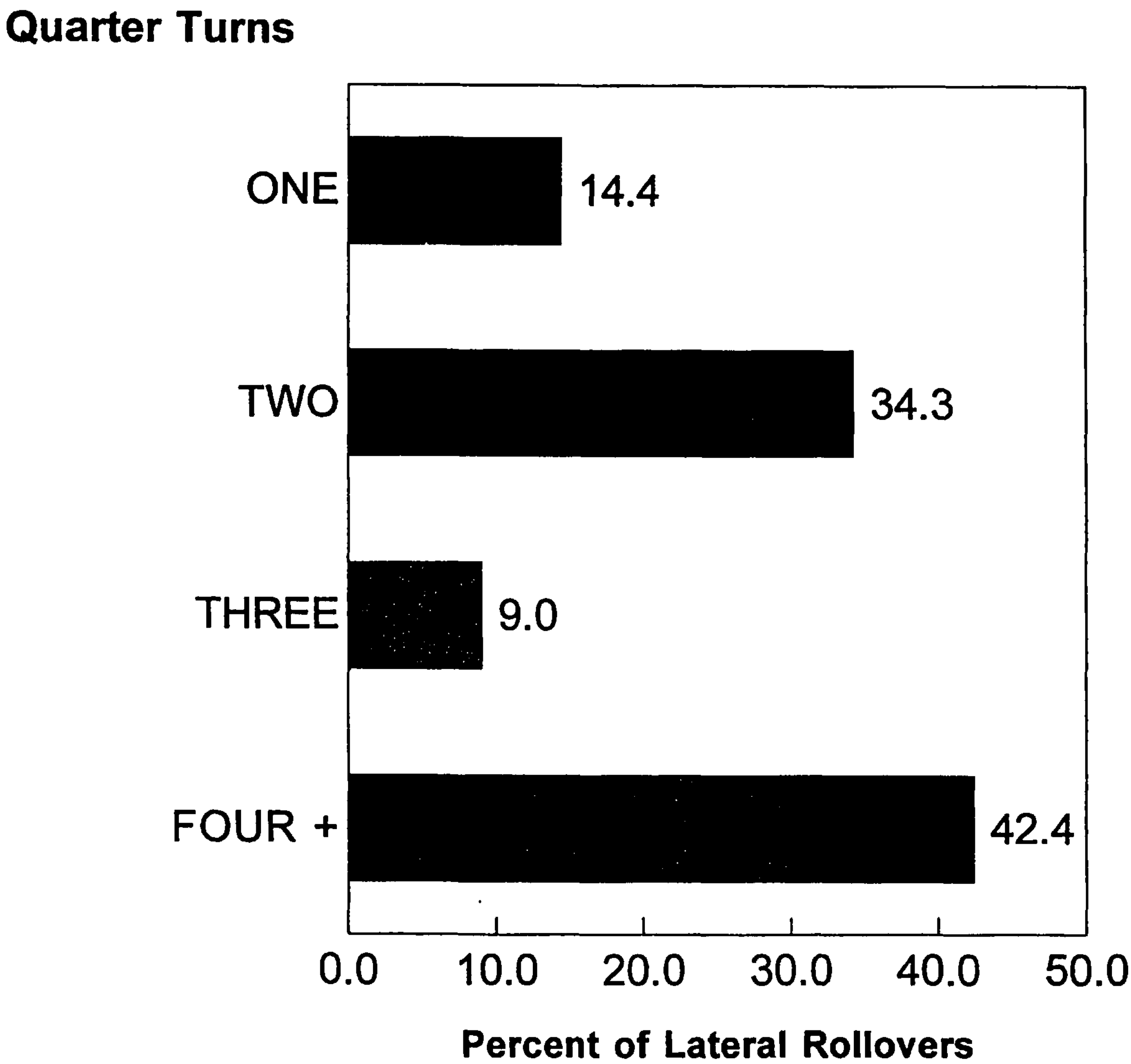


All Rollover Crash-Involved Vehicles = 100%

1988-1991 NASS GES weighted
Injury level measured as maximum injury in rollover crash vehicle.
Pedestrian, bicycle, and train crashes excluded.

FIGURE 2

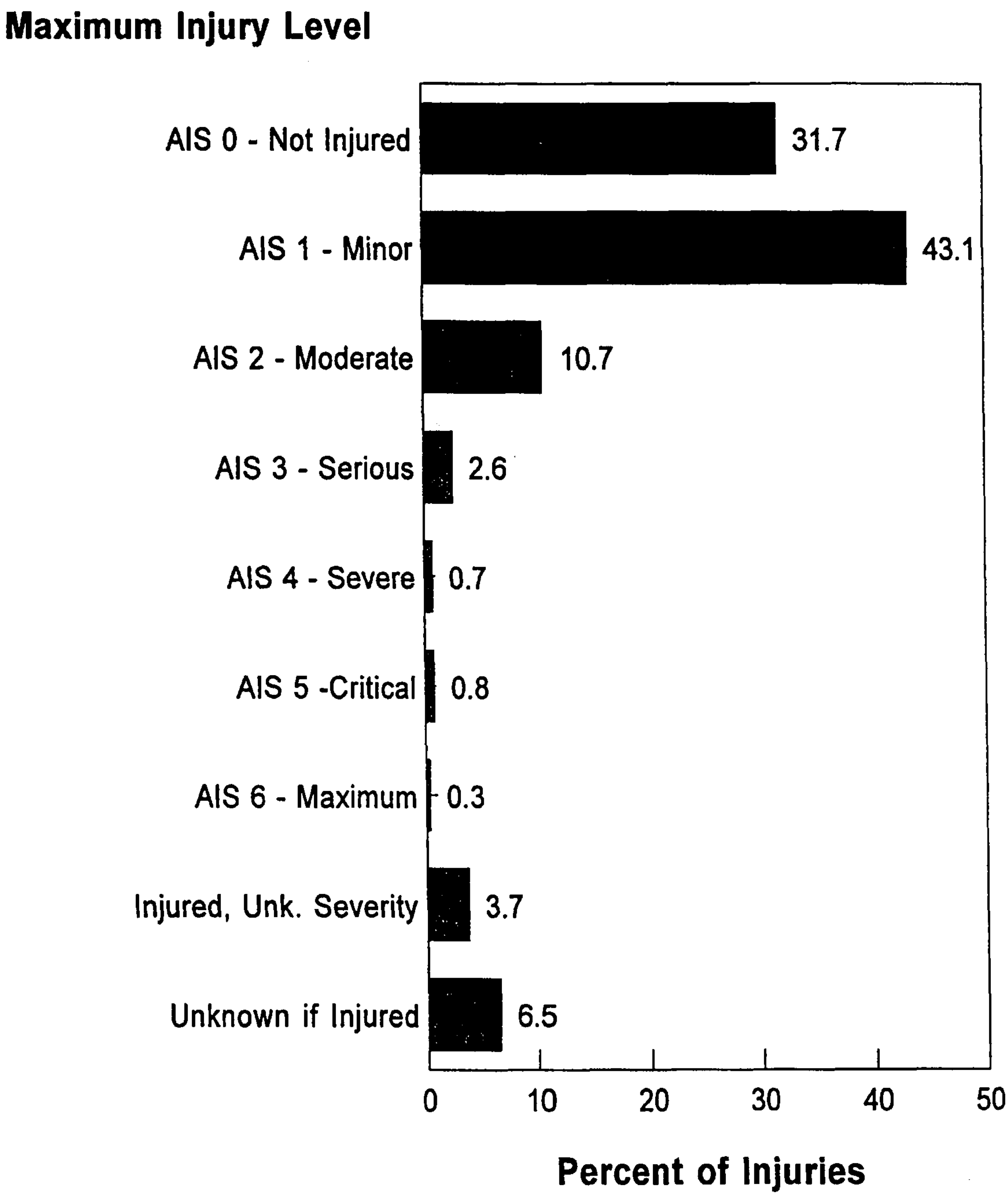
MOST FIRST EVENT SINGLE-VEHICLE LATERAL ROLLOVERS INVOLVE MORE THAN ONE QUARTER TURN



1988-1991 NASS CDS Nationally Weighted
Passenger car, pickup truck, van, and utility vehicle single-vehicle crashes.
Pedestrian, bicycle, and train crashes excluded.

FIGURE 3

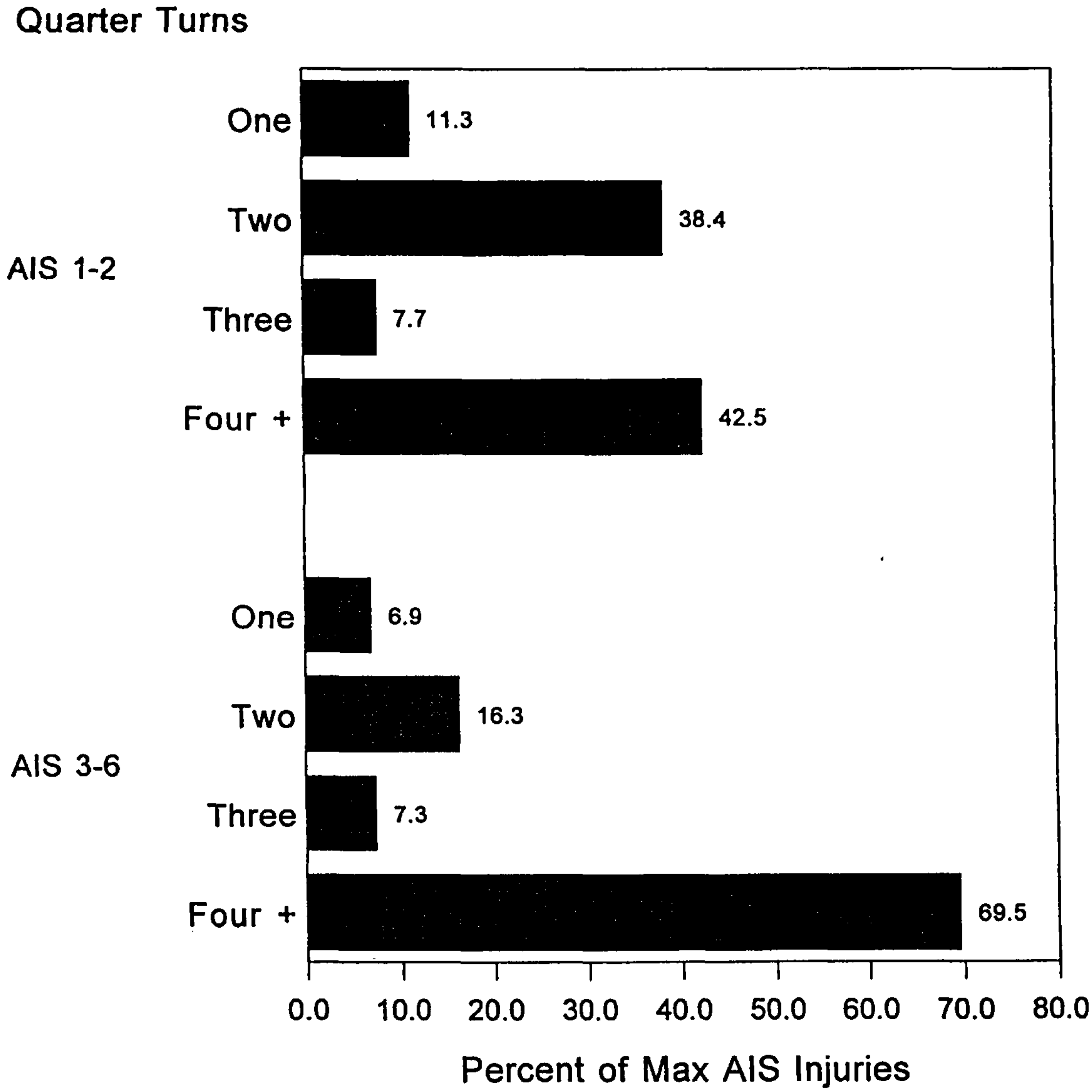
INJURY DISTRIBUTION IN SINGLE-VEHICLE CRASH LATERAL ROLLOVERS



1988-1991 NASS CDS Nationally Weighted
Passenger car, pickup truck, van, and utility vehicle single-vehicle crashes.
Pedestrian, bicycle, and train crashes excluded.

FIGURE 4

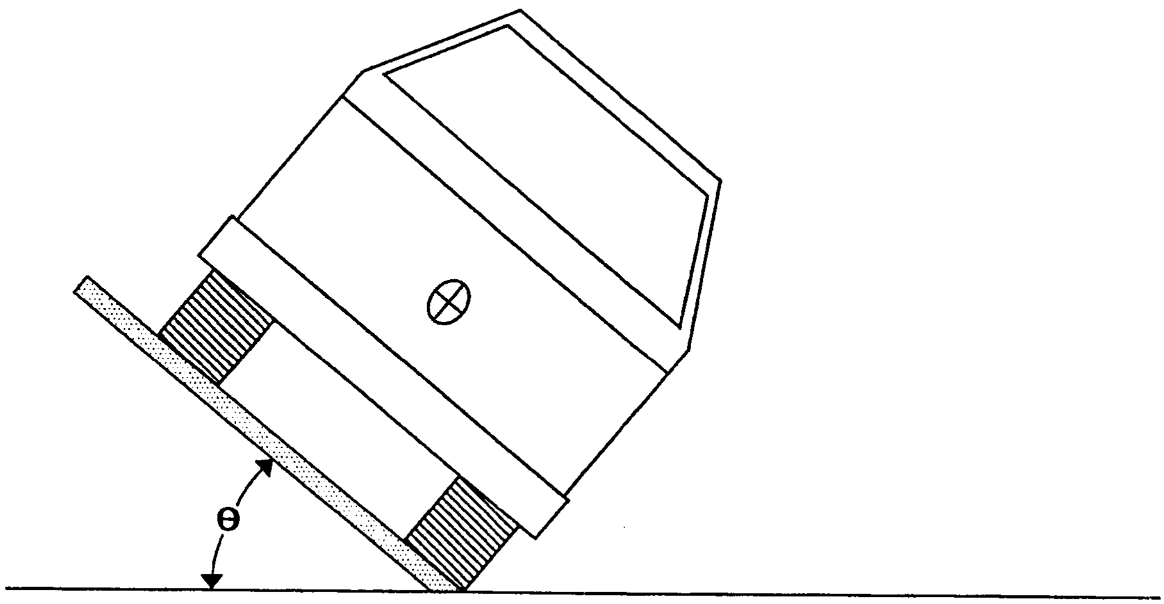
FORTY PERCENT OF MINOR INJURIES (AIS 1-2) AND SEVENTY PERCENT OF MAJOR INJURIES (AIS 3-6) OCCUR IN FOUR OR MORE QUARTER TURN ROLLOVERS



1988-1991 NASS CDS Nationally Weighted
Passenger car, pickup truck, van, and utility vehicle single-vehicle crashes.
Pedestrian, bicycle, and train crashes excluded.

FIGURE 5

TILT TABLE ANGLE



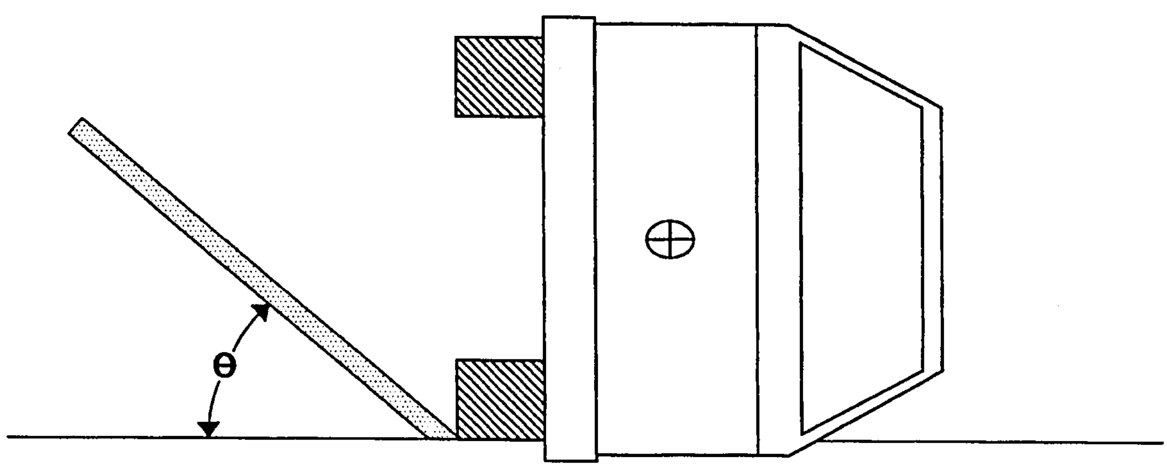


FIGURE 6

CRITICAL SLIDING VELOCITY

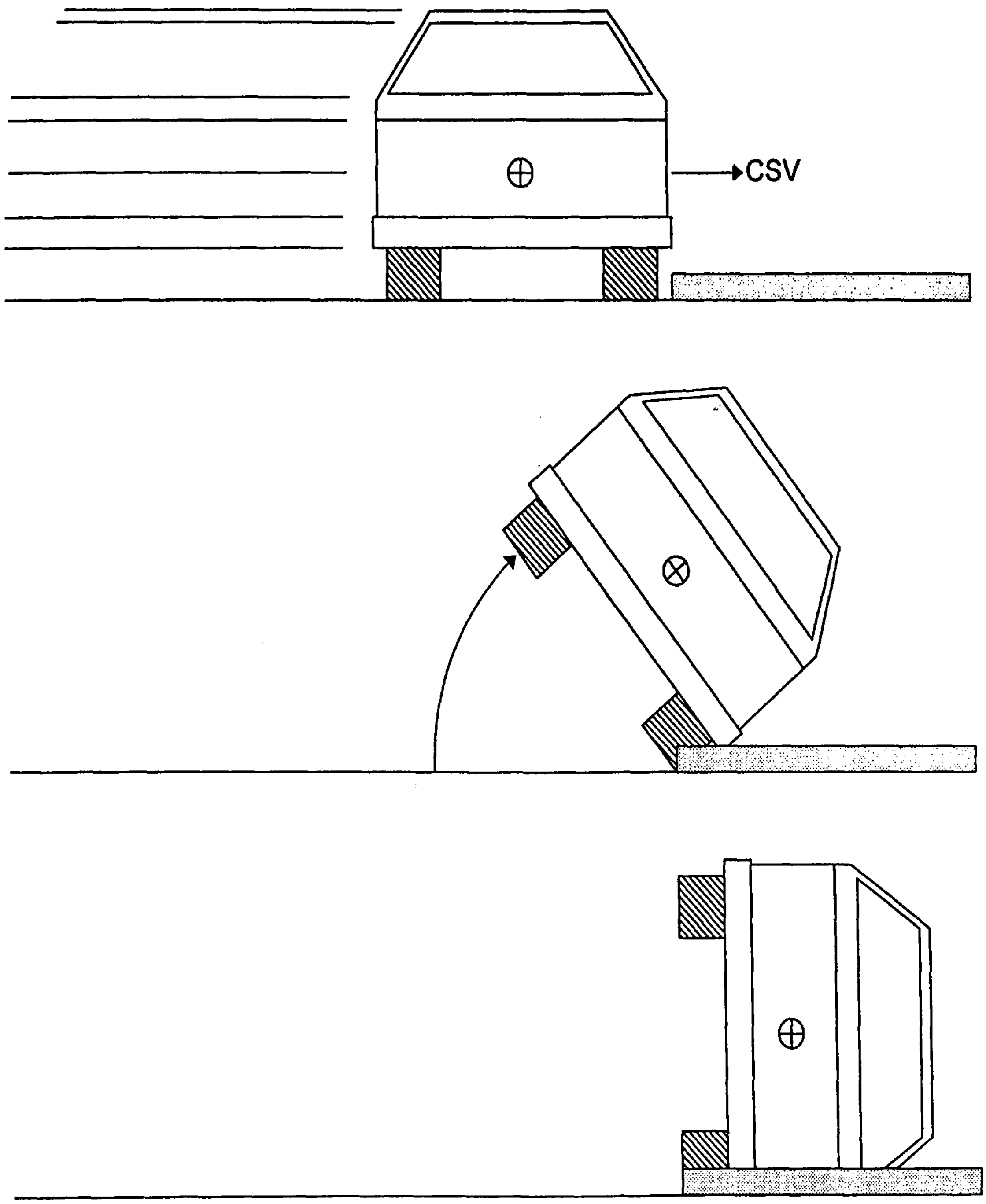
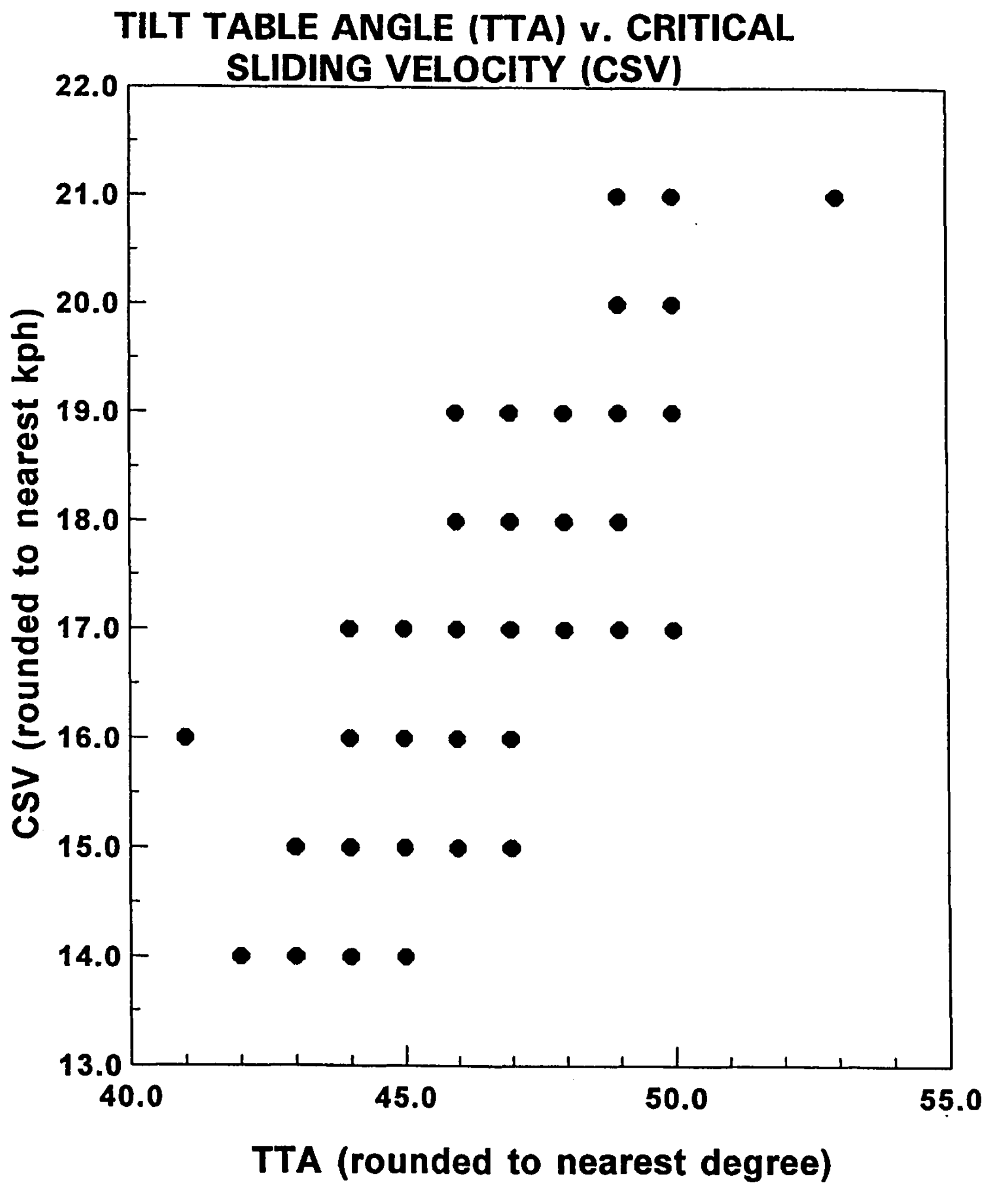


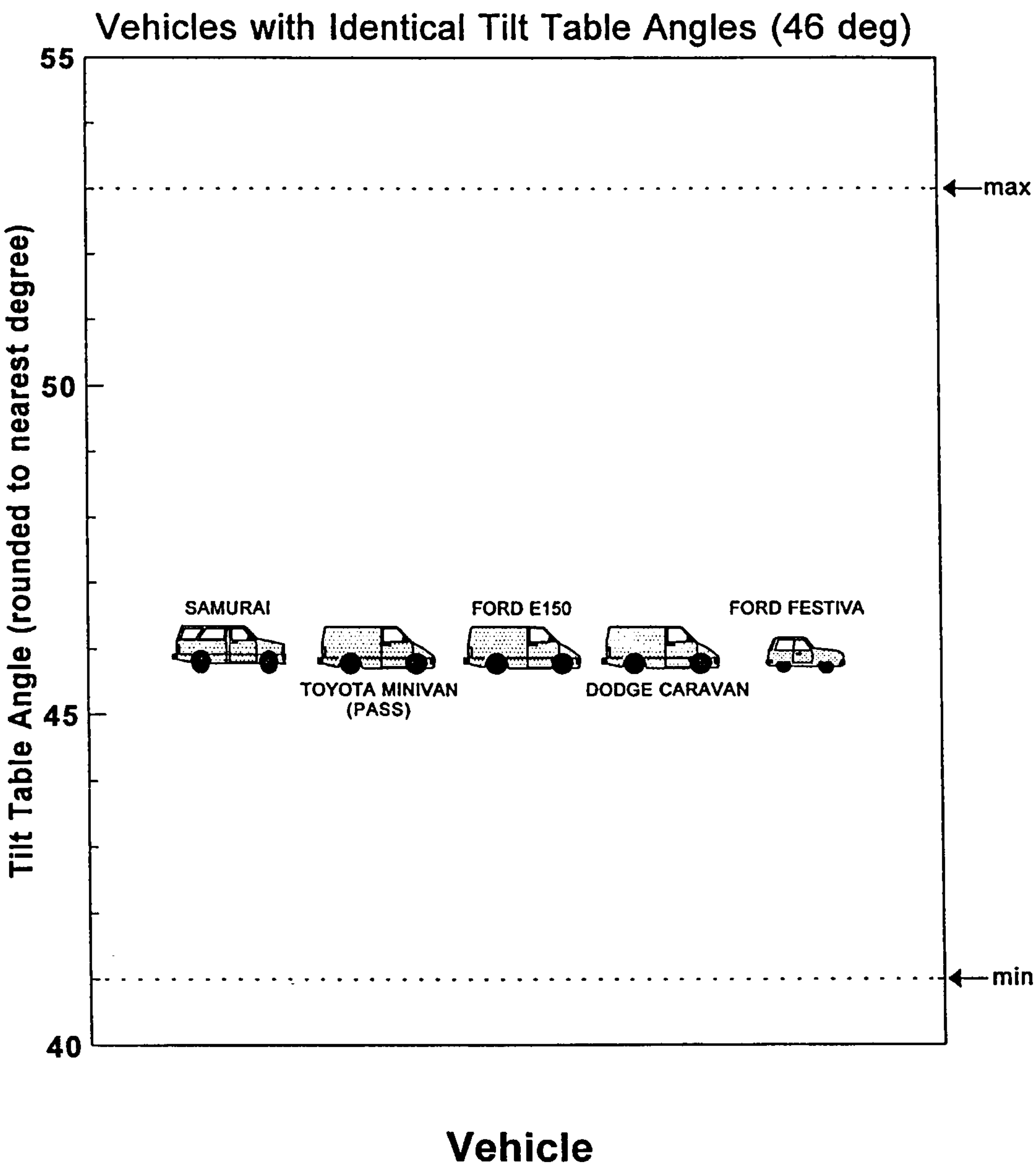
FIGURE 7



Source: Data for vehicles in NHTSA's November 1993 submission
to Docket 91-68 which include both TTR and CSV values.

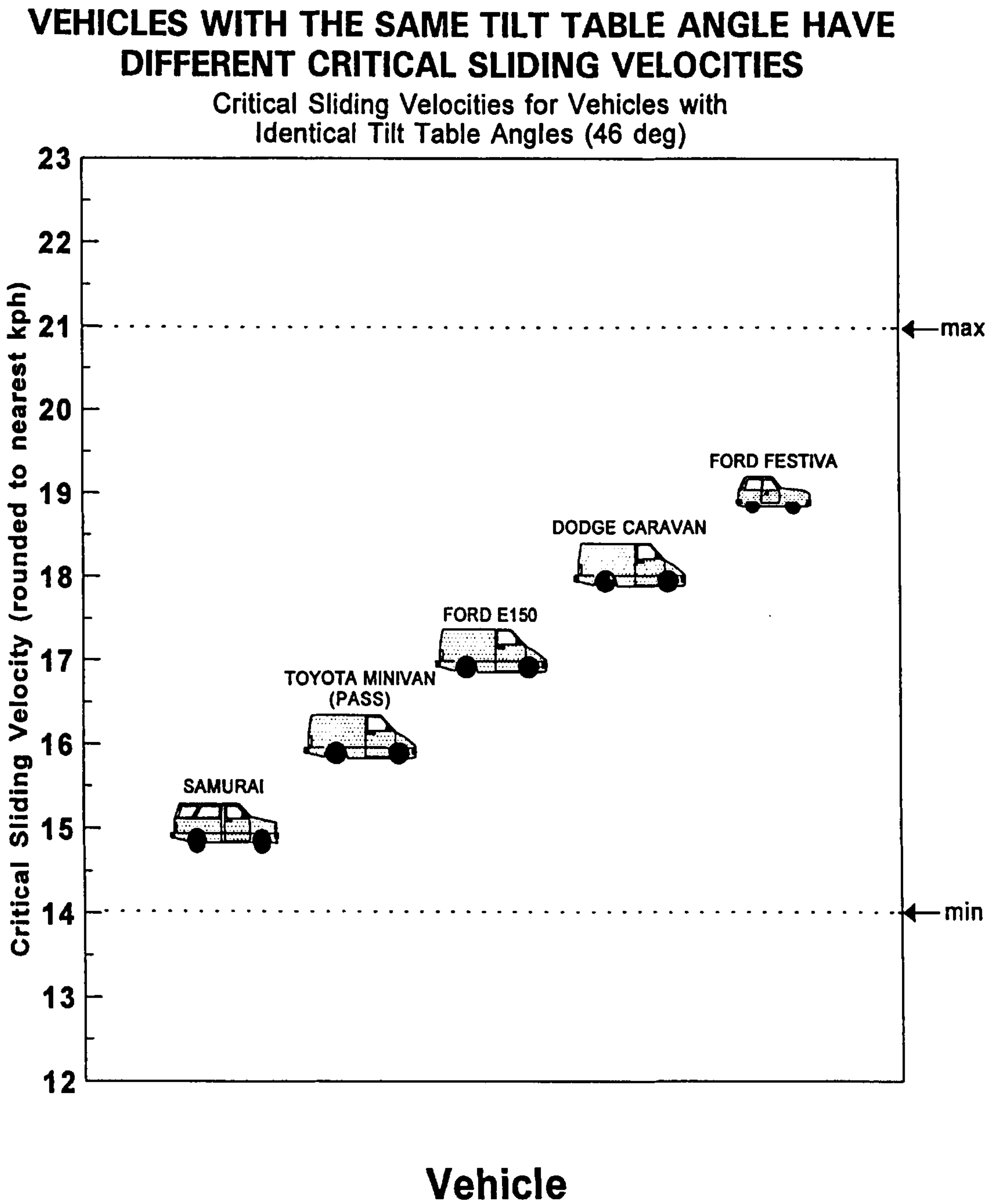
FIGURE 8

MANY TYPES OF VEHICLES HAVE THE SAME TILT TABLE ANGLE



Source: NHTSA November 1993 Submission to Docket 91-68

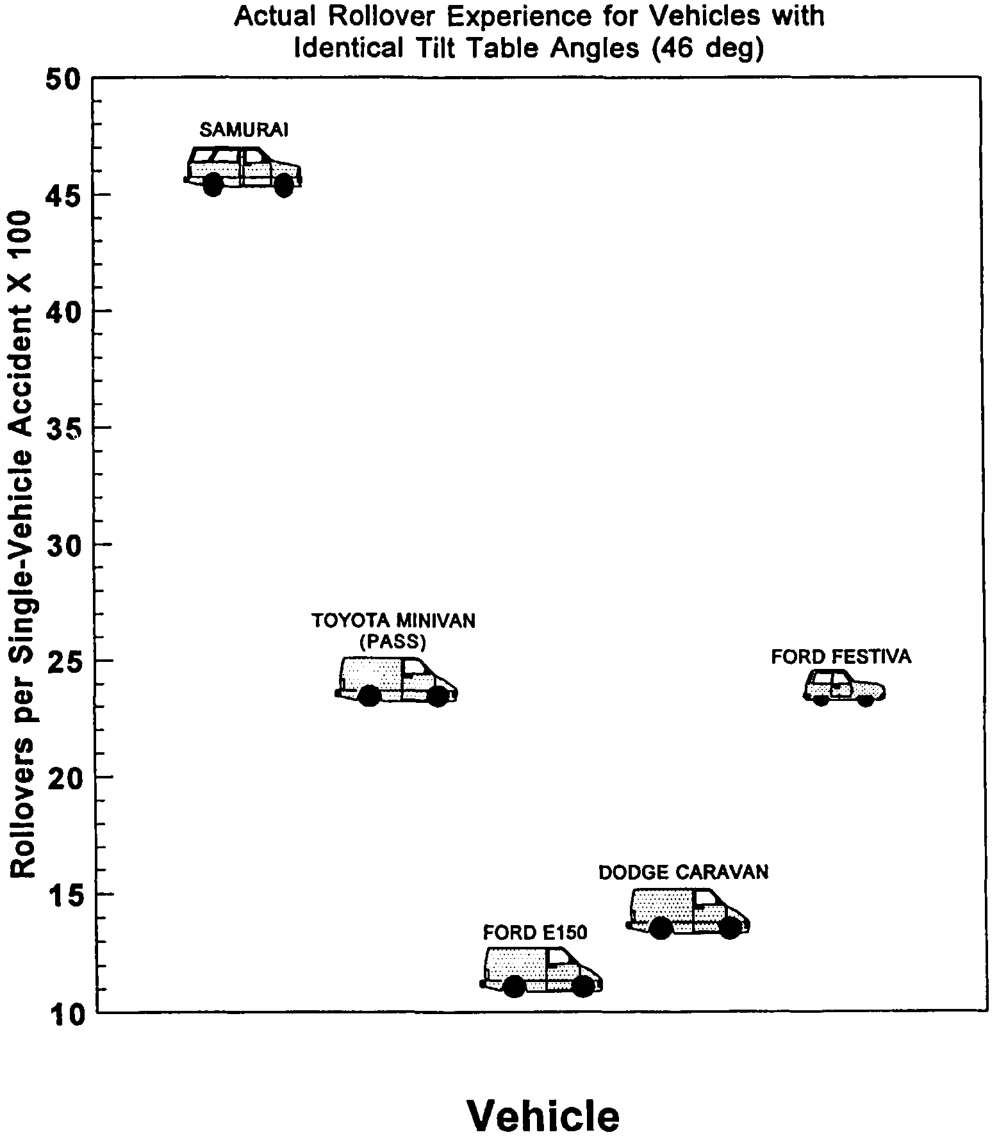
FIGURE 9



Source: NHTSA November 1993 Submission to Docket 91-68

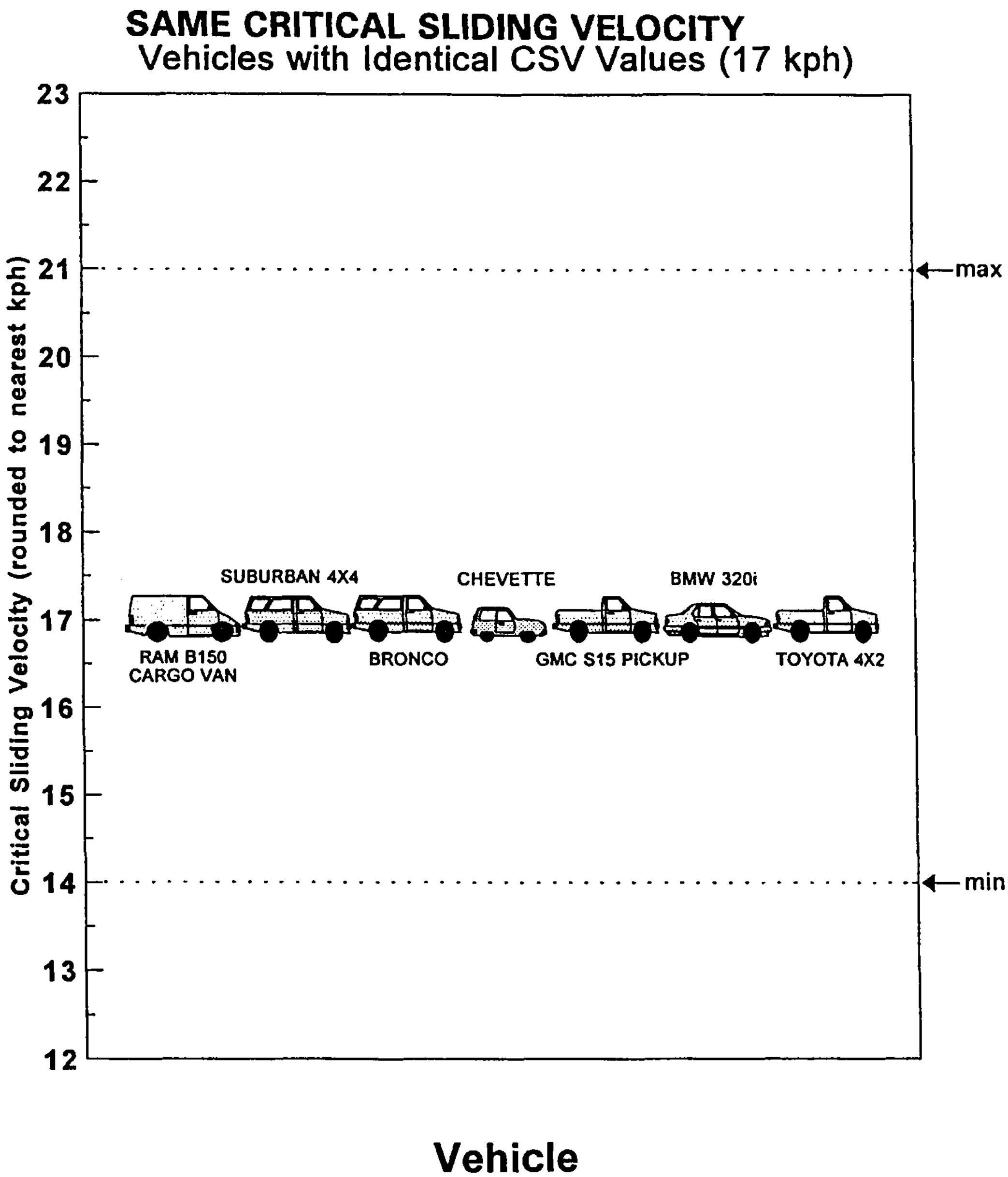
FIGURE 10

VEHICLES WITH THE SAME TILT TABLE ANGLE HAVE DIFFERENT ROLLOVER FREQUENCIES



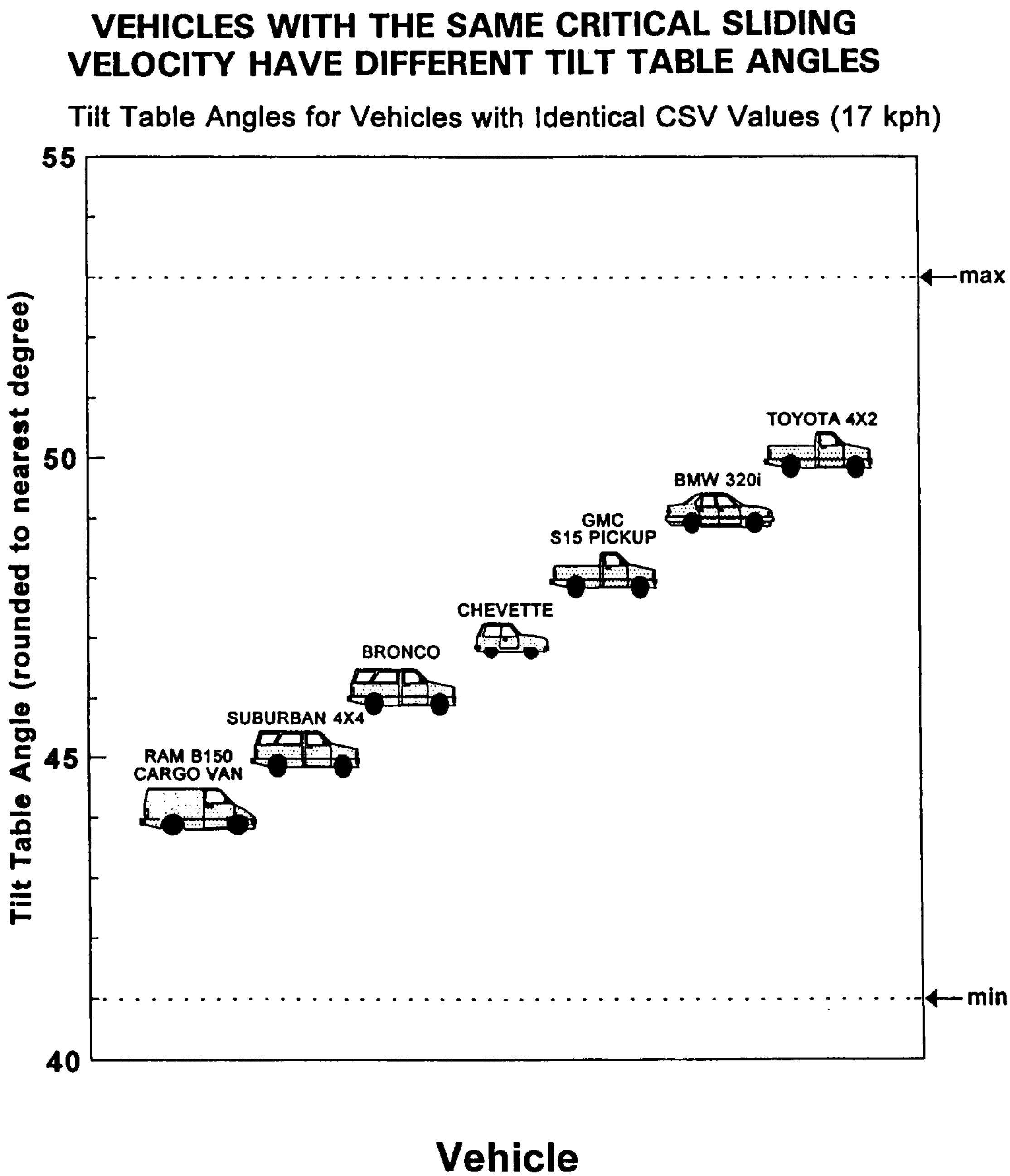
Source: NHTSA November 1993 Submission to Docket 91-68

FIGURE 11



Source: NHTSA November 1993 Submission to Docket 91-68

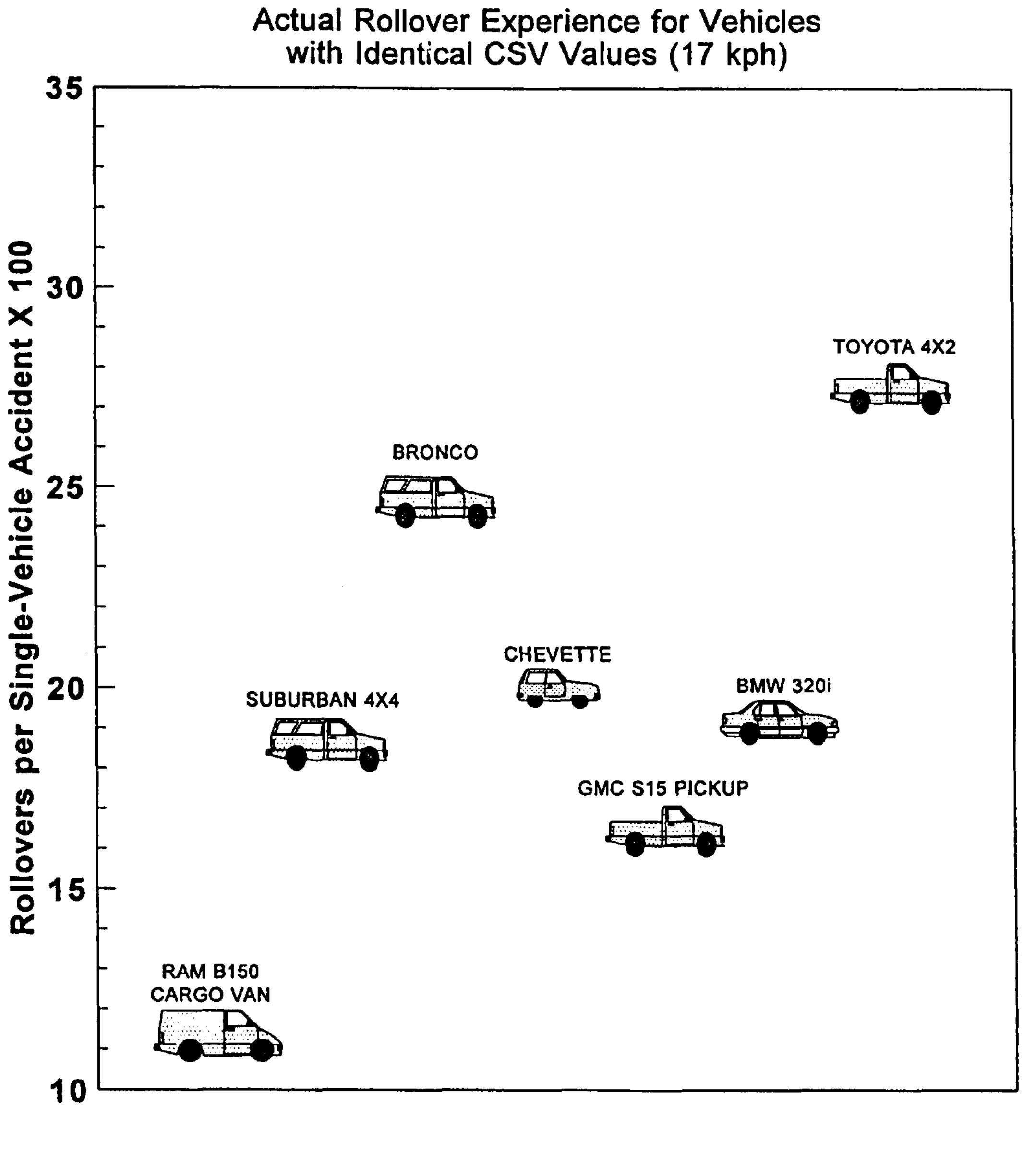
FIGURE 12



Source: NHTSA November 1993 Submission to Docket 91-68

FIGURE 13

VEHICLES WITH THE SAME CRITICAL SLIDING VELOCITY HAVE DIFFERENT ROLLOVER FREQUENCIES



Source: NHTSA November 1993 Submission to Docket 91-68

AN ANALYSIS OF BODY LOADS DURING ROLLOVER TESTS; ROOF CRUSH AND OCCUPANT PROTECTION

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Volkswagen AG
Germany
Paper Number 96-S5-O-09

ABSTRACT

The only standardized test procedure for vehicle rollover tests is described in FMVSS 208. An often raised objection to this test procedure stresses the fact that, for a given car, the number of revolutions as well as the final position of the test car after the crash vary from test to test. Consequently, there are no body deformations that resemble those of the previous test.

This paper presents an analysis of the contact forces. Such contact forces are determined from the accelerations of the car bodies during the rollover employing a numerical calculation method. It became obvious that the structural forces are reproducible, in spite of the differences in rollover kinematics. The body load strongly depends on the geometric design and the stiffness distribution of the car body. Another conclusion of the test series is that good results in roof crush tests are not necessarily an indicator for good rollover performance.

Biomechanical data are measured by Hybrid III dummies. In addition to measuring the forces and torque of the head and the neck, the surface pressure on the skull was determined by using a pressure sensitive foil. With the exception of extreme roof crushes, there is no correlation between roof crush and biomechanical loads. This result is in accordance with the analysis of real world crashes.

From the test series one may conclude that rollover crashes are a useful instrument for developing cars. However, such crashes only allow a qualitative investigation of crash performance, but they are not suitable for establishing any ratings of safety for different types of cars.

INTRODUCTION

Generally rollover crashes occur at high driving speeds. Most likely the vehicle leaves the road as a result of the accident. Many rollover initiating events are known; the most frequent one is this: the vehicle skids, trips on an obstacle, e.g. a curbstone, and finally rotates along its longitudinal axis. But rollovers can also be secondary crashes, like the reaction of a car to a side impact.

Consequently there are a great variety of car body damages and passenger injury levels. Statistical analysis of rollover accidents in Europe show, that the severity of injuries is comparatively small. A sample of 5000 accidents with injuries in the Hannover area in the years from 1985 to 1993 [1], involving 6200 vehicles, contains 99 rollovers, corresponding to 1.6%. All involved passengers who used their seat belts. It became clear, that only 28% of the involved persons had an AIS of 0; that means they were completely uninjured. Most persons had only light injuries, corresponding to AIS 1, and no case with AIS greater than 3 was recorded. Another analysis, based on data from the UK, describes a similar picture [2].

This positive situation is typical for European countries, where the percentage of belted passengers is greater than 90 %. In the US, where only approximately 50% of car passengers use their seat belts, the fatality rate is more serious. If accidents with non-belted occupants are considered, ejection of occupants can occur. This is an extremely dangerous event: in a study of Huelke [3] only 2% of the non ejected, but 47% of the ejected occupants suffered AIS of 5 and above. The risk of being ejected depends strongly on the use of seat belts: nearly all belted passengers remain in the passenger compartment, but about a quarter of the unrestrained occupants are ejected during a rollover [3,4].

BEHAVIOR OF THE CAR BODY

To eliminate the diversity of real world rollovers, the dynamic rollover test as described in FMVSS 208 is widely used to initiate a rollover along the longitudinal axis of the car. The vehicle to be tested is placed on a dolly, perpendicular to the direction of travel. The dolly moves with a speed of 30 mph against a hydraulic braking cylinder and comes to an abrupt stop. The car gets thrown off the dolly, hits the ground and starts to roll (Figure 1).

Up to the first rotation, the experiment is reproducible. But the total number of rolls, the final position of the car after the test and the damage of the car body differs from test to test.

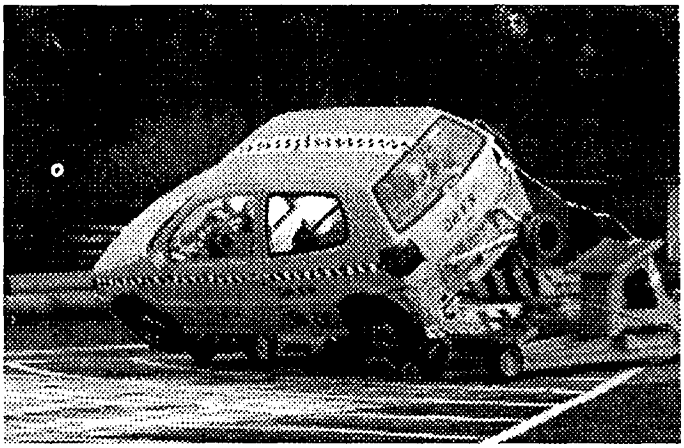

Figure 1. Rollover according to FMVSS 208.

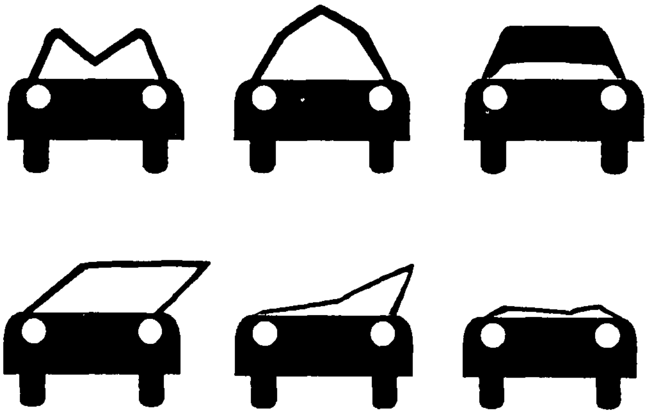

Figure 2. Typical Forms of Roof Deformation after Rollovers.

This is especially the case for roof deformations. For one model of car only the overall appearance of roof deformation is characteristic. Some typical types of roof deformations are shown in Figure 2.

Spread of Roof Deformation

As an example, Figure 3 illustrates the deformation of the roof structure of the Volkswagen Polo. The first curve represents the position of the front roof crossrail (i.e. the upper part of the windshield frame) of a new, undeformed vehicle. For two laboratory tests according to FMVSS 208 the course of the deformed crossrail is marked in the diagram. The two further curves are obtained by measuring cars after real world rollover accidents.

In all crashes the roof deformation is stable, i.e. no collapse of the upper body structure occurs. But the maximum displacement of the roof in the vertical direction, in the case of the Polo the upper end of the A-pillar, ranges from 20 mm up to 200 mm.

Because of this huge spread of roof deformations, the use of absolute values of displacements as a measure for crash-worthiness is not advisable. Especially a comparison of two tests with good results isn't possible. The only reliable general rule is that a complete collapse of the roof structure can serve as an indicator for poor crash performance.

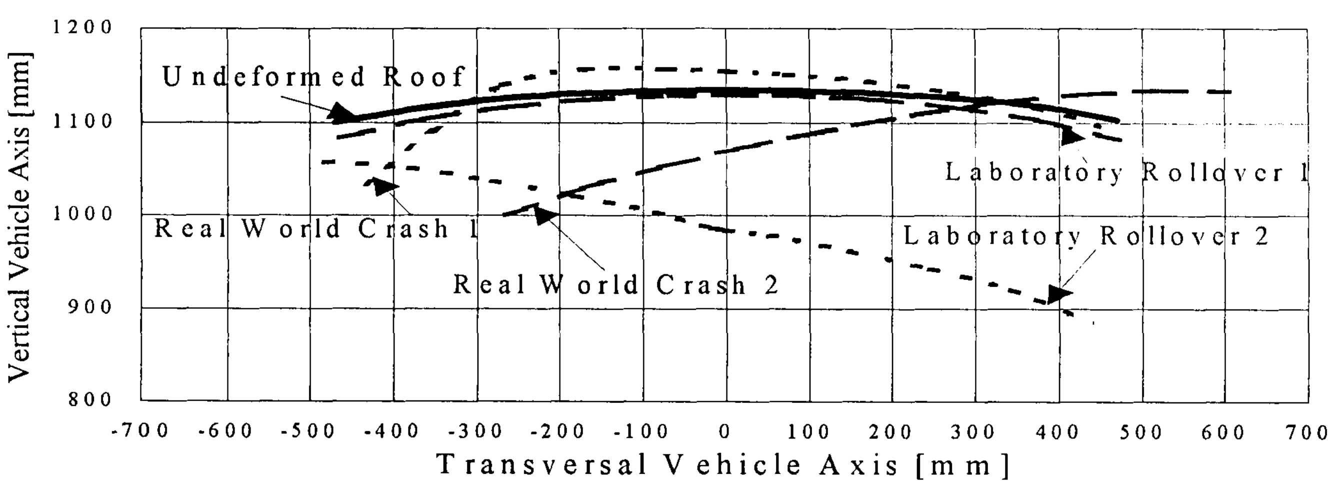

Figure 3. Deformation of the Front Roof Crossrail after Laboratory Rollovers and after Real World Rollover Crashes.

Determination of Body Loads

For the construction of the car body, the knowledge of the structural forces is essential. Contrary to frontal crashes, no direct measurement of contact forces during a rollover is feasible.

If the accelerations of the cars center of gravity (CG) could be measured, the resulting force on the car body would be known too. Unfortunately, for most cars this point is situated in the middle of the passenger compartment at a position where no sensor can be installed. However, an acceleration sensor, which isn't placed exactly in the vehicles CG, always measures a superposition of translatorial and rotational accelerations. This makes it necessary to measure six linear independent accelerations at three different locations of the car and use them for calculating the three unknown translatorial and the three angular accelerations.

To interpret the results of the test and to compare the calculated rollover kinematics with the films of the rollover, the values of accelerations have to be based on the inertial coordinate system. The data, obtained by the measurement during the rollover test is given in the rotating vehicle coordinates.

To convert these values to the inertial coordinate system, the position and orientation in space of the vehicle has to be known. Before the test the vehicle stands on the dolly at an angle of 23°. Starting with the initial position, the position and orientation of the vehicle can be calculated incrementally. Now the acceleration of the vehicle's center of gravity is determined and the external force, based on the inertial coordinate system, can be calculated.

Volkswagen uses this method to support the development of a new car. A more detailed description of the evaluation method is given in [5].

Spread of Roof Force

With respect to the very large spread of the roof deformations, the variation of the contact forces obtained by rollover tests is of special interest. In figure 4 the maximum loads during the rollover tests of figure 3 are sketched.

In all cases the peak forces appear at the same stage of the turning over; in our example when the car touches the ground with the edge of the A-pillar and the roof crossrail. In spite of the different deformations, the peak values as well as the directions of the force are the same. Only in the case of soft rolls do smaller contact forces occur. Otherwise the force level is reproducible, here approximately 25 kN. The reason

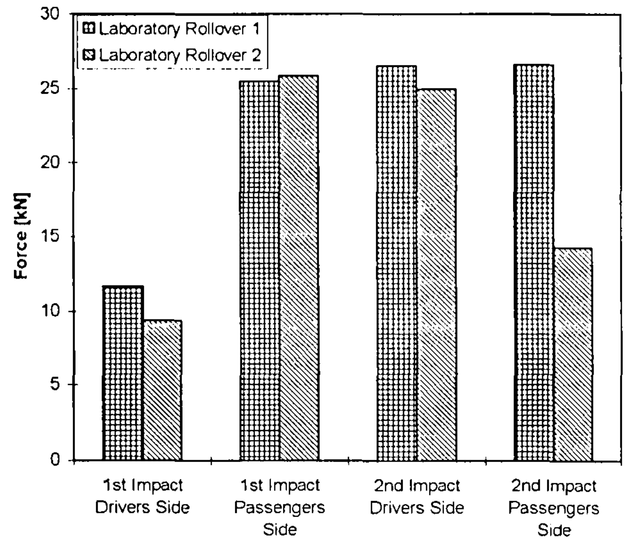

Figure 4. Maximum Roof Loads.

for the different roof deformations isn't found in the value of the force, but the exposure time.

This value of the contact force is determined by the geometric design of the car body and by the roof strength. A quasistatic roof crush was made for comparison, following the pattern of FMVSS 216. Deviating from the common crush test, the direction of the crushing force was selected equal to the one delivered by evaluation of the dynamic rollover test. This quasistatic test delivered a crush force of 21 kN. The difference of 20% between the force measured during the rollover tests and the quasistatic tests can be explained by the different contact zones and deformation speeds.

Roof Strength and Crush

Normally the geometric dimensions of the car body like vehicle height and width can't be altered at the time when the first prototypes are available. In consequence, only the stiffness of the body can be influenced.

The first suggestion is to brace the roof structure in order to lessen the roof deformation. If that strategy is feasible, development on the basis of quasistatic crush tests would be possible. To verify this idea, the roof structure of an existing car body type was both - reinforced and weakened. Four variations of the car body were selected for a detailed investigation.

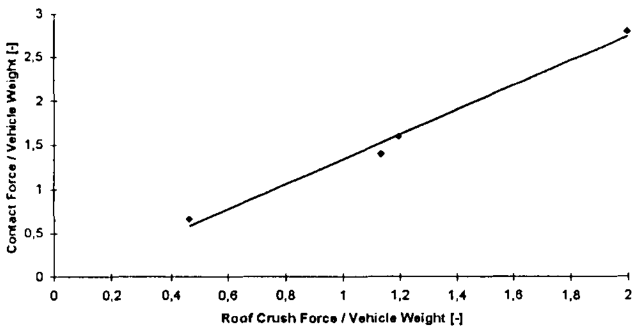

Figure 5. Influence of Roof Strength on the Contact Force during Rollover.

Figure 5 displays the interdependency between the quasistatic crush force and the maximum contact force during dynamic rollover tests. The values of the forces are normalized with the vehicle weight. The results of the two different test methods show a good correlation.

The next step is to investigate the influence of the crush force on the roof deformations. Due to the large spread of the deformations, average values of multiple deformation measurements are used during the further discussion.

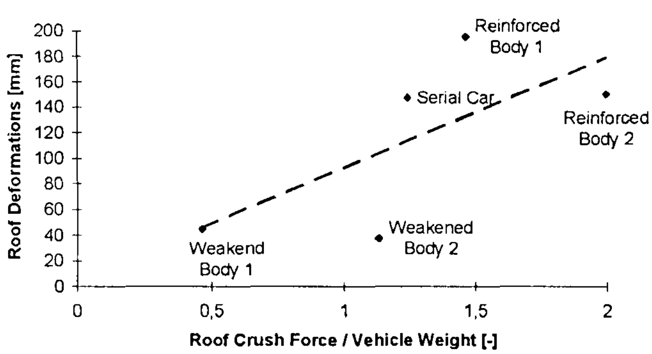

Figure 6. Influence of the Roof Strength on the Roof Deformations.

Figure 6 illustrates the results of the different body variations. It became clear, that an increase of the roof strength doesn't imply smaller deformations. It's quite interesting that the smallest deformations are obtained by a body structure whose crush force is smaller than the vehicle weight. Due to favorable stiffness distribution, this particular structure supports the flow of the turning over. At the end of all rollover tests with that structure, the car was situated on its wheels. But in spite of the good results in the laboratory tests, such a structure isn't desirable. If, due to a real world crash, the car should drop on its roof, a complete collapse of the passenger compartment occurs.

On the other hand, the two investigated body reinforcements don't have any positive influence on the roof deformation. Quite contrary, one of the two reinforcements implies a higher roof crush than that of the original car.

The test series demonstrates, that the roof deformations are determined by the kinematics of the rollover. The associated contact forces are simply defined by the roof strength. This connection shows that a reinforcement of the roof only raises the value of the contact forces, but has no influence on the body deformations. To reduce the deformations, the stiffness distribution of the body has to be changed in a way to support the rolling.

BEHAVIOR OF THE OCCUPANTS

The question of whether there is a relationship between roof crush and severity of the occupants' injuries is discussed controversly. The following discussion is restricted to belted occupants. The movement of non belted occupants during a rollover, especially if they are ejected out of the passenger compartment, is stochastic to such an extent that a systematic laboratory investigation is impossible.

Biomechanic Measurements at Laboratory Tests

Rollover tests at Volkswagen are accomplished with instrumented Hybrid III Dummies. These dummies are suitable for measurement of physical parameters like accelerations of the head or force and torque at the neck.

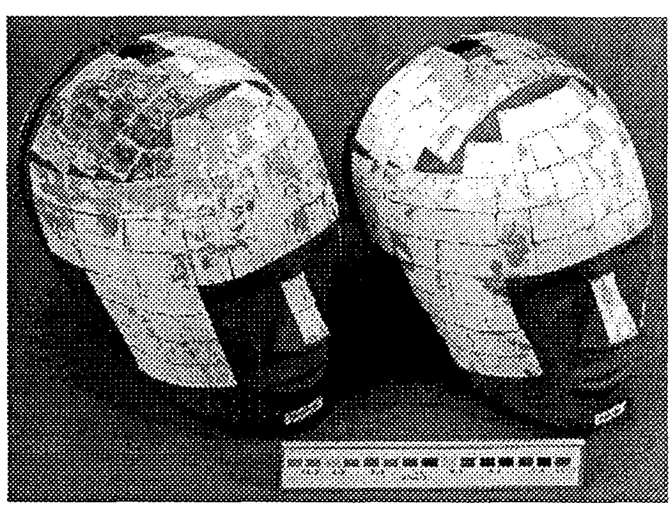

Figure 7. Surface Pressure on the Dummy Heads.

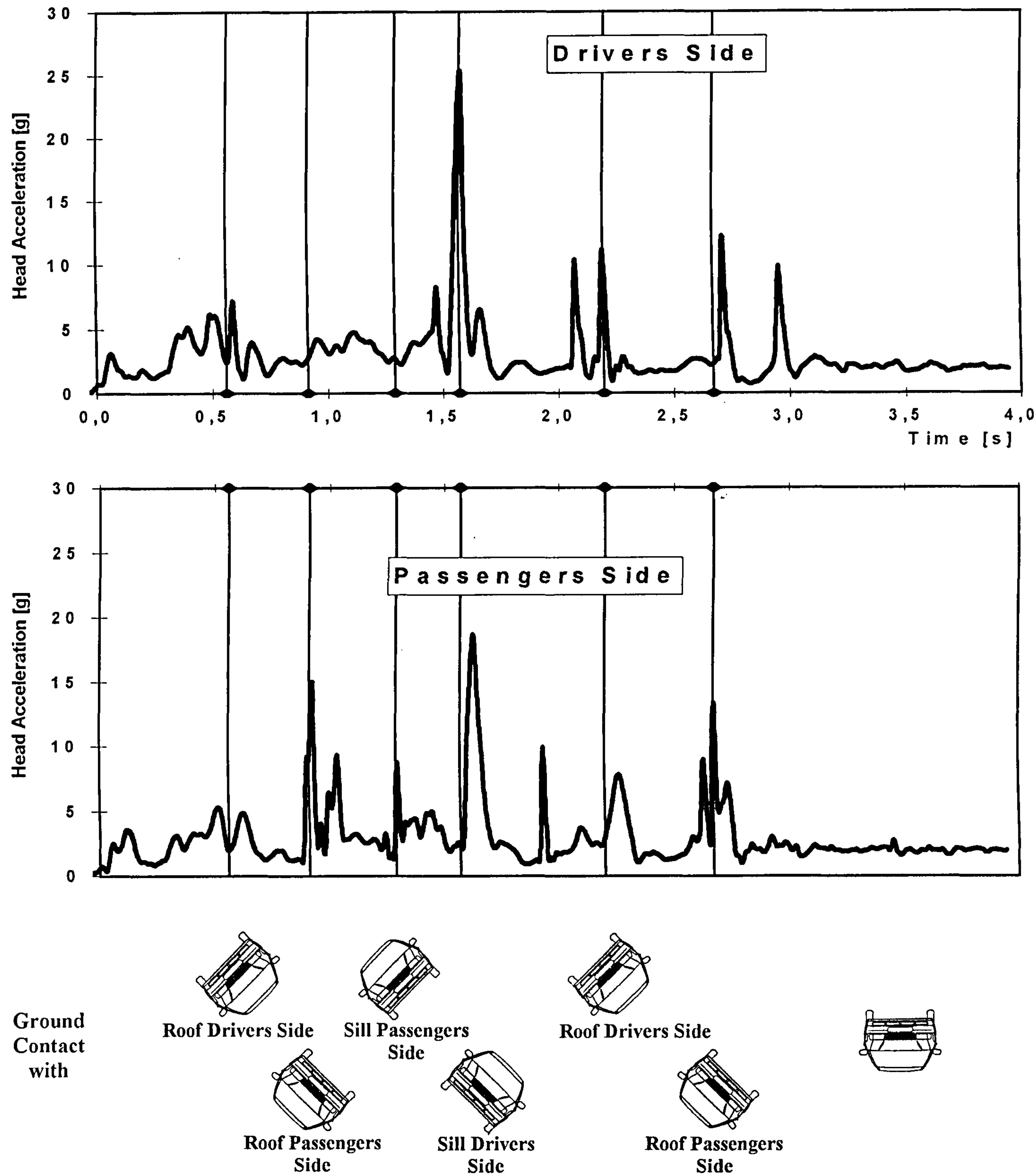

Figure 8. Time History of the Head Accelerations.

The dummy only delivers information on the resulting force which acts on the head. The influence of sharp edges or other perilous details of the car's interior can't be grasped by such values. Therefore the dummy head was equipped with a pressure sensitive foil, which ascertains the pressure distribution on the skull, Figure 7.

A difficulty to be dealt with when interpreting the measured physical values is that no biomechanical derived limits exist. The established criteria like HIC as a criteria for the head accelerations are derived either for frontal or for side crashes. But in the case of rollover accidents not only is the direction of the load different, but also the exposure time is much larger: in case of frontal crashes the whole impact takes less than 100 ms while a rollover lasts several seconds. Therefore, it is not possible to use the injury criteria, developed for short time impacts, to predict the injury level after a real world rollover accident with multiple, separated impacts. Nevertheless, the data gained from the dummies help reaching an understanding whether a particular change of the body structure or a padding has any effect with respect to occupant safety.

Injury Mechanisms

Rollover tests are very spectacular experiments. Contrary to other crash types, they take a long time and an observer can get the impression that they are extremely harmful for the occupants. This first impression is confirmed if the car is heavily damaged, especially in the case of large roof deformations.

This intuitive valuation is not confirmed by dummy measurements. Exemplary the 'Laboratory Rollover 2' of figure 3 will be discussed in more detail. In the course of this test the car executes one and a half turns. During the last phase of the crash it slides on its roof until it comes to a stop after approximately 10 m.

A film camera, mounted in the passenger compartment, documented the movement of the heads of the dummies. Due to the centrifugal force, the heads move outwards and lean at the door frames. When the roof hits the ground the first time on the driver's side an impact was registered not only by the driver's dummy but by the passenger's dummy also, Figure 8. This is a consequence of the direct contact of the head with the car body.

It is of special interest, that the dummies get their hardest impact the moment the car hits the ground with its sill. Due to the greater stiffness of the floor assembly in comparison to the roof, the contact forces respectively the body accelerations, reach their maximum value. This effect is the reason why a reinforcement of the roof or a rollcage can have a negative effect on the dummy data, [6,7].

During the last phase of the crash, when the car slides over the ground and comes to a stop, there are some further impacts to the heads, the severity of which didn't reach critical values.

Also the other biomechanic measurements during the laboratory crashes suggest no danger for the occupants. In accordance with the laboratory tests, the occupants of the two crashed vehicles, quoted in figure 3, are completely uninjured or endured only light injuries.

CONCLUSIONS

On the condition that a complete collapse of the car body does not occur, the main points of the investigation can be summed up as follows:

- The maximum value of the contact force during dynamic rollover tests shows a good correlation with the crush force, determined by quasistatic tests. But there exists no correspondence between the crush force and the roof deformations.

- Compared to the spread, the variation of the experimental results is so small, that a rating to compare different makes of cars doesn't seem to make sense. But the tests are a valuable tool to asses construction details in respect to their influence on the occupant protection.

- Biomechanical measurements vary only slightly. In particular, no significant influence of the body deformation can be found.

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ROLLOVER CRASH STUDY - VEHICLE DESIGN AND OCCUPANT INJURIES

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Paper Number 96-S5-O-10

ABSTRACT

The aim of the project was to investigate the relationship between vehicle design and the nature and severity of occupant injuries. A critical appraisal of the literature was made and a detailed investigation was undertaken of 43 rollover crashes with injury severity ranging from none to fatal. From the literature, the weight of evidence indicated a relationship between roof crush and occupant injury and a particular association between roof crush and spinal cord injury.

The crash study generally supported these conclusions but also identified factors which could lead to spinal cord injury when there was small roof intrusion. Another injury mechanism was found to be partial ejection of belted occupants leading to crushing head injury. Recommendations for vehicle design improvements include side window integrity, roof framing strength, geometric design of door/ roof framing, interior padding, restraint design and door integrity.

BACKGROUND

This paper is based on a report entitled Rollover crash study - Vehicle design and occupant injury (1).

Rollover crashes are a common cause of occupant injury, especially on non-urban roads. Their importance increases with injury severity: they constitute 19% of occupant fatalities in Australia. This percentage rises to 44% in rural Western Australia and 54% in rural Northern Territory. Rollover crashes are an important source of very severe trauma such as spinal cord injury (2).

Vehicle types have been identified with risk of rollover, various utility type vehicles having rates much higher than passenger cars (3, 4). This variation appears to be due mainly to the relation of the centre of gravity to track as measured by the Tilt Table Angle or Tilt Table Ratio (5, 6). Wheelbase is thought to have a smaller influence: shorter vehicles having a greater propensity for entering a potential rollover situation.

Rollover is usually initiated by a form of tripping mechanism such as engagement of the wheels with a kerb or small obstacle, or sideslip through soil or sod or friction with the road surface after a yaw manoeuvre (7, 8). This mechanism requires a lateral velocity and induces substantial accelerations in the occupants. These lead to collisions by the occupants with the vehicle interior and/or partial or complete ejection (8, 9).

All body regions are implicated in rollover casualties but the more severe injuries involve the head, neck, chest and abdomen (see Table 1), (10). Injuries are much more frequent in ejectees (11). In terms of "Harm", ground and vehicle exterior contact account for 35%. Significant fractions are associated with head/brain and neck injury from contacts with the roof and upper interior structure. Harm also shows a pronounced effect of vehicle speed, rotation in pitch and number of quarter turns (7). The amount of rotation is usually not great - in 85% of rollovers there were less than two turns (12); and 80% of urban rollovers did not exceed one turn and only 6% exceeded two turns (10).

Table 1.
Injury site and severity in urban rollovers.

MAIS	1	2	3	4	5	6	all	% with MAIS 3 or more
Body region								
head	95	37	12	1	1	4	153	11.8
neck	51	8	3	0	1	3	66	10.6
chest	41	10	10	6	6	2	75	32.0
abdomen	14	2	1	3	7	0	27	40.7
u/limbs	95	28	6	0	0	0	129	4.7
l/limbs	78	11	11	0	0	0	100	11.0
	374	96	43	10	15	9	550	14.0

(Source: Mackay et al 1991 (10). The AIS scale is 1985. MAIS is the highest AIS of any body region.)

There has been no unanimity about the role of roof crush in injury production. There are associations between crush and injury (10, 11, 13) but investigators have tended to ascribe the association to crash "severity" as the variable responsible for both crush and injury.

A particularly severe and disabling injury is to the spinal cord (SCI). Rollovers have been shown to be three times as productive of SCI as road crashes generally (see Table 2), (14, 15, 16). In an Australian study (17), of 44 vehicle occupants with SCI, 38 were from rollovers. In another study series of SCIs, 14 of 41 occupant casualties were from rollovers. In six restrained occupants, the injury was related to a crushed roof and three more to "crushed roof support" adjacent to the SCI occupant (18). This series, in which the data were unusually complete, had high belt-wearing and low ejection rates. A feature was roof crush localised to one seating position, that of the SCI casualty, the other occupants having minor or no injuries. This local effect was noted in NASS data (19).

Table 2.
Rollovers and spinal cord injuries

Injury counts per 1000 exposures						
	cervical			thoraco-lumbar		
AIS	1	2	3+	1	2	3+
Frontal	62	1.7	1.4	26	1.7	1.0
Side	63	1.6	1.8	41	1.8	0.38
Rear	270	2.0	0.65	93	1.3	0.34
Rollover	82	8.9	7.3	100	13	2.9

(Source : Yoganandan et al (15). Ejectees excluded.)

Ejection is a notable feature of rollovers - understandable in the light of the body motions revealed in rollover tests (8). In seven simulations described by Oberfegel (20) the occupant slipped out of the shoulder belt. A frequent ejection path is through glazed areas. In simulations, occupants have exerted loads on windows sufficient to break the glass and even impose loads on the door in excess of those specified in the design rules. At least one window disintegrates in 65 % of rollovers.

At least three inter-related factors operate to influence injury severity. Travelling speed at the time of the event is a "powerful predictor of severity" (21), but speed estimates tend to be unreliable. The rollover itself and ejection are the other two predictors. Terhune (22) found that in single vehicle crashes, rollovers tend to occur at higher speeds than non-rollovers, but the rollover event increases the risk of driver serious injury by 1% to 50%. There is no independent measure of "crash severity" applicable to rollovers as there is, for instance, in frontal collisions.

Experimental rollovers have been described by Orlowski (38), Habberstad (9), Bahling (23), Sakurai (24) and Kaleps (25). Those of Bahling are of special interest as the dummies were restrained by lap and shoulder belts. Two sets of production cars were used, identical except that half had the roof reinforced with a rollcage. Axial compression in the dummy's neck was taken as the dependent variable. The production cars had "potentially" injurious head impacts in which the neck loads averaged 5768N compared with 3388N in the rollcaged cars.

These and most other experimental rollovers have been criticised on the grounds that they lack one of the three element of a real life rollover: travelling speed, rotation and lateral velocity. The actual mechanism of neck injury may not always be axial compression alone (the criterion used by Bahling) but bending with or without significant compression.

The relation between roof crush and injury severity, about which there has been disagreement, has been investigated by Moffat and Padmanaban (26) with an additional variable, roof strength. In a very large sample of police-reported rollovers to 41 passenger car models (1981-1991) the independent variable "injury rate" for a particular model was defined as the ratio of fatal plus incapacitated casualties to all occupants. Roof crush was defined on the TAD scale and roof strength as the ratio of the reacted load on the FMVSS test to vehicle weight.

Logistic regression identified the following factors as significant: belt use, roof damage; alcohol; gender; age; body style; aspect ratio (roof height/track); land use; vehicle weight. Roof strength was said not to appear as a significant variable. The authors therefore took roof damage as a surrogate for "crash severity". Of the vehicle factors, roof damage was found to be the most important increasing the risk of severe occupant injury by a factor of three.

While a number of authors have argued that roof crush is not causally related to injury, it may be that roof crush is the intermediate mechanism for head and neck injuries. The other aspect of the dismissal of roof crush as having a role in injury causation is that it runs counter to the "survival space" concept that is fundamental to occupant protection in frontal and side impact crashes.

Further, the geometry of the FMVSS 216 test may be less appropriate for the modern passenger car than for earlier models. Toscano (18) found, in a series of crashes leading to non-fatal SCI, that 27% of pre-1981 model vehicles experienced significant roof crush, compared with 100% of later model cars.

It is also pertinent to note the (largely successful) practice for racing cars to have well braced frame structures installed in cabins to reduce the risk of roof and other intrusion, and in turn injury severity.

CRASH INVESTIGATIONS

A sample of 43 rollover crashes was investigated in detail, with a summary of vehicle type and injury category given in Table 3. Injury information was obtained from the State Coroner's Office and major hospitals including the Spinal Unit at the Austin Hospital Detailed inspection was made of the vehicle, and, wherever possible, the crash site. Although it was intended that the sample should be representative of all grades of severity, in the event it was enriched in more serious crashes. There were 25% urban and 75% rural crashes with 40% of the latter on major roads and highways.

Table 3.
Characteristics of crash sample - vehicle type and injury category

Injury category	Vehicle type: Passenger cars	Four wheel drive vehicles	Forward control vehicles	Total cases
Fatality	8	2	2	**12**
Hospital -general -spinal -total	 8 8 16	 4 1 5	 1 1	 13 9 22
Minor	5	-	-	5
No injury	4	-	-	4
Total cases	33	7	3	**43**

The following sections summarise the key characteristics observed from these cases, and are considered in terms of injury severity, vehicle characteristics and rollover location and type. A summary of crash details and sketches for the referenced crashes is given in Appendices 1-5.

Characteristics Of Fatal Crashes

Twelve crashes involving 13 fatalities were investigated. In 10 of the fatalities, the rollover was the primary mechanism involved in the injury outcome; including six cases of full ejection. In the remaining 3 cases, rollover was involved as a secondary factor only. The seat belt wearing and ejection characteristics for these crashes are set out in Table 4. Partial ejection is defined by the head or upper part of the body moving outside the vehicle with resulting severe injury (also described by Botto et al (36) in relation to buses).

Table 4.
Seat belt wearing/ ejection characteristics for Fatalities.

Seat Belt	Nil Ejection	Partial Ejection	Full ejection	Total Fatalities
Yes	4	2	-	6
Perhaps ?	-	-	2	2
No	-	1	4	5
Totals	4	3	6	13

Table 5 sets out the measured post-crash seating height remaining after roof deformation), in comparison to the 'normal seating height' for adults (for non -ejectees only).

Table 5.
Fatal cases, Comparison of measured residual seating height left after roof intrusions and "normal sitting height"

Case No.	Gender	Height (m)	Age	Normal sitting height (mm)	Measured seating space left: 'A' (front of seat) mm	'B' (at back rest) mm
R6	M (driver)	1.71	28	870	600	800
R14	F (driver)	1.62	63	820	-	620
R39	M (pass.)	1.79	23	900	660	750

<u>**Injuries and Injury Mechanisms**</u>: Severe and gross head injuries were found to be characteristic of certain types of the rollover crashes, and occurred whether seat belts were worn or not. In these crash types, 'partial ejection' of the occupant occurs, the head is exposed outside the vehicle and may be crushed between the edge of the roof (or pillars) and road surface (Case R39, seat belt worn; refer Fig. 3 & App. 4); or abraded along the road surface as well as being crushed (Case R6, seat belts worn; refer Fig. 1 & 2, & App. 1).

In these cases the roof intrusion was both vertically and laterally towards the occupant, thereby exposing the occupant's head. Had the side window remained intact, the consequent head exposure may well have been avoided, eliminating the risk of this injury type (providing roof integrity is maintained).

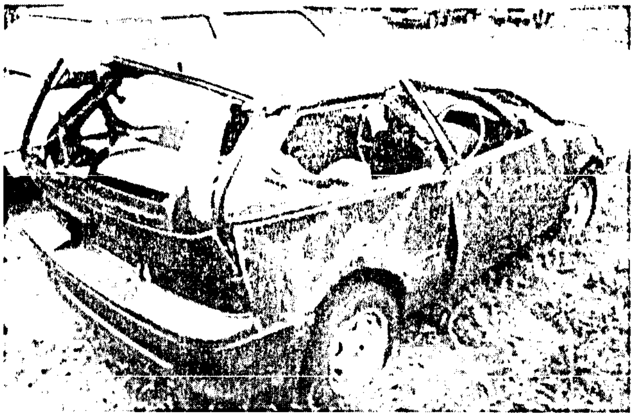


Figure 1. View of Vehicle in rollover Case R6.

Figure 2. Detail view of abrasion of A pillar of vehicle in Figure 1.

In Case R43, (1 fatality; refer Figure 4) all three unbelted occupants were ejected from a Toyota Landcruiser, mainly as a result of loss of roof integrity with the roof sheeting fully separating from its attachments to the roof framing.

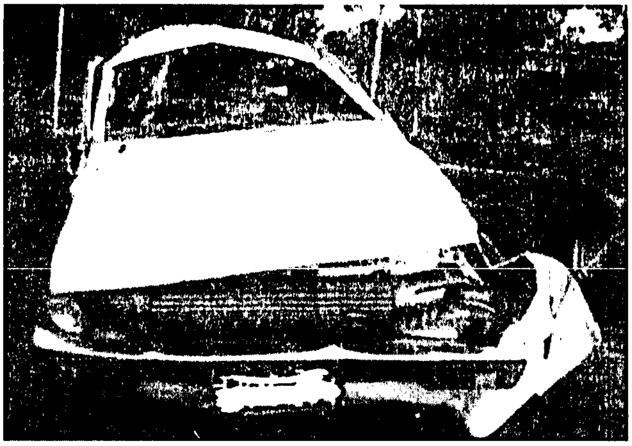


Figure 3, View of vehicle in rollover, Case R39.

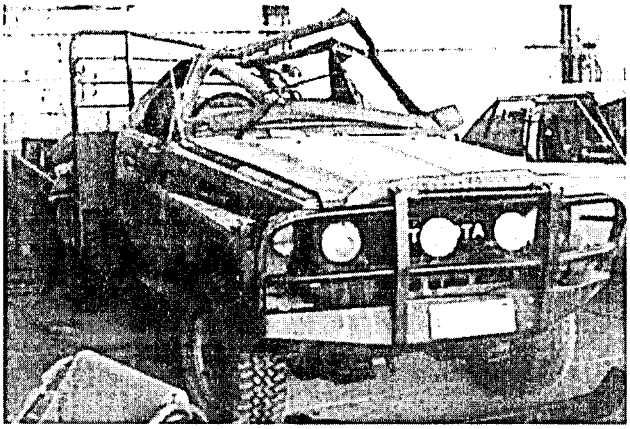


Figure 4. View of vehicle in case R43.

Characteristics Of Cases Involving Hospitalisation

This category comprised 22 crashes where the most severely injured occupant required hospital treatment. Nine of these involving severe spinal injuries which are considered separately.

These crashes are categorised by a variety of injuries and a range of vehicle damage characteristics. Head and facial injuries typically arise from contact with the roof, roof header and cant rail. In cases of severe roof crush the maximum roof deformation was located away from the occupant's seating position. Generally the roof lining consists of thin vinyl fabric, with light insulation material, running over the exposed metal edges of the roof framing, providing no effective energy absorbing padding for head contact.

Characteristics Of Spinal Injury Cases

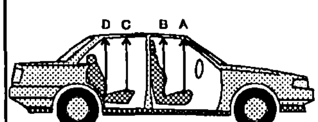
Of the nine cases, seat belts were determined to have been worn in four, another four were possible but uncertain and one case of ejection. For these cases, roof crush was rated on an increased severity level from 1 to 3 as set out in Table 6.

Table 6.
Roof deformation values for the 9 spinal injury cases

Degree of roof crush (vertical)	1 <150mm	2 150-300mm	3 >300mm	Total cases
Driver	3	1	2	6
Passenger	1	1	1	3
Total cases	4	2	3	9

The measured post-crash seating height remaining after the roof deformation, in comparison to the normal seating height for adults is set out in Table 7.

Table 7.
Spinal injury cases: Comparison of measured residual seating height left after roof intrusions and "normal sitting height"



Case No.	Male/ Female	Person Height m	Age (y)	Measured seating height post crash 'Normal sitting height' mm	A (front of seat) mm	B (at back rest) mm
R17	M	1.75	23	-	-	-
R21	M	1.64	22	810	400	640
R28	F	1.58	25	800	460	750
R34	F	-	-	-	750	900
R40	F	1.63	38	840	760	840
R42	M	1.83	30	920	560	740
R44	M	1.80	62	900	470	730
R45	F	-	-	-	790	900

Two main categories of severe spinal injury cases involving paralysis are apparent - those associated with high levels of roof deformation and those associated with low to moderate levels.

For the high levels of roof intrusion the main mechanisms appears to be axial and bending loading on the spine, via the head, arising *mainly from the reduction in vertical space for the occupant,* as indicated in Table 7. This can be seen in cases R21 (Fig. 5 & 6, & App.2) and R28 (Fig. 7 & 8; App. 3) where the roof deformation consists of the A pillar intruding significantly into the driver's head space area with the roof coming down to below steering wheel level. In each of these instances the vertical roof deformation is also associated with significant lateral deformation. This lateral deformation involved intrusion of the roof framing and the A and B pillars' encroaching on the injured occupants' seating position. In some cases the roof frame and B pillar intrude over the centre of the seat thus further reducing the occupant's head space. The occupant's spine (via the head) is, in these circumstances, subject to high bending loads.

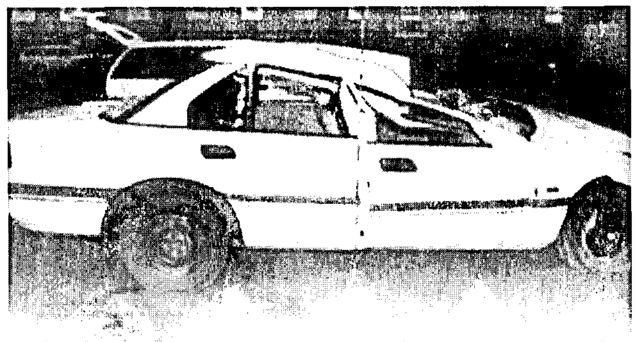


Figure 5. View of vehicle roof damage, case R21

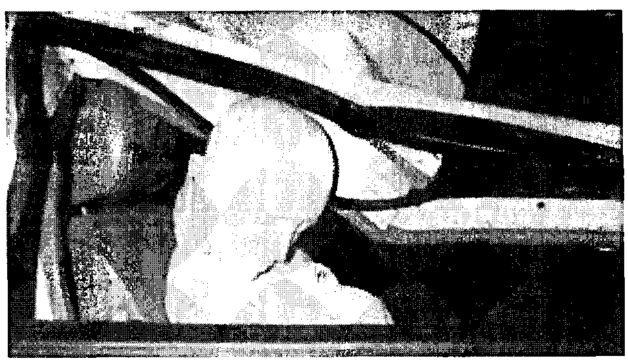


Figure 6. Detail from Fig 5. showing reduction in the occupant seating space.

This mechanism for spinal loading was clearly apparent in a fatal rollover case (not in this series) investigated by the author (27) where similar roof crush to these cases was observed. The driver was found still in an upright position in his seat, jammed between the seat and the intruded roof structure, and with his head forced down. The driver's injuries consisted of transection of the spinal column at C6, scalp lacerations and a fractured sternum. The driver's head was forced downwards by the intruded roof and roof framing, as illustrated in the sketch in Figure 9.

Figure 7. View of roof crush, Case R28

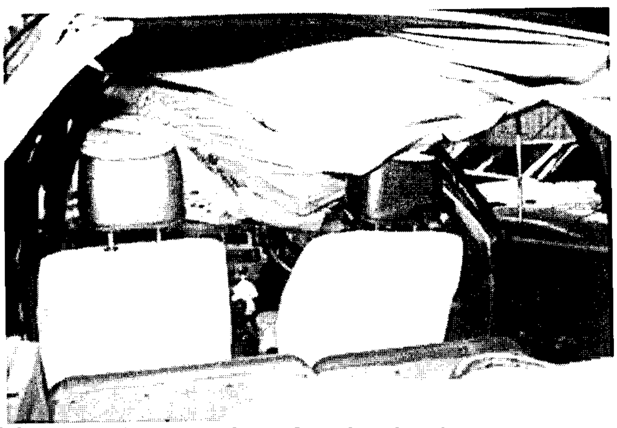


Figure 8. Interior view of vehicle in Fig 7.

In contrast, the front passenger (female) suffered minor injuries only, with no roof crush on this side of the vehicle. Another important feature to note from this case was that although the roof crush was large, this arose from the deformation of the A pillar and associated roof framing - with this high deformation being more of an indication of the relatively low load capacity of the A pillar-roof beam over the driver, than of impact severity as is sometimes asserted in the research literature.

For low levels of roof crush the spinal injuries appeared to be similar (eg. R40: C5-C6; refer Fig. 10 & App.5). In these cases the injury mechanism could have arisen from the occupants head becoming wedged against the ledge formed by the underside of the roof/door framing: as the vehicle rolls the occupant's cervical spine is loaded in bending by the body mass combined with the intrusion (and possible the shock loading transmitted from the impact with the ground through the metal framing).

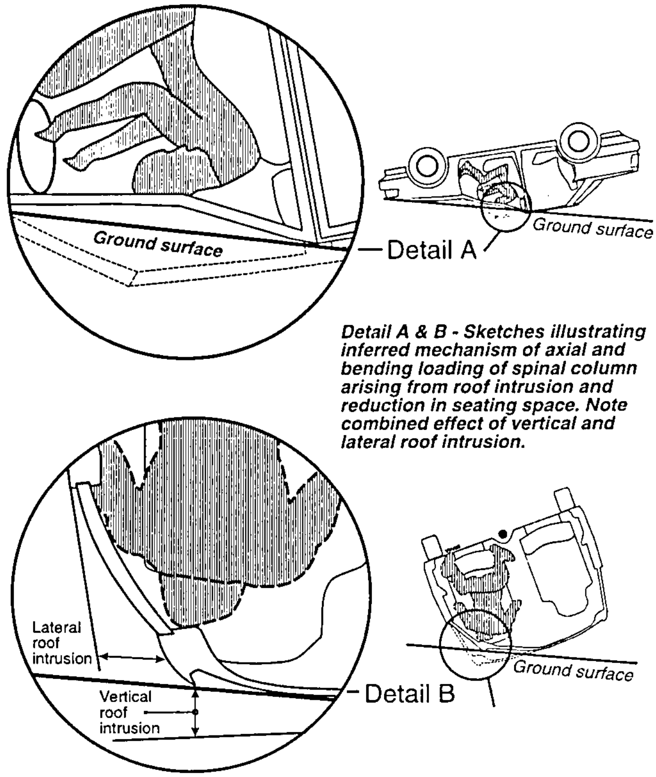


Figure 9. Mechanism of severe spinal injury in rollover crashes as may occur in cases involving high levels of roof intrusion, both vertical and lateral (head contact against ledge formed by roof/ door framing).

Figure 10. View of low level of roof crush, Case R40.

This inferred injury mechanism is illustrated in Figure 11 In these cases, because of the close proximity of the head to this position on the vehicle structure, significant loading on the head/cervical spine can occur, with intrusion accentuating this effect. Since this area is typically not padded, the impact load transmitted to the head is fairly direct and not at all attenuated.

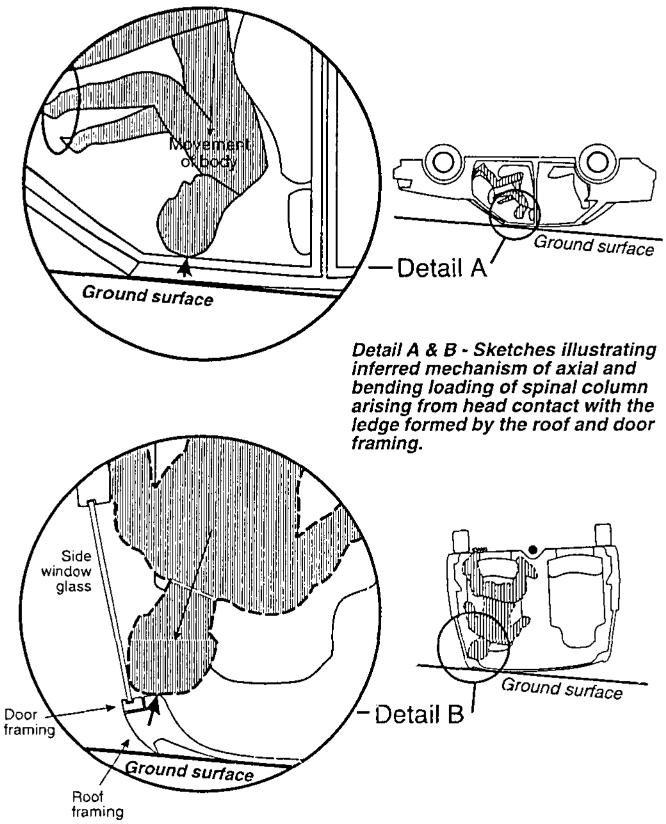


Figure 11. Inferred mechanism of spinal injury in rollover crashes as may occur in cases involving low levels of roof intrusion (head contact against ledge formed by roof/ door framing and side window glass).

Thus spinal injuries can arise directly via a 'simple mechanistic' effect where high levels of intrusion occur, or in cases with low intrusion by the head contacting the 'ledge' formed by the underside of the door/roof frame. In this latter case, where the head is in close proximity or in contact with the steel framing as it impacts the ground (or hard road surface) significant decelerations could be experienced by the body-spine-head structure. This contrasts with the non-impact side of the vehicle where typically occupant space is not compromised and in many cases the space is actually increased.

In summary, three main effects are seen as contributing to spinal injuries:

Mechanistic - simply loss of vertical head space through intrusion, and hence compression loading of the spine. This loading can of course far exceed any body inertial effects, as the car's mass itself is activated in applying this compression loading, with the spine acting as a beam/column between the roof and seat base. In these cases the extent of roof crush does not necessarily represent high levels of impact loading. For example, the energy involved in the FMVSS 216 roof crush test is typically equivalent to dropping the vehicle on to its roof from a height of only 200-300mm (28 - 33). The proposition that the risk of spinal injury would be similar with or with out roof crush is therefore invalid in these cases.

Impact loading - Through head contact with the ledge formed by the underside of the roof framing/door frame. This can be regarded as lack of head clearance space as well, although the loading appears to be inertial (body mass) combined with roof impact on a hard surface (the road). Another way of looking at this is via the concept of supported intrusion - that is, where the head strikes the vehicle structure which is supported (by the road in this case) the resulting increased stiffness may lead to more serious injuries (34, 35). This type of spinal loading could also arise in cases of moderate roof intrusion, where head contact may be with the roof structure itself, rather than the side framing.

Lack of energy absorbing padding on head contact surfaces. The main consequence of this is that decelerations are not reduced, with consequent increased head and spinal loads, and thus injuries.

These conclusions on the spinal injury mechanism gain support from the fact that there were no cases where spinal injuries were received by occupants on the vehicle side which did not impact with the ground surface. The view often expressed (that spinal injuries are not a function of roof intrusion but of impact severity) is not in agreement with the lack of injury to occupants on the non-impact side, who would also be undergoing overall body decelerations of possibly similar magnitude to the impact side occupants.

Impact severity appears to be reflected more by overall vehicle deformation, including the lateral deformation of the roof; and deformation over the bonnet (hood) /fender area, than simple vertical roof crush. Vertical roof crush is more related to the relatively low strength of the roof structure over the A pillar area, than to the high severity of impact. Of course the other factor of note is that the relevant intrusion appears to be that occurring over the particular occupant's seating position, not roof intrusion in general.

Characteristics Of Minor And Non-Injury Cases

This category comprised nine crashes where the occupants received only minor or no injuries. These cases act as a contrast group to the serious injury cases, and help clarify those characteristics of vehicle damage, roof crush and injury mechanisms which contribute to the occurrence of serious injuries in rollover crashes.

These crashes vary from low levels of roof crush to significant levels - but typically over the unoccupied seating area. For example, Case R5 (Fig. 12) shows significant roof intrusion over the vacant passenger seat area, and is suggestive of roof crush arising from a low roof strength relative to the vehicle mass rather than any significant vertical drop height.

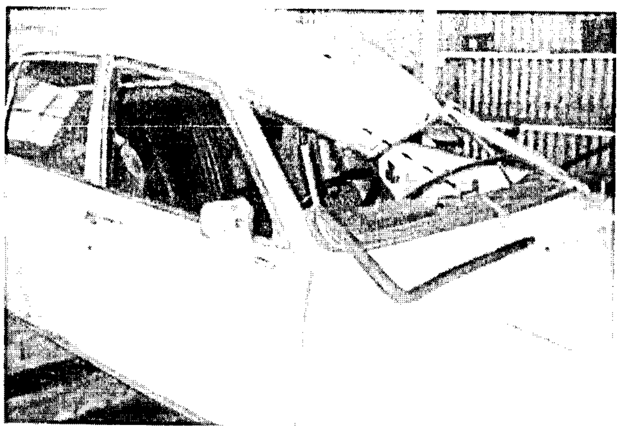


Figure 12 View of roof crush for Case R5.

In addition the degree of vehicle damage is typically less than observed in the severe injury/fatal cases where not only the roof area is commonly deformed significantly but other parts of the structure (over the bonnet, rear panels) are also involved. Overall the impression is, not surprisingly, that as with other minor injury crash types, the impact energies dissipated are generally somewhat less than in serious injury crashes.

Effectiveness Of Seat Belts In Rollover Crashes

From the 43 crashed vehicles, the most 'severely injured' occupant was considered from each crash, forming a sample of 44 occupants (one case was a double fatality). Belt use by these occupants was: belt worn 25; doubtful 13; not worn 6.

In regard to fatalities, six of the occupants did not wear seat belts and were found ejected, with injuries arising mainly from crushing as the vehicle rolled on them. There were also cases involving partial ejection where seat belts were worn, but were not effective in preventing head excursion outside the vehicle, leading to head injuries by crushing and exposure to the road surface.

In the nine spinal injury cases, four of the occupants wore seat belts, with the others uncertain. From a theoretical perspective seatbelts could increase the risk of spinal injuries in the case of significant roof intrusion as the occupant would tend to be kept upright and hence more susceptible to loading from the roof structure. This contrasts with the situation where the occupant is unrestrained and thus free to move sideways and may not necessarily be as vulnerable to bending and compression loading of the cervical spine arising from roof intrusion. These observations are not intended to discount the benefits of seat belts in rollovers, but rather to emphasis the need for compatible system design: good occupant restraint systems to be effective need to be complemented by suitable vehicle structures able to maintain the "survival space".

Overall it would appear that seats belts are partially effective in helping to reduce occupant injuries, but do not necessarily protect occupants from severe or fatal injuries where significant roof deformation (vertical and lateral intrusion) occurs or where high impact energies are involved. Seat belts do not appear to protect occupants from severe head injuries resulting from partial ejection. This process has also been noted for coach windows (36). The solution appears to be related to providing side glazing which maintains its integrity (eg. laminated glass (37)) and provides containment, hence preventing head excursion.

It is also possible that some seat belt buckle designs may be vulnerable to opening (through elbow or other limb contact with the buckle in rollover crashes), as noted in the review of fatal cases, suggesting the need for further scrutiny of buckle design for this crash type..

Vehicle Structure And Roof Structure Performance

For sedans and station wagons, the main observations in the higher severity crashes relates to the buckling and bending of the A and B pillars - resulting in roof intrusion and side sway of the roof structure. The consequence of this has been described above in terms of occupant injuries and risk.

For 4WD type vehicles, in particular, the structural integrity of the roof structure was often inadequate. The roof sheeting can separate from the framing and permit ejection of the occupants. In addition the structural strength of the A pillars appears weak relative to the vehicle mass.

The other features noted were the lack of any significant roof padding either on the roof itself or around the roof framing; and the inappropriate placement of grab handles above the occupants' heads, which may add to head injury risk.

Since rollovers can also result in significant lateral impact of the occupant against the inside face of the door, door padding (similar to that required for side impacts in general) would be expected to help reduce injuries from these impact motions.

CONCLUSIONS

- Ejection and partial ejection are significant factors in fatal rollovers. Partial ejection of the head can occur for belted occupants, with both cases in this series resulting in crushing head injuries. This partial ejection can arise due to side glass window breakage and vertical and *lateral* deformation of the roof, thus exposing the occupant's head.

- Current seatbelt designs are only partially effective in rollover crashes:

  (i) They provide little effective restraint against head excursion outside the vehicle.

  (ii) In some vehicles, the seatbelt buckle design may be deficient for rollover conditions, and may unlatch during the rollover.
- Lack of roof integrity on certain vehicle models (particularly 4WDs) contributes significantly to severe injury risk. In a number of cases, including fatalities, roof sheeting came free of the roof structure either allowing partial or full ejection.

- Nearly all roof structures and framing are unpadded and contribute to occupant head injuries, including scalp lacerations, skull fractures and brain injury.

- Severe spinal injuries arise from three main vehicle design related factors:

  (i) Mechanistic- simply the loss of vertical occupant space (by significantly exceeding the occupant's normal sitting height) through roof intrusion, both vertical and *lateral*, thereby imposing bending and compression loading of the spine.

  (ii) Impact loading - via head contact with the ledge formed by the underside of the roof and door frame resulting in inertial body load acting on the cervical spine, as well as the transmission of roof impact loading to the head from roof contact with the road surface. In these cases the head loading and impact is with "supported" structures, a situation which is known to exacerbate the risk of serious injuries

  (iii) There is a lack of effective padding on potential head contact surfaces, and consequently no intervening energy absorbing structure to reduce impact and acceleration loads acting on the spine or head.

- High levels of roof intrusion do not necessarily reflect the severity of the rollover, but also relate to vehicle roof structural load capacity, particularly over the A-pillar area. Only when combined with general levels of vehicle structure deformation is roof intrusion a more accurate reflection of impact severity. Roof crush is only really relevant when it occurs near the particular occupant's seating position.

- Severe injuries (to non-ejected occupants), particularly to the head, spine, or thorax, only appeared to occur to occupants seated on that side of the vehicle where significant roof contact with the ground/road occurred, or where there was significant roof crush. Thus these severe injuries cannot be ascribed to crash severity alone.

- The actual injuries received by occupants in a rollover, are also partially dependent on "luck"- e.g the exact position of the occupant's body as the vehicle rolls. In some cases roof crush can be high, but it would appear that the occupant's trunk and head is positioned either forward, to the side, or rearwards, ensuring that significant direct loading is not applied to the body.

RECOMMENDATIONS FOR VEHICLE DESIGN IMPROVEMENTS

Based on the observations from the detailed investigations of the 43 crashes and the literature review, the following vehicle design changes would both reduce the risk and severity of serious injuries arising in rollover crashes.

(i) Maintain side window integrity (eg. laminated glass) to prevent head excursions outside vehicle.

(ii) Increase roof framing and A and B pillar strength, for axial loading and side-sway loading, and require a minimum standard for roof integrity.

(iii) Provide interior energy absorbing padding to potential head contact surfaces - the roof itself and the framing above the door.

(iv) Modify the design of door/roof framing to reduce risk of occupant's head being able to 'lock in' against this framing and hence result in excessive spinal loading.

(v) Improve the performance of seat belts to reduce vertical and lateral movements of occupants in rollovers.

(vi) Improve door integrity and add energy absorbing side padding.

ACKNOWLEDGMENTS

The authors are indebted to many individuals and institutions for assistance. They are fully identified in the parent report (1). The project was funded by VicRoads (the roads and traffic authority for the State of Victoria).

The views expressed in this paper are those of the authors and not necessarily those of VicRoads.

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APPENDIX 1. ROLLOVER SUMMARY - CASE R6

ROLLOVER CASE SUMMARY — CASE R6

INJURY CLASS	VEHICLE DETAILS
Fatality.	Suzuki Swift, 1990 2 door hatch.

CRASH DETAILS

Vehicle struck two cows on highway, it rolled an estimated 3 times, landing back on it's wheels.

VEHICLE DAMAGE

Minor frontal deformation, right-hand roof crush, with high levels of side abrasion evident on the A-pillar and door frame on both sides of vehicle. On drivers side roof was crushed down to top of head rest with 150mm lateral intrusion.

OCCUPANT INJURIES

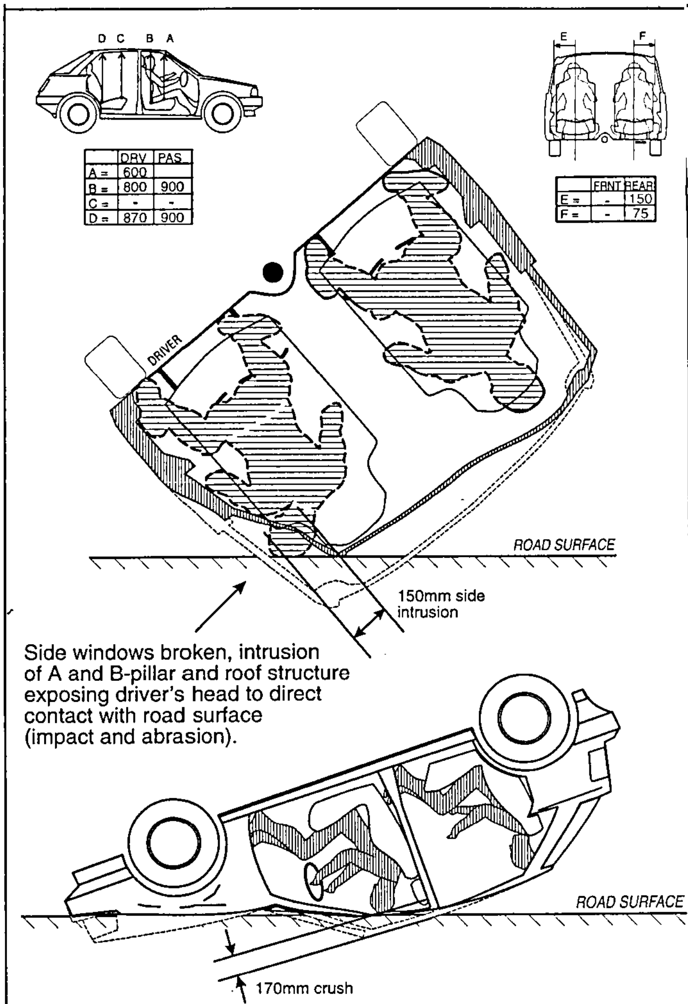
OCCUPANT	INJURIES	AIS	SOURCE OF INJURY
Driver (Male, 28y,171cm, 'slim' build) Seat belt worn.	• Scalp abrasions/lacerations • Multiple skull fractures • Diffuse sub-arachnoid haemorrhage • Disruption of cerebral cortex on frontal lobe. • Liver laceration • Contusion -face • Abrasion face & hand	1 2 3 6 2 1 1	Head exposed to direct contact/impact and abrasion with road surface.
Front Passenger	Minor injuries	-	
Rear Psgr (Drvr side)	Head injuries	-	[Details not available]
Rear Psgr (Psgr side)	Minor injuries	-	

VEHICLE DESIGN COMMENT

Roof crush plus lateral intrusion of roof on drivers side exposed the drivers head to the road surface.

POTENTIAL DESIGN COUNTERMEASURES

- Reduce roof crush and lateral deformation.
- 'Laminated' side window to retain occupant and reduce exposure to road surface.



APPENDIX 2. ROLLOVER SUMMARY - CASE R21

ROLLOVER CASE SUMMARY — CASE R21

INJURY CLASS	VEHICLE DETAILS
Severe injuries - Paraplegia.	Ford Falcon EB, 11/1992.

CRASH DETAILS

Japanese tourist driving on unmade country road in hire car. Driver lost control on long seeping bend, went off side of road and rolled after hitting embankment. [3 accidents in same section of road in past 12 months.]

VEHICLE DAMAGE

Heavy intrusion of A and B-pillar on driver's side. Crush over front fender/bonnet.

OCCUPANT INJURIES

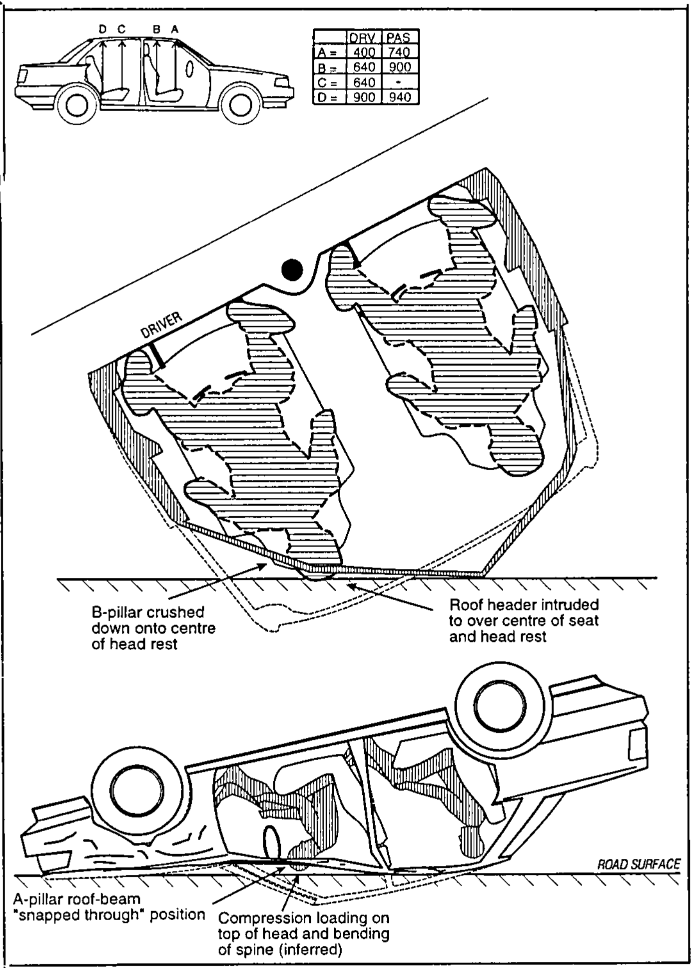
OCCUPANT	SIGNIFICANT INJURIES	AIS	SOURCE OF INJURY
Driver, 22y, 164cm, 60kg (Seat belt)	*Spinal injuries causing paraplegia* • Clinical fracture base of skull • # vertebral body C7 • Ruptured-T1 • # vertebral body T1 • Contusion L & R orbits • Lacerations skull • Contusion L & R - shoulders	 3 2 2 2 1 1 1	Loading from contact with intruded roof structure (refer sketch) resulting in compression loading on head and excess bending of spine.
Front Passenger (seat belt)	Slight injuries - minor abrasions on foot.	-	-
Rear passenger	Uninjured	-	-

VEHICLE DESIGN COMMENT

Roof structure for this model has been improved by use of void fillers in A & B pillars - however A-pillar strength inadequate.

POTENTIAL DESIGN COUNTERMEASURES

Strengthen A & B pillar to reduce intrusion in rollovers.



APPENDIX 3. ROLLOVER SUMMARY - CASE R28

INJURY CLASS	VEHICLE DETAILS
Hospital. (quadriplegic)	Ford Laser, 1989. 5d hatch

CRASH DETAILS

Driving on winding, gravel road in hilly country. Driver lost control on bend, and went over the edge of the road down long steep embankment, rolling several times.

VEHICLE DAMAGE

Extensive panel damage, heavy roof crush and intrusion on driver's side.

OCCUPANT INJURIES

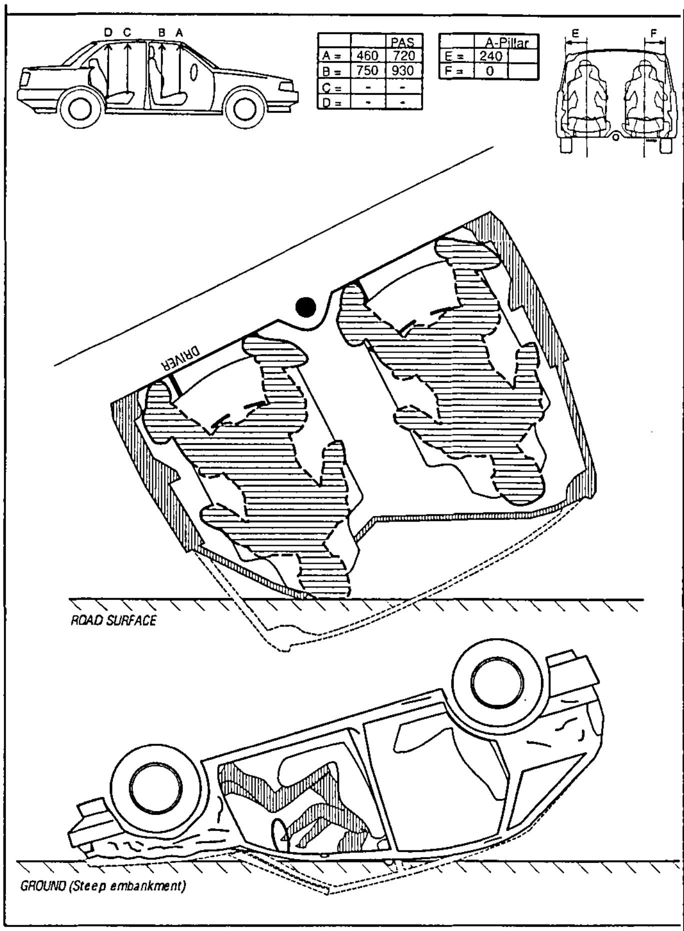
OCCUPANT	INJURIES	AIS	SOURCE OF INJURY
Driver, Female, 25y, 44kg, 158cm (seat belt on)	*Quadriplegic*:		Loading on head during rollover & roof crush. Belt came loose and drivers head was out of window during one roll, according to passenger (husband). Also stated that saw his wife's head tilted in extension and lateral flexion, possibly during roof crush phase..
	• # C2 -stable	2	
	• #C6/C7 -displaced, unstable	2	
	• LOC (1hr) retrograde amn.	2	
	• Abrasion R side head	1	
	• L lung contusion	3	
	• Contusions-R upper arm	1	
	• Abrasion R shoulder	1	
	• Abrasion R leg	1	
	• Contusion R leg	1	
	• Contusions L leg	1	
	• Lacerated L knee, ear	2	
Front Passenger, M, 28y, 70kg, 183 cm (seat belt worn)	• Lacerations to scalp (vertex)	1	roof contact and general interior contact
	• Contusion face	1	
	• Soft tissue damage to back	-	
	• Laceration R leg	1	

VEHICLE DESIGN COMMENT

Inadequate roof strength over driver's space

POTENTIAL DESIGN COUNTERMEASURES

Increase roof strength particularly over A-& B-pillar areas, to reduce deformation into cabin



APPENDIX 4. ROLLOVER SUMMARY - CASE R39

INJURY CLASS	VEHICLE DETAILS
1 Fatality Hospital	Datsun 120y wagon 1975

CRASH DETAILS

Car went onto soft edge of road and overcorrected, rolled several times on bitumen road. Deceased passenger found still in seat with seat belt on.

VEHICLE DAMAGE

Roof crush on passenger side (fatality) and heavy distortion of vehicle body. Significant panel abrasion and distortion; heavy outward distortion of driver's door, passenger's door also bowed outwards. Driver's seatbelt found torn.

OCCUPANT INJURIES

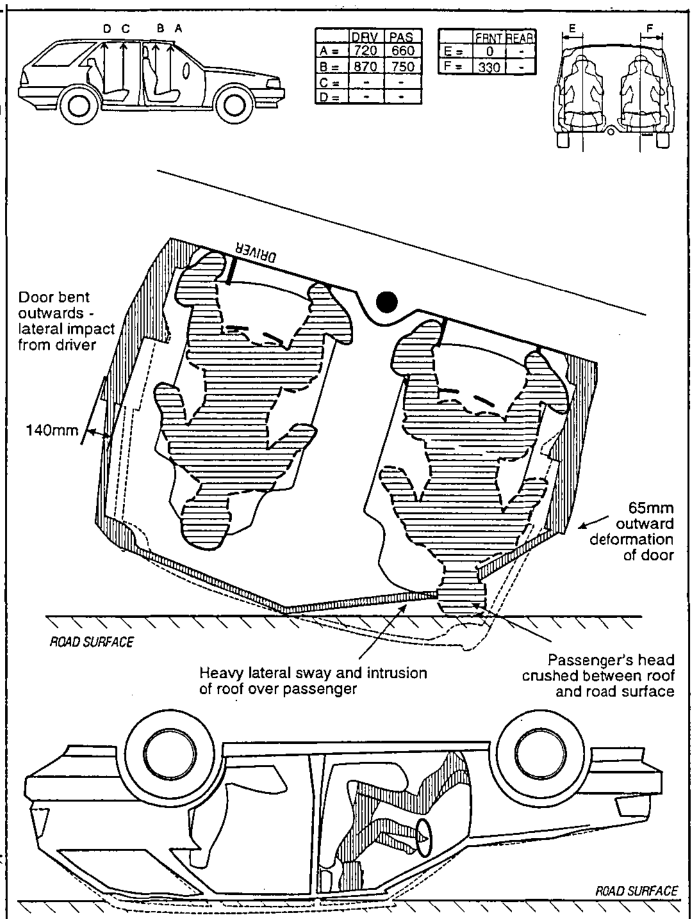
OCCUPANT	INJURIES	AIS	SOURCE OF INJURY
Driver, Female 21y, 48kg, 152cm	• Fractured pelvis (pubic ramus)	2	-impact with door-
	• Lung contusion	3	
	• Abdominal mesenteric tear,	2	-seat belt(?)
	• Abrasions L & R foot	1	
Front Psgr, Male, 23y, 76kg, 179cm (seatbelt on)	• *Deceased.*		Head crushed between roof (above door frame) and road surface.
	• Contusion R lung	3	
	• Comminuted # vault of skull	3	
	• Avulsion cerebral hemispheres from cerebellum & brainstem	6	
	• Extradural haemorrhage	4	
	• Subdural haemorrhage	4	
	• Abrasions forehead, neck, ear, forearm, wrist, hand, fingers	1	

VEHICLE DESIGN COMMENT

- Driver's door was bent outwards 140mm and 'flattened' ie. the inner and outer faces of the door were compressed together, probably from the lateral impact of the driver. The compression of the door meant that there was no 'padding' left to absorb energy from occupant contact, particularly if contact was made as the vehicle rolled and the door was reacting against the road surface.
- The roof intrusion and lateral deformation of the roof structure exposed the driver's head to the road surface and crushing between the vehicle and road.

POTENTIAL DESIGN COUNTERMEASURES

- Stronger roof structure to reduce intrusion and sidesway.
- Laminated side windows
- Improved door structure, with interior padding for occupant protection in lateral loading



APPENDIX 5. ROLLOVER SUMMARY - CASE R40

ROLLOVER CASE SUMMARY	*CASE R40*

INJURY CLASS	*VEHICLE DETAILS*
Hospital. Quadriplegic.	Holden Commodore station wagon 1992

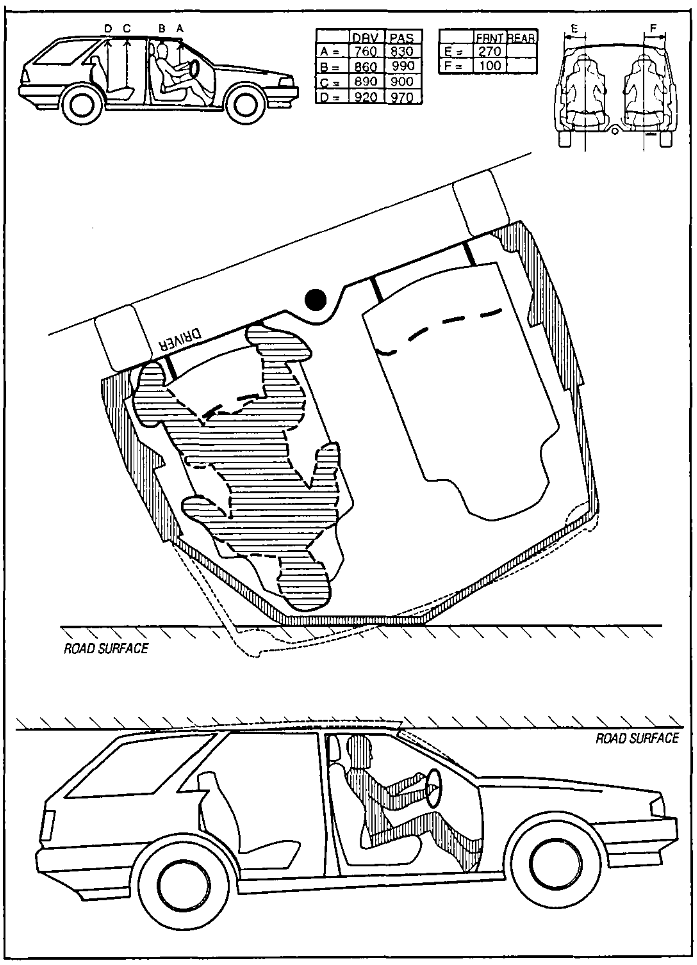
CRASH DETAILS
Rural road, driver lost control on gravel shoulder, overcorrected, cut across bitumen road and struck drainage ditch: witness claimed that the car 'somersaulted' 2 to 3 times.

VEHICLE DAMAGE
Impact on right-hand side, some side sway of cabin, moderate roof crush, driver's window broken, no signs of loading on seat belt.

OCCUPANT INJURIES			
OCCUPANT	*INJURIES*	*AIS*	*SOURCE OF INJURY*
Driver, Female, 38y, 57kg, 163cm (seat belt probably)	*quadriplegia*		Head impact with roof-probably door framing/ roof framing.
	• # dislocation C5-C6	2	
	• # vertebral body C5	2	
	• Contusion forehead	1	
	• Lacerations R forehead	1	
	• LOC (few seconds)	2	
	• Contusion-R arm, L arm	1	
	• Laceration R hand, L hand	1	

VEHICLE DESIGN COMMENT
Roof structure withstood rollover loading fairly well, with low levels of intrusion.

POTENTIAL DESIGN COUNTERMEASURES
• Seatbelt pretensioner • Roof padding



DEVELOPMENT AND TESTING OF THE UNIVERSAL COACH SAFETY SEAT

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Paper Number 96-S5-O-11

ABSTRACT

The paper presents the development and performance results of a new 'universal' coach safety twin seat (patent pending) which carries three-point belts, but also protects occupants sitting behind. Four main scenarios investigated comprised an empty and fully loaded seat impacted from behind by unbelted or lap-belted dummies. The extreme 12 g reverse acceleration pulse was applied on a HyGe sled rig and the pitch of 750 mm was interpreted as 'spacing' as defined in the ECE Regulation 36. All the injury criteria specified in the ECE Regulation 80 were met in both Hybrid II and Hybrid III dummies, as well as the Hybrid III neck injury levels. The seat performed well with 95 % and 5 % dummies, as well as with a pitch of 650 mm, and it passed the 76/115/EEC belt anchorage test. The mass of the feasibility prototype twin seat with one leg and anchorages is 36.3 kg, it is made using conventional materials and production techniques, demonstrating full technical and commercial feasibility.

INTRODUCTION

A number of serious coach accidents in UK and elsewhere in Europe recently generated a growing concern about the feasibility of improving coach safety by fitting seat belts. In the late 1994 the European Commission initiated a project to deal with the problem. The results have been presented in Ref. 1 and some of the main conclusions were :

(a) Passenger *ejection* is a major cause of death and injury particularly in minibus and coach rollovers, but also in frontal and side impacts. *All* minibus and coach seats are 'exposed' in rollovers and many also in other accidents.

(b) Seat belts can significantly reduce or prevent passenger *ejection*, but the whole system : seat, seat belts and all anchorages must be properly designed, manufactured, installed and used.

(c) Some of the R80 compatible seats approved with *unbelted* dummies can maintain acceptable injury levels even when dummies are lap-belted. However, lap belts increase the head strike exposure and the belt angles should be controlled to reduce danger of abdominal injuries.

(d) Seats with three point belts offer, in principle, the best protection to belted occupants, but can be 'too hard' to those sitting behind, particularly if they wear lap belts. This raises important questions whether 'hard' 3-point belt seats can be mixed with other types and whether unbelted occupants also deserve protection.

(e) Combined loading, with lap or 3-point belted dummies in the test seat and unbelted dummies behind impose very much higher seat and anchorage loads and a significant geometry change in comparison with a standard R80 test. Combined loading should be considered, so that safety of belted occupants is not compromised by the unbelted occupants sitting behind.

(f) Dynamic tests on seats, anchorages (and obstacles if any exist in front of a seat with lap-belts) ought to be applied to reflect the body dynamics in front impacts and particularly the head-strike.

(g) An 'ideal' solution would be a seat that can protect occupants under all conditions, i.e. unbelted, lap belted and 3-point belted, as well as any combination of these. These options were met with much scepticism in terms of their commercial feasibility (size, cost, weight).

PROJECT OBJECTIVES

The objective of the Project was to investigate the feasibility of an *M3 coach twin seat* that would meet the following requirements :

(a) the *current* ECE R80 (empty) seat test with unbelted dummies sitting behind ;

(b) the *proposed* conditions combining the current ECE R80 (empty) seat test with *lap-belted* dummies behind (Draft Commission Directive III/5162/94/EN concerning the Directive 74/408/EEC) ;

(c) the *combined loading*, involving 3-point belted dummies in the seat and *unbelted* dummies behind;

(d) the *combined loading*, involving 3-point belted dummies in the seat and *lap-belted* dummies behind ;

(e) *weight* and *cost-related constraints* that would make the seat *commercially* feasible ; this feasibility may be justified even if such seat can be *safely* used (as regards rear passengers too) only in the 'exposed' seating positions ; however, the maximum possible benefit would arise from an 'all round', large quantity seat to which lap or 3-point belts can be retrofitted.

Conditions (c) and (d) go much beyond the proposed conditions under (b), not only in terms of the level of loading conditions, but also as regards the geometry change of the tested seat occupied by belted dummies.

The current R80 compatible seats are designed to protect unbelted occupants only and the Australian Design Rule 68/00 (first approved in October 1992) concerns safety of the 3-point belted occupants only. To the best knowledge of all people involved in this Project, a seat meeting criteria above neither existed, nor has a similar attempt been made in the past.

GENERAL APPROACH TO THE PROBLEM

The work programme included :

(a) Engineering background study,
(b) Component tests,
(c) Occupant and structural simulation studies,
(d) Concept selection and parametric studies,
(e) Proposal for the main deforming and non-deforming components,
(f) Blending the safety features with the other functional and manufacturing constraints of an assumed 'production seat',
(g) Production of seat test prototypes,
(h) Dynamic tests on a HyGe reverse accelerator rig,
(i) Static test on anchorages according to the Directive 76/115/EEC.

ENGINEERING BACKGROUND AND DEFINITION OF TEST CONDITIONS

This investigated the conclusions arising from the EC seat belt project (Reference 1.), past CIC projects on aircraft, railway and coach seats and from literature. The current study generated the framework within which the concept solution was found and identified priority issues in the theoretical and experimental work. Decisions reached at this stage were, for example, to :

(a) achieve the solution by a controlled and progressive collapse of the seatback and underframe structures ;

(b) interpret the 'seat pitch' as 'seat spacing' in the ECE Regulation 36 (i.e. measured between the front of the seat back of the auxiliary seat and the rear of the seat back of the seat tested), which increases the distance between the 'equivalent' points on the auxiliary and test seats by approximately 50 to 60 mm and creates a *more severe* test scenario ;

(c) make the test conditions *as severe as possible* while still within the bounds of Regulation 80 on *acceleration and total velocity* change. The test pulse was therefore tailored to be very close to 12 g over approximately 30 ms of acceleration, with a total velocity change of 30 kph - this has brought the seat as close as was possible to the M2 minibus category too.

COMPONENT TESTS

Component tests comprised :

(a) *Static bending collapse tests on a selection of mild steel tubes* to establish the reference material properties. These were used to predict bending moment-rotation curves and section optimisation using in-house program WEST .

(b) *Static compression tests on the seat cushions and backs* to enable correct dummy positioning in the dynamic occupant / seat simulations using program MADYMO from TNO.

(c)*Dynamic pendulum tests on the energy absorbing materials* to estimate the impact response of different combinations of materials and geometries and generate MADYMO input data. The tests were carried out using a pendulum (Figure 1.), made of aluminium which carried a mahogany head former sculptured to represent closely the face of the HYBRID III dummy.

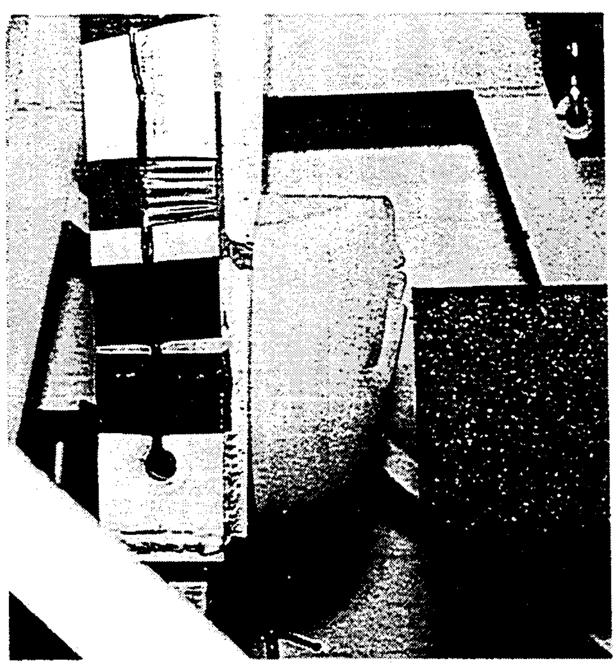

Figure 1. Pendulum tests with a Hybrid III head form were used to investigate the head-strike region ; this close up side view shows the impactor close to the point of contact with a foam block

An accelerometer was mounted centrally immediately behind and in line with the head-to-seat-back impact. The seat back test specimens with different padding of the head impact zone were mounted on a frame with a seat-back like geometry. The angle of impact was adjusted to approximate the relative kinematics of the head-strike of a lap belted occupant against an empty seat in front. Test results included the impact velocity, deceleration signal, visual inspection and photography.

ANALYTICAL STUDIES

The analysis activities were carried out broadly in parallel with the component tests, with some additions after the first full scale tests. Due to the extreme time pressures and difficulties with access to more elaborate one-pass dynamics analysis using program DYNA3D, the simulation relied on *quasi-static structural* study with the in-house program SIMSTAT (recent version of CRASH-D), linked in an iterative loop with a *dynamic occupant / seat inter- action* analysis with program MADYMO.

The main role of the *static* analysis was to :

(a) 'translate' the general collapse properties of the seat structure indicated as favourable by the dynamic simulations into design recommendations on the main deforming and non-deforming members at the levels of the overall layout and component collapse properties.

(b) feed input data back to the dynamic analysis after assuming a certain design and generating the component and overall collapse properties ;

(c) control the compatibility of the MADYMO and structural collapse modes and identify design problems and favourable collapse modes.

Component moment-rotation curves at plastic hinges were generated using program WEST and fed into program SIMSTAT for the analysis of the complete structure. The loads were applied in two or three directions at seat belt anchorage points, knee, torso and head contact region using axially collapsible beams whose load-deflection curves reflected the loading sequence observed in earlier tests (Reference 1), or dynamic analysis.

However, static analysis has serious limitations in identifying the truly dynamic effects of the interaction between the dummy and deforming seat, represented by highly oscillating time histories with shifts between different contact regions.

Dynamic analysis was carried out to :

(a) investigate different concept solutions, both in qualitative and quantitative terms,

(b) investigate the *chosen* concept as regards the effect of the collapse properties of the structure and contact regions on the occupant kinematics and injury criteria.

The main difficulties concerned reliability of the input data and the potentially high sensitivity of the injury criteria to relatively small variations of input parameters (both aided by quasi-static analysis and pendulum tests).

The seats were studied in the following four configurations (Fig. 2), each implying different conditions for both the seat and dummies (all 50 %ile male) :

(a) Front dummies 3-point belted - rear lap-belted,
(b) Front seat empty - rear dummies lap-belted,
(c) Front dummies 3-point belted - rear unbelted
(d) Front seat empty - rear dummies unbelted.

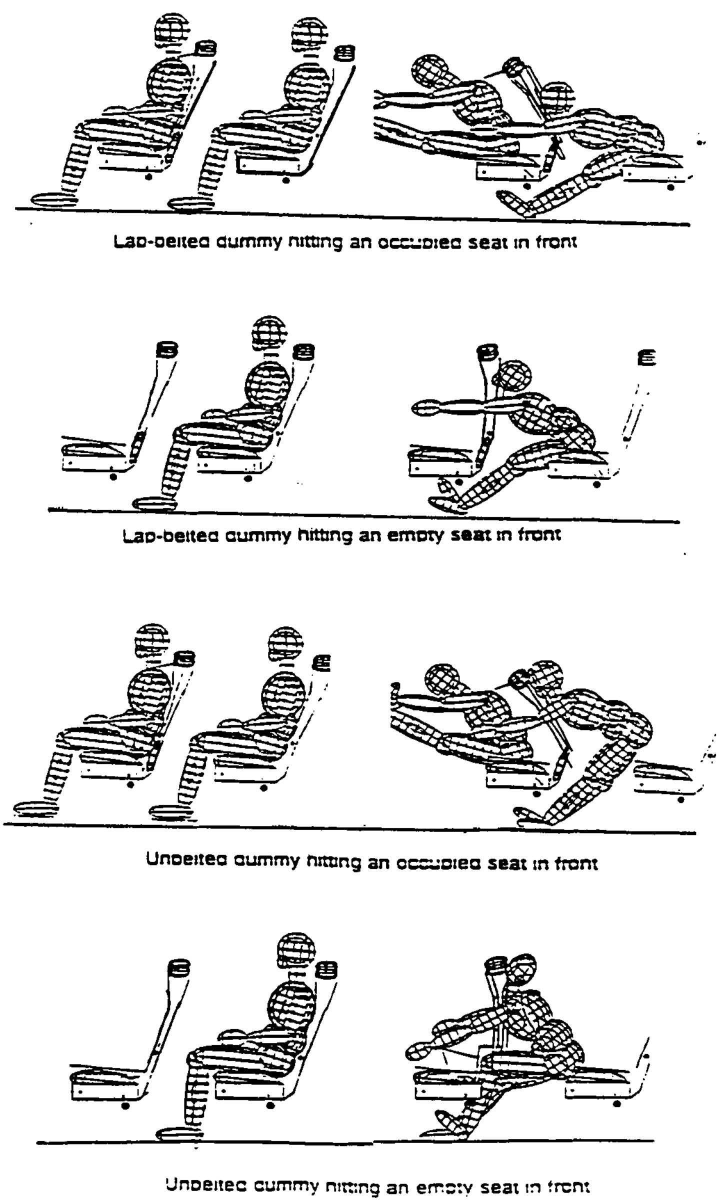

Figure 2. Occupant / seat simulation on the four characteristic test scenarios

The parameters considered in the dynamic study were the head and knee contact stiffnesses, seatback structure strength, underframe stiffness and strength (in a 'macro' sense, i.e. not including individual structural elements) and seat belt stiffness.

Table 1. shows the dummy results obtained with a 'recommended' design used in the **first** test on each scenario in Figure 2. Results correspond to the rear seat dummies with top figures predicted by analysis and the bottom two corresponding to the test results (first the "window" side occupant (50 %ile Hybrid II), then, after the '/' sign, the "aisle" side (50 %ile Hybrid III).

Table 1.

Front seat	Rear seat	HAC theory/ test limit : **500**	Chest accel. theory / test (g) limit: **30**	Femur load theory / test (kN) limit : **10**
Empty	No belt	264 213/ 359	18.3 29.2/ 21.2	7.34 4.5 / 4.7
Empty	Lap belt	803 436/ 356	18.3 29.8/ 19.0	4.89 3.7 / 4.0
3-point belted	No belt	457 567/ 302	9.7 21.2/ 12.3	5.53 5.1 / 4.3
3-point belted	Lap belt	482 871/ 633	23.8 22.6/ 14.1	2.35 3.6 / 2.8

The Table is illustrative not only as regards the agreement between the theoretical and the first experiments, but also in terms of the scatter of the test results. Further 'fine' tuning of the seat was necessary, mainly based on additional pendulum tests.

THE BASIC DESCRIPTION OF THE NEW SEAT

The concept of the new seat was based on the desire to achieve both the safety *and* commercial requirements. This immediately led to a decision to aim for :

(a) utilisation of conventional materials and production methods, typical for the bus and coach seat manufacturing industry,

(b) a design that would, ideally, be suitable as a standard seat for *all seating positions*, resulting in the following benefits :

b1 : high number of units, hence lower unit cost,
b2 : possibility to offer lap or 3-point belts as options on the same seat,
b3 : possibility to retrofit lap or 3-point belts after the original purchase.

The same requirements also enhanced the need to minimise the weight of the seat. Weight minimisation often implies use of advanced materials, such as aluminium and composites. However, it was only practical (and fully justifiable) to go for the same basic materials used in other seats, i.e. steel tubes, plywood, low-cost foams (meeting the Directives on the burning behaviour of interior materials used in buses in coaches - which is not yet in force) and a range of brackets, joints, etc. also made of steel and using conventional production procedures. The total weight with one leg and anchorages was 36.3 kg.

The 3-point belt was of standard configuration. Both shoulder belt slots were in the middle of the seat, to remove them from the aisle and have a common seat for the left and right side of the coach. The retractors were under the cushions.

The seat design, which is patent pending, was chosen to combine :

(a) compatibility with all the geometry / installation requirements for M3 vehicles ;

(b) stiff and strong leg and all anchorages, attached to standard vehicle rails ;

(c) deformable seat underframe ;

(d) deformable and detachable seat back, for production / fit consideration ;

(e) provision for the reclining mechanism (which was however reinforced and blocked in all tests, except for one rear seating position in Test 6) ;

(f) contact properties in the knee, head and chest contact regions.

FULL SCALE DYNAMIC TESTS

Full scale dynamic tests were carried on a HyGe reverse accelerator sled rig, with test conditions adjusted to be *as severe as possible* for the M3 vehicles, as described above. The auxiliary (launch) seats were standard production seats and compatible with ECE Regulation 80. In view of the geometric similarity of the front face of the standard and the new seats and since the R80 seats with lap-belted dummies did not appear to deform permanently, it is argued that the test conditions reasonably represented the case where all seats are of the new type.

All seats were mounted on rails identical to those used in standard reference coaches.

All the tests (except Test 5 which involved seat rows 1 and 2 only) examined two scenarios using a single firing of the HyGe sled rig (Figures 3 and 4). One scenario was tested using rows 1 and 2 and the other with rows 3 and 4. The test arrangement was :

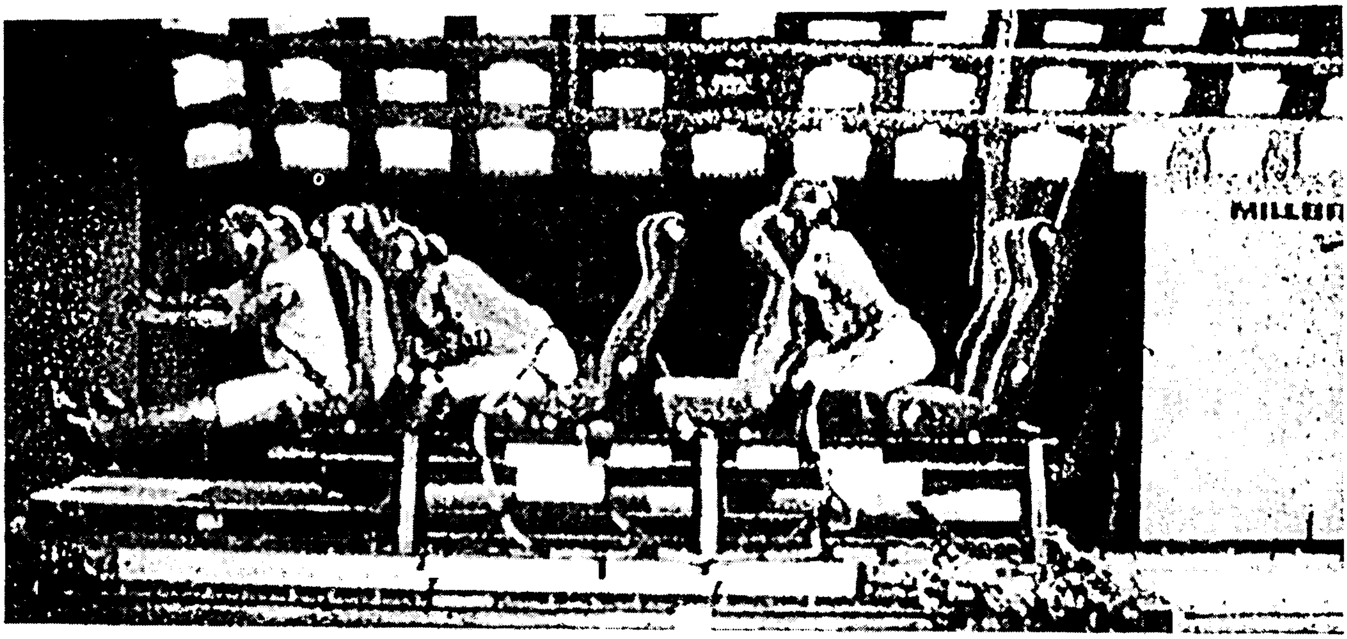

Figure 3 HyGe sled test of the universal coach safety seat with 50 %ile male dummies
front row (new seat) : 3-point belted, uninstrumented Hybrid II in both seats,
second row (R80 seat) : lap belted, instrumented dummies - near (window) seat Hybrid II, next to Hybrid III
third row (new seat) : empty
fourth row (R80 seat) : unbelted, instrumented dummies - near (window) seat Hybrid II, next to Hybrid III

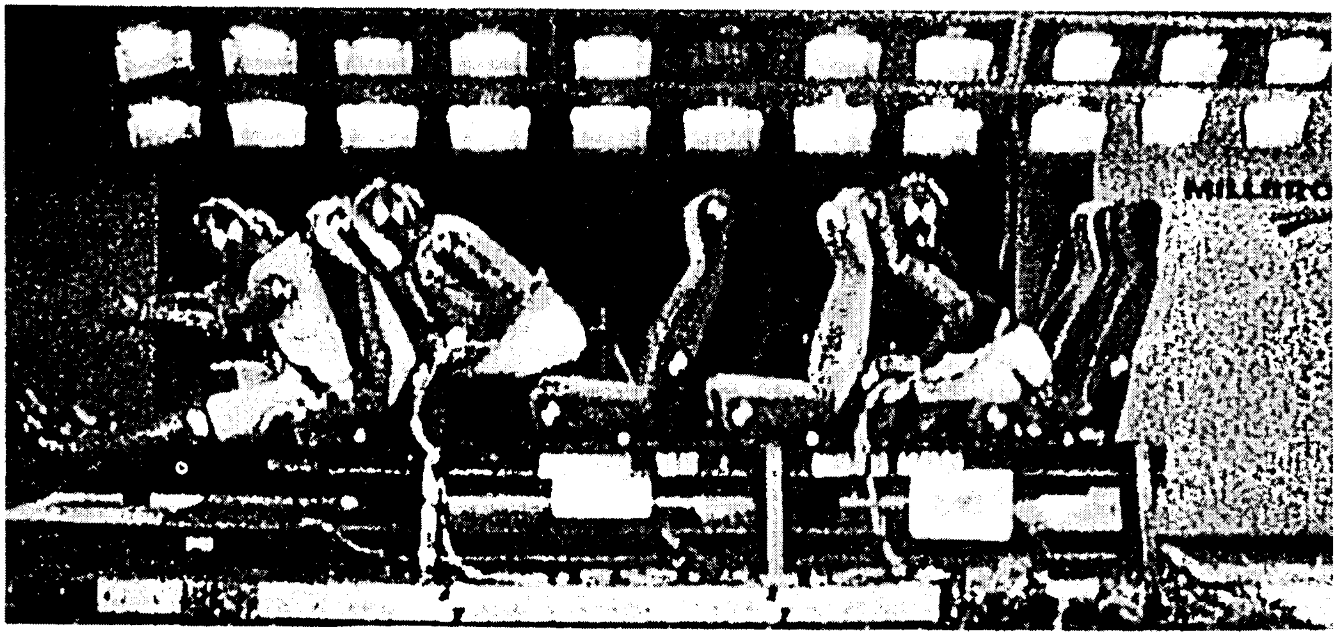

Figure 4 HyGe sled test of the universal coach safety seat with 50 %ile male dummies
front row (new seat) : 3-point belted, uninstrumented Hybrid II in both seats,
second row (R80 seat) : unbelted, instrumented dummies - near (window) seat Hybrid II, next to Hybrid III
third row (new seat) : empty
fourth row (R80 seat) : lap-belted, instrumented dummies - near (window) seat Hybrid II, next to Hybrid III

Row 1 : tested seat - new seat in Tests 1 to 6, loaded with uninstrumented, 3-point belted Hybrid II dummies,

Row 2 : launch seat - R80 seat in all tests, loaded with instrumented Hybrid II (window seat) and Hybrid III (aisle seat) dummies (50 %ile in tests 1 to 5, and 95 %ile (aisle) and 5 %ile (window) in Test 6) ; dummies were unbelted in Tests 1 and 2, or lap-belted in Tests 3 to 6,

Row 3 : tested seat - empty in all tests,

Row 4 : launch seat - standard R80 seat in all tests, with instrumented Hybrid II (window) and 50%ile Hybrid III dummies, unbelted in Test 3, otherwise lap-belted.

The *'pitch'* (i.e. seat spacing) between the test and launch seats was 750 mm in all Tests, apart from rows 3 and 4 in Test 6, which had a spacing of 650 mm. The gap between the seat back in row 2 and cushion in row 3 was approximately 150 mm, so that there was no interaction between the seats in rows 2 and 3.

The reference injury criteria required by the ECE Regulation 80 and the new draft amendments to the 76/115/EEC Directive were:

(a) the head injury/acceptance criterion (HAC) : 500

(b) the chest (thorax) acceleration acceptance criterion (ThAC) : 30 g for up to 3 ms

(c) the femur force acceptance criterion (FAC) : 10 kN (8 kN for more than 20 ms).

Although not part of any safety legislation, the *neck injury criteria* in most Hybrid III dummies were also measured to investigate an important injury mechanism. These are based on the comparison of the processed time histories (level - duration) of the neck loads with a 'tolerance corridor' and expressed in percents of the tolerance limit. An 'acceptable' result reads less than 100 %.

The main test conditions and results are summarised in Table 2. Extracts from the high speed films from each test scenario are shown in Figs 3 and 4.

Test 1 was supposed to be preceded by a static seat belt anchorage test to investigate whether joint separation may occur. Unfortunately, this could not be done and indeed the seat back in the first row fractured at the reclining mechanism and had to be re-designed.

Test 2 and all subsequent test had an identical main structure. The seat met all the structural, dummy kinematics and injury criteria, including the extra neck-related data, with the exception of the unbelted Hybrid II dummy in the second row whose HAC was just above the limit (567). There was no damage to anchorages and all seats stayed firmly anchored after test. This demonstrated that the same seat can carry *3-point belted occupants* and also restrain the *unbelted* passengers behind. When empty, it also protects the rear *lap-belted* passengers.

Test 3 included some improvement of the head strike region and met all the structural, dummy kinematics and injury criteria (Table 2) for the *unbelted* occupants sitting behind the *empty* seat. The fore and aft neck criterion was, however, marginally above the limit (110 %). The head strike region had to be improved again also because both lap belted dummies in the second row failed the HAC (871 and 633), but passed other requirements.

Test 4 confirmed that the worst case scenario (at least with this particular seat) when the seat loaded with 3-point belted passengers also had to protect lap-belted occupants behind.

Test 5 showed (Table 2) full compliance with all the injury criteria in the last remaining case of the seat protecting both the *3-point belted* 50 %ile dummies and simultaneously, the *lap-belted 50* %ile dummies behind.

Test 6 was organised to investigate how the seat in Test 5 would perform with the arrangement involving :

- row 1 - loaded with 3-point belted 50 %ile dummies,
- row 2 - lap belted dummies: 5 %ile small female dummy in the window seat next to 95 %ile (large male)
- row 3 - empty
- row 4, at 650 mm pitch - lap belted dummies : 50%ile Hybrid II (window seat) next to a 50 %ile Hybrid III.

All the injury criteria were below the limit (Table 2), except the HAC of the small (5 %ile) dummy in the second row, which read 587. This dummy hit the lower edge of the foam block that was, only here, stiffened to blend the padding with the back of the seat. This may have raised the HAC and confirmed the higher sensitivity of the smaller dummies as observed in the earlier EC project by the same team. The lower injury readings with 650 mm pitch confirmed the earlier findings that the smaller pitch benefits the safety of occupants.

STATIC TEST ON BELT ANCHORAGES TO THE 76/115/EEC DIRECTIVE

The static test of the seat belt anchorages to the Directive 76/115/EEC, as amended in 90/629/EEC was carried out (Figure 5) on one seat after completion of the dynamic tests. The M3 specification required 450 daN for each torso belt and for each lap belt 450 daN plus 6.6 times half of the seat weight (i.e. a total of 567 daN). The requirement was met (Figure 6) with the seat top moving forward to a maximum of approximately 200 mm and without any material separation.

CONCLUSIONS

A feasibility prototype twin coach seat with three-point belts has met all the ECE Regulation 80 injury criteria under the 'worst' acceleration and seat pitch conditions for the M3 coaches under the following scenarios : empty seat hit by unbelted dummies, empty seat hit by lap-belted dummies, fully loaded seat hit by

Table 2.

Summary of all the Dynamic Test Configurations, Pulse Details and Injury Results

Seats	Seat Spacing	Test 1	Test 2	Test 3	Test 4	Test 5	Test 6
1st	All seat spacing 750mm, except between rows 3 & 4 in Test 6	new	new	new	new	new	new
2nd	and all seats in Test 7 (650mm)	R80	R80	R80	R80	R80	R80
3rd		new	new	new	new	Not used	new
4th		R80	R80	R80	R80	Not used	R80
	Dummies						
1st	All dummies Hybrid II (50 %ile male)	3-point belted	3-point belted	3-point belted	3-point belted	3-point belted	3-point belted
2nd	window : Hybrid II (50%ile) aisle Hybrid III (50%ile), except test 6 with : : window 5 %ile, aisle 95 %ile	Unbelted	Unbelted	Lap Belted	Lap Belted	Lap Belted	Lap Belted
3rd		Empty	Empty	Empty	Empty	Not used	Empty
4th	Hybrid II in window seats for other tests	Lap Belted	Lap Belted	Unbelted	Lap Belted	Not used	Lap Belted
Dummy	Parameter						
Second row Window	Head HAC	397	567	871	697	413	587
	Chest 3ms max (g)	11.1	21.2	22.6	24.9	21.6	23.6
	Maximum Left Femur Load (kN) (compression +ve)	5.4	5.1	3.6	3.4	4.0	-1.7
	Maximum Right Femur Load (kN) (compression +ve)	1.5	2.9	0.7	0.4	1.0	-1.3
Second row Aisle	Head HAC	Channel Malfunction	302	633	604	439	421
	Fore / Aft Neck Criterion (%)			39.9 / 24.3	14.6 / 76.7	59.3 / 11.9	
	Tension / Compression Neck Criterion (%)			56.3 / 5.3	54.7 / 5.1	79.4 / 57.5	
	Chest 3ms max (g)	10.0	12.3	14.1	12.8	12.6	15.1
	Maximum Left Femur Load (kN) (compression +ve)	1.7	4.3	2.8	2.7	2.8	3.4
	Maximum Right Femur Load (kN) (compression +ve)	3.9	3.4	2.5	2.3	2.8	3.0
Rear row Window	Head HAC	240	436	213	347		275
	Chest 3ms max (g)	25.8	29.8	29.2	24.5		22.8
	Maximum Left Femur Load (kN) (compression +ve)	5.4	3.7	4.5	4.1		4.9
	Maximum Right Femur Load (kN) (compression +ve)	2.6	3.3	3.8	2.6		3.2
Rear row Aisle	Head HAC	307	356	359	575		177
	Fore / Aft Neck Criterion (%)	48.3 / 16.6	15.4 / 65.9	10.4 / 110	12.1 / 86.8		
	Tension / Compression Neck Criterion (%)	3.5 / 93.1	95.1 / 2.7	91.8 / 9.6	71.6 / 4.6		
	Chest 3ms max (g)	18.1	19	21.2	15.5		11.7
	Chest Deflection (mm)			3.7			
	Maximum Left Femur Load (kN) (compression +ve)	3.1	3.4	4.7	4.9		3.5
	Maximum Right Femur Load (kN) (compression +ve)	3.8	4.0	3.7	3.5		3.1
Pulse Details	Maximum Deceleration (g)	11.9	11.9	12.0	11.9	11.2	12.1
	Maximum Velocity (kph)	30.6	29.9	29.9	29.9	30.6	30.24
	Average Deceleration (g)	7.54	7.73	7.69	7.16	7.19	

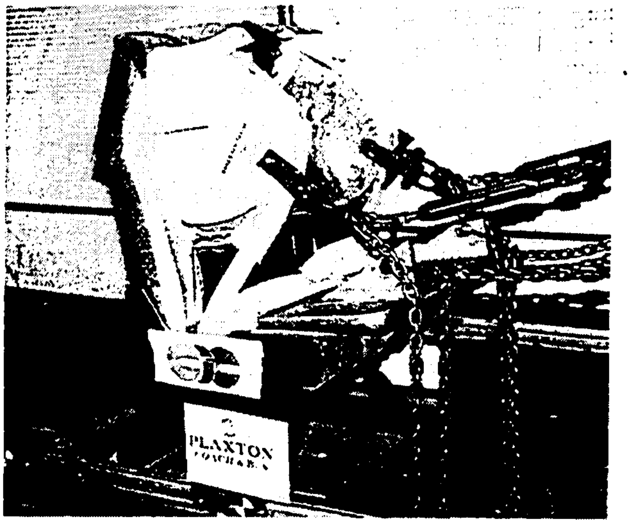

Figure 5 View after belt anchorage test: 76/115/EEC

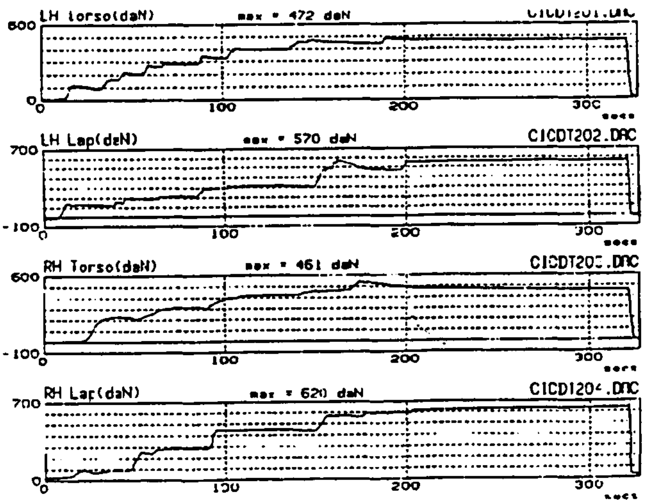

Figure 6 Load vs time during the belt anchorage test

unbelted dummies, fully loaded seat hit by lap-belted dummies

The seat also performed well when fully loaded seat was hit by lap-belted 95 %ile male and 5 %ile female dummy, as well as when seat spacing of 650 mm was applied between an empty seat and seat carrying lap-belted 50 %ile dummies.

The seat also passed the 76/115/EEC Directive on seat belt anchorage loads, although this requirement was not seen as essential after the whole system seat/ belt/ anchorages met the dynamic test conditions.

The seat thus passed all the conditions of the *current proposal* for the new EEC Draft Directives on seats and seatbelts (50 %ile unbelted or lap-belted dummies hitting an empty seat in front), but also all the conditions of the *combined loading* where unbelted or lap belted dummies hit the seat in front which also carries 3-point belted dummies.

The seat is no bigger than a typical European current production seat and is made using conventional materials and production methods. The mass of the new twin seat of 36.3 kg compares well with some in *current* use. The seat was tested with standard mounting rails on the coach body. All these provide a sound basis for the production development of a commercial seat that would be suitable for *all seating positions*, with the following possibilities :

(a) high number of units, hence lower unit cost;

(b) possibility to offer lap or 3-point belts as options on the same seat, perhaps as a retrofit.

The new seat also provides a sound basis for resolving all of the main outstanding issues regarding the safety of coach seats (i.e. protection of *all* passengers, 'mixed mode operation', etc.).

ACKNOWLEDGEMENTS

The Project was initiated and funded by the *Department of Transport* (DoT), London, Vehicle Standards and Engineering Division (contact officer : D Macdonald) and completed by:

- *Cranfield Impact Centre Ltd* : Dr D Kecman, N Randell, P Popely, R Jones, R Butler and support staff provided project management, background studies, selection of the overall safety concept and the main components, numerical and safety design analysis, component tests, reporting ;
- *Plaxton Coach and Bus*: A J Dutton, C A Jones (supported by *A W Chapman Ltd*) and other staff provided design consultations, 3-point belt geometry, blending the safety features with the other seat/ production requirements, R80 compatible auxiliary seats for dynamic tests and trims for the new seats ;
- *Millbrook Proving Ground* : J Wilkinson, G Williams, E Wilkinson, P Farrow and support staff carried out all dynamic HyGe sled tests and the static seat and belt anchorage test.

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THE ABILITY OF 3 POINT SAFETY BELTS TO RESTRAIN OCCUPANTS IN ROLLOVER CRASHES

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ABSTRACT

Three point safety belts are intended to restrain front seat occupants in motor vehicle crashes. Their purpose is to reduce the severity of occupant collisions with the interior of a vehicle and thus to reduce occupant injury. Manufacturers and the government test occupant protection in frontal collisions both for compliance with federal requirements and under a federal consumer information program. No consensus exists for a test of the ability of seat belts to prevent harmful contact with the roof and roof structure of vehicles. This paper describes a simple test procedure and provides data from tests of some common production safety belt systems. These tests demonstrate that most of the production belts place the head and neck in potentially injurious positions in a rollover. These tests also show that simple geometric improvements could provide substantial head and neck protection in rollover crashes.

INTRODUCTION

Head and neck injuries in motor vehicle accidents account for 36.8 percent of the most severe injuries and 50.6 percent of the HARM. The rollover crash, as an accident mode, places an occupant's head and neck in a position of potential injury. Rollover crashes, while occurring in only 2.1 percent of accidents, are accountable for 18 percent of the fatalities. These facts are surprising considering rollovers are the least severe mode of accident.

Vehicle deceleration rates in rollovers are typically low compared with frontal or side crashes. Therefore, it is not unreasonable to believe that occupant protection in rollovers is feasible. Improving belt usage and belt technology has the potential to reduce 86.7 percent of the HARM caused in rollover crashes. By reducing the severity of head impacts with the roof of a vehicle, manufacturers could substantially enhance occupant protection in this mode. Potential improvements in vehicles include increasing and maintaining the relative distance between the restrained occupant and the contact surface, and improving the energy absorption characteristics of the contact surfaces. New federal requirements for energy absorption may have addressed the latter problem in the roofs of new vehicles. The retention of the occupant survival space is the duty of the safety belts and the roof structure.

Part of the reason that rollovers remain such a deadly accident mode is that cars are not tested in this mode; therefore, insight and improvements remain untested. Dolly rollover tests remain part of FMVSS 208, yet it is rarely carried out. This paper will discuss a simple apparatus and protocol to test safety belt effectiveness in a rollover environment. It shows that typically contemporary safety belts provide almost no vertical restraint, which is particularly damaging when coupled with the small amount of head room and the propensity for roof crush in many production vehicles. When analyzing belt system performance in rollover accidents, a simple and inexpensive design change can have significant effects.

TEST SET UP

The test subjects were three volunteers selected to approximate the 5th percentile female, 50th percentile male and the 95th percentile male.

Physical measurements of all three subjects consisted of subject height, seated height and weight (Table 1). Subject height was taken while subject was standing with heels against a wall, with a level resting on top of the head. Seated height measurements were made by positioning the subject on top of a rigid surface with the lower legs hanging naturally over the edge of the surface.

Table 1.
Test Subject Measurements

	Small Female	Medium Male	Large Male
Height	147.3 cm	174 cm	189.9 cm
Weight	53.1 kg	74.8 kg	106.6 kg
Seated Height	78.7 cm	88.0 cm	99.1 cm

The lower spine and buttocks were placed against a vertical wall, and a level was again placed on top of the head. Subject weights were taken by having the subjects stand on a weight scale. Body fat measurements were not taken for any of the subjects.

Subject vertical excursion measurements were taken with the aid of a helmet mounted measuring device, reference line and video camera. The measuring device was a 3/4" diameter tube marked with inch tape, attached to the top of the helmet such that it was positioned perpendicular to the roll axis. When the helmeted occupant was seated in the test apparatus, the measuring device was viewed by the video camera against the background of the reference line. To avoid optical parallax, the center of the camera lens and the reference line were placed at the same height. While the occupant was seated upright, the arbitrary position of measuring reference line superposition was noted. When the occupant was fully inverted, a similar superposition measurement was taken. The absolute difference between these two numbers was taken to be the maximum occupant vertical excursion.

The testing apparatus consisted of a 'buck' that included a driver's seat and restraint system. For each production vehicle tested, the matching production seat was used with the corresponding belts. The buck, which approximates a driver's compartment, was mounted on a spit fixture and allowed to rotate about the longitudinal center of the buck. This apparatus is shown in Figures 1-4. Four video cameras were placed as follows: camera 1 focused on the stock retractor; camera 2 was mounted level with and looking aft towards the reference sighting line; camera 3 focused on the stopwatch and angle finder; camera 4 was mounted approximately 13 feet from the datum on the passenger side of the fixture to capture an overall side view. Each camera's output was fed to a video quad mixer which compressed each output and allowed simultaneous display of all four views on the monitor. The output of the quad mixer was also fed to a VHS video recorder.

Figure 1. Front view of test apparatus with large male test subject.

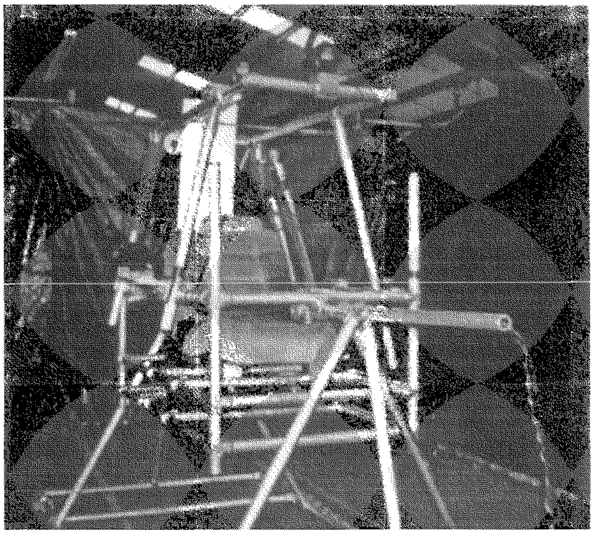

Figure 2. Oblique view of test apparatus.

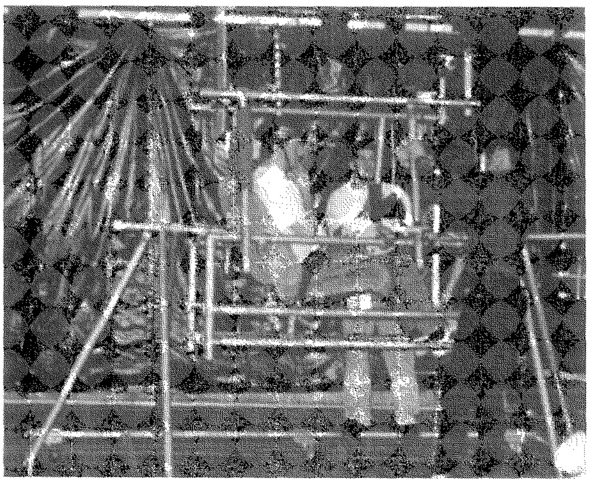
Figure 3. Side view of test apparatus with large male test subject.

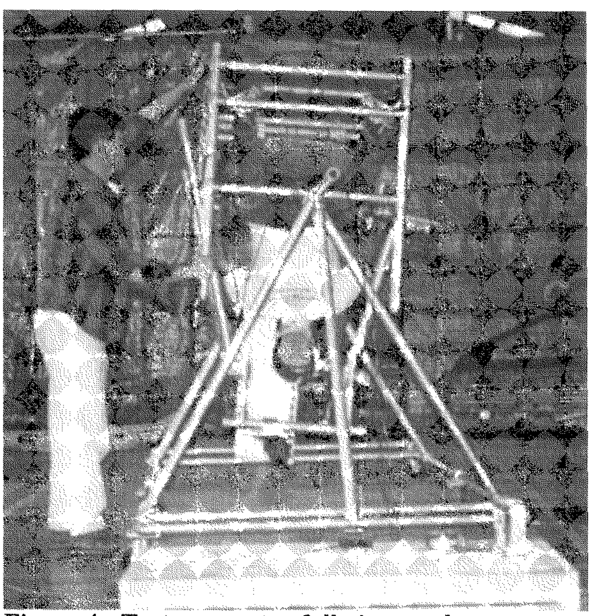
Figure 4. Test apparatus fully inverted.

The buck was configured to match three vehicles that are common to the population of cars on the road today. In the interest of simplicity, configuration was limited to seat, roof and seat belt relations. The seat position was changed for each of the three occupants. For the 5th percentile female the seat was set to the full forward position. For the 50th percentile male the seat was positioned half way between full forward and full rear. To accommodate the 95th percentile male the seat was adjusted to full rear. The belts tested were as follows:

- Belt A - From a mid 1980s 4 door sedan. It has a single retractor and a continuous loop belt with a pass through latching plate;
- Belt B - From a late 1980s 4 door sedan that complies with the automatic crash protection requirements of FMVSS 208 using three point, door-mounted belts. It has two retractors and a stitched latch plate;
- Belt C - From a mid 1980s sport utility vehicle. It has two retractors and a stitched latch plate;

The above production belts were then compared to a restraint system which will be referred to as the improved geometry belt. It was a standard self-tightening retractorless lap belt. The anchor configurations were in compliance with the *SAE Vehicle Occupant Restraint Systems and Components Standards Manual*. This system, in an attempt to eliminate the seat as a variable, was used with each production vehicle buck. In vehicles B and C, this lap belt system was placed 25.4 cm apart, centered on the seat back, at the rear intersection of the seat back and seat cushion and routed forward through that slot. Vehicle A was configured similarly, except the separation was increased from 25.4 cm to 31.75 cm due to the seat construction.

PROCEDURE

The test matrix consisted of three occupants in three vehicles. Each occupant was subjected to three static and three dynamic test runs with production belts (belts A, B or C), as well as three static and three dynamic test runs with the improved geometry belt. All belts were worn comfortable by the volunteers without additional slack or tensioning induced. These test runs were averaged and summarized in Table 3. In order to negate any retractor effects, the production belt retractors were pre-locked during the static runs. During dynamic runs the production retractors were allowed to lock-up in their standard manner. Each test run began with the occupant seated upright and secured with the appropriate belt system. A reading was taken from camera 2 to determine a starting height. The buck was then rotated passenger side leading at an approximate roll rate of 100 degrees per second. A reading was again taken when the occupant was fully inverted. The absolute difference between these readings was the maximum vertical excursion of the occupant. Between each test run, the occupant removed and replaced the belt system. Adjustments were made to remove any excess slack, but no tension was added to that of the retractor. The occupant remained seated throughout each vehicle matrix.

RESULTS

The manual lap belt system with experimental anchor locations decreased excursion by as much as 75 percent. This belt system was equally effective on the small female and the medium male. The system showed smaller reductions for the large male. The best of the production belts used only limited the three occupants to an average of 13.9 cm of vertical excursion. The door mounted belt system allowed the large male to move vertically up to 26.7 cm.

Table 2.
Static Occupant Vertical Excursions (Centimeters)

	Sm. Female	M. Male	L. Male
Production Belt A	11.4	17.4	19.9
Improved Geometry	4.7	8.5	16.1
Reduction	**6.8**	**8.9**	**3.8**
Production Belt B	18.6	17.8	23.3
Improved Geometry	8.5	6.8	15.2
Reduction	**10.2**	**11.0**	**8.0**
Production Belt C	11.9	15.7	16.9
Improved Geometry	5.1	5.1	14.8
Reduction	**6.8**	**10.6**	**2.1**

Table 3.
Dynamic Occupant Vertical Excursions (Centimeters)

	Sm. Female	M. Male	L. Male
Production Belt A	9.3	11.4	21.6
Improved Geometry	7.2	7.2	17.8
Reduction	**2.1**	**4.2**	**3.8**
Production Belt B	20.3	20.7	25.8
Improved Geometry	8.5	7.2	19.5
Reduction	**11.9**	**13.5**	**6.4**
Production Belt C	20.7	16.5	23.3
Improved Geometry	5.1	6.4	13.5
Reduction	**15.7**	**10.2**	**9.7**

DISCUSSION

The reductions for the small female and medium male generally ranged between 50-60 percent. The results for the large male were not as pronounced, which can be attributable to several things. As the large male erects during inversion, he has the greatest seated height and therefore creates the most excursion while erecting (6 cm). Also, this subject has the greatest soft tissue effects of the three volunteers. Other contributing effects are greater belt elongation and belt-to-cushion interaction due to the larger mass.

Table 4.
Front Seat Headroom in Typical 1993 Cars and Trucks

Vehicle	Headroom	Head Clearance*
Acura Integra	98.3 cm	6.9 cm
BMW 750iL	97.3 cm	5.9 cm
Buick Regal	98.6 cm	7.2 cm
Cadillac DeVille	99.8 cm	8.4 cm
Chevrolet Cavalier	98.0 cm	6.6 cm
Dodge Colt	97.3 cm	5.9 cm
Ford Mustang GT	94.0 cm	2.6 cm
Lincoln Town Car	99.1 cm	7.7 cm
Mercedes 560 SEL	94.7 cm	3.3 cm
Mercury Sable	97.3 cm	5.9 cm
Ford Ranger Pickup	96.5 cm	5.1 cm
GMC 1500 Pickup	101.6 cm	10.2 cm
GMC S-15 Pickup	96.5 cm	5.4 cm

***BASED ON NOMINAL SEATED HEIGHT OF 91.4 cm**
Automotive News 1993

When examining the typical front seat headroom in Table 4 and the vertical excursions seen in tables 2 and 3, it is easy to see that production belts leave many occupants effectively unrestrained from contacting the vehicle's roof and upper structure. When in a rollover, an occupant is essentially unrestrained since they must move 20 cm before the belts apply restraining forces, yet the roof may only be 5-10 cm away. In a majority of cases the volunteers tested in production belts would travel unabated towards roof, placing the head/neck complex in a potentially injurious position. This potential for injury is shown even without taking into account several other real world factors. The reduction of roof clearance due to roof crush, vertical decelerations due to impact and higher centrifugal forces will increase the chances of injury further. It is our intent to explore these things with the computer simulation program, DYNAMAN. Preliminary results show good correlation between the simulation model and the tests already completed. Further development of these computer models will explore these other effects not easily tested in the real world. Once validation has occurred, research will continue into the effects of different roll rates, vertical impulses through the fixture and roof impacts. Also, optimal belt arrangements, pretensioners, human soft tissue effects and shoulder belt influence will be analyzed.

Shoulder belts have several positive influences on occupant kinematics during a rollover. During the rollover portion of an accident, occupants tend to erect due to centrifugal force. The shoulder belt, if properly placed and functioning, can maintain an occupant in a reclined position as if normally seated. Shoulder belts, similar to those already in production on integrated seats, have the potential to achieve this goal. Assuming that the recline angle of seat and occupant are on the order of 20 degrees, the large male, for example, can have vertical excursions of almost 6 cm by merely erecting from the recline. For most occupants a shoulder/lap belt system, that merely prevents occupant erection, can provide 2.5-5 cm excursion reduction than a lap belt only system. It can also provide restraint of the upper torso in all directions resulting in lower relative velocities and displacements between the head/neck complex and the roof. This should provide further reduction in injury potential. The testing of these hypotheses will be analyzed in future fixture tests and simulations.

Many production cars sold in Europe have pretensioners on front outboard belts. However, many of those could be triggered by rollover sensors to provide increased occupant restraint. Roll sensors are a simple technology that have been successfully incorporated into production convertibles to actuate roll bars for rollover protection. Pretensioning can reduce the slack inherent in a belt system, while simultaneously drawing the occupant into the seat away from upper contact surfaces.

The excursion attributable to the rotation of the belts to vertical, as the occupant loads the belt, is another area for potential reduction in vertical excursion. The improved geometry already tested achieves a partial reduction in this effect by reducing the rotation radius of the belt; therefore, it reduces the vertical excursion required to align the belt into a loading configuration. By pre-aligning the lap belt angle, the belt would apply its restraining load early in the accident event, reducing occupant vertical motion in a rollover. The effects of increasing lap belt angle with respect to the horizontal should be considered in the overall occupant protection system for all crash modes.

CONCLUSIONS

Restrained occupant head impacts with the roof of a vehicle during a rollover are a major cause of serious injury. The tests described herein show that the performance of many production belt systems permit far too much vertical excursion in a roll event, leaving the occupant virtually unrestrained. If something as simple as an anchor location change can provide such dramatic reductions, more advanced state of the art techniques could nearly eliminate vertical excursion. Rollover actuated pretensioners, web grabbers and geometry improvements are techniques that could limit vertical excursion to 2-4 cm or less.

These tests suggest that a standardized, practical, repeatable test procedure is feasible. They also show that limiting the potential for vertical excursion in the design of safety belts is both possible and well within the state of the art. More advanced methods exist that could further lower occupant excursion towards the roof.

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BODY ENGINEERING CONSIDERATIONS TO IMPROVE OCCUPANT SAFETY IN MINIBUSES AND COACHES

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Paper Number 96-S5-O-13

ABSTRACT

Modern passenger cars offer a high level of occupant protection compared to minibuses and coaches. Whereas a typical car is now equipped with lap and diagonal seat belts and airbags for the front seat passengers, the majority of minibuses and coaches in use have either no restraint system or simple lap-only belts that offer no upper torso restraint in frontal impacts.

Following several tragic accidents involving minibuses, raised awareness has led to a demand for seat belts to be fitted to existing vehicles. This has resulted in a number of companies providing a service to retrofit seat belts, or fit new seats with integral belts.

Bolting seat belts to an existing seat is generally unsatisfactory because the seat and its mountings were not originally designed to withstand the loads imposed by the occupant under impact conditions. Thus the seat either detaches from the floor structure or collapses onto the occupant. Specially designed structural seats with integral belts provide the maximum occupant safety, but the vehicle floor generally requires significant strengthening to successfully transfer seat belt loads into the structure.

The majority of companies that convert vehicles mount these replacement seats directly to the sheet metal floor. Research work at MIRA has confirmed that this method is insufficiently strong to prevent the seats from tearing out of the floor when subjected to the quasi-static seat belt anchorage test, ECE Regulation 14 (M2 vehicle loads).

To deal with this installation problem, MIRA has developed a system which enables the fitting of replacement seats with integral belts into minibuses. An 'under floor' solution was adopted, to prevent the loss of headroom and weight penalty incurred with alternative 'over floor' framework designs. The design, development and subsequent system validation process is described, illustrating the methods adopted for transferring occupant belt loads through the sheet metal floor, into the body structure.

INTRODUCTION

Recent improvements in passenger car safety have highlighted the inadequacy of safety solutions for other categories of vehicles. In particular, most minibuses and coaches either have no restraints fitted or simple lap belts that offer limited performance. Tragic accidents involving minibuses have attracted much media attention and there is growing public awareness and concern for safety, especially for school children where this form of transport is widely used. The UK Government has acted unilaterally in requiring the fitment and use of seat belts in minibuses and there is a growing demand worldwide for improvements in occupant protection for these vehicles and coaches.

This has lead to a number of investigations into the installation of seats and belts that have evaluated opportunities for enhanced occupant protection (Ref. 1).

Technical solutions for the structural strength of the seat and floor, together with reasonable seat belt ergonomics are achievable without great difficulty, but the major challenge lies in producing a vehicle that offers a high level of safety whilst keeping the vehicle weight and cost within reasonable limits.

VEHICLE CONSTRUCTION METHODS

Many coaches and buses use a wooden floor that is attached to steel cross-members. These cross-members mount onto the vehicle chassis and also support the body sides. Seats are mounted through the floor or on tracks that are fixed to the floor. Although this design allows for a quick and simple process in assembling vehicles, generally it renders the seat mounting insufficiently strong to withstand the loads that seat belts exert through a seat fitted with integral belts.

Minibuses are generally light vans that have been modified to carry passengers. This usually involves cutting windows into the body sides, bolting seats into what was the cargo area and trimming the rest of the interior for occupancy. Converting vans into minibuses is often carried out as a post-build activity by a specialist convertor, because the numbers involved do not warrant factory built minibuses. As with coaches and buses, the standard floors generally are not strong enough to withstand seat belt loadings without reinforcement.

DESIGN PHILOSOPHIES FOR OCCUPANT PROTECTION

The two most significant crash scenarios for minibuses and coaches are frontal impact and roll-over (Ref 2, 3). Frontal impact results in occupants impacting the rear face of the seat in front and roll-overs often result in ejection of passengers from the vehicle. Once ejected, the likelihood of severe injury is increased due to being crushed by the vehicle or striking hard objects. The adoption of seat belts is thus a primary requirement for the safety of minibus and coach passengers.

It has previously been proposed to install rearward facing seats in school buses (Ref 4), which offer enhanced occupant protection, since the seat back provides support for passengers upper torso without the need for lap-and-diagonal (3-point) seat belts. However, lap-only (2-point) belts would be required to prevent ejection. The principal problem with rearward facing seats is that they are unpopular with passengers.

Airbags are a common safety feature on private cars but the adoption of these devices on minibuses and coaches is problematic. The cost and weight associated with airbags would significantly increase the price of these vehicles if an airbag was provided for every occupant. Also, for larger vehicles, occupant acceleration levels are generally significantly lower in frontal accidents because of the higher overall mass compared to passenger cars. Since airbags provide no protection in roll-overs, there is currently very little interest in fitting them into minibuses and coaches. The cost of replacing a complete set of inflators on an accident damaged coach is a further incentive for designers to develop alternative methods for optimising passenger safety.

The design of seats is highly influential on occupant injuries. For many years seats have been designed to collapse when struck by occupants, thus reducing contact loadings. This solution works well if there are no requirements for the seat other than to support the weight of the passenger. However, leaving occupants unrestrained and relying on hitting weak seats to minimise injuries is no longer acceptable and does not address the problem of ejection in a roll-over.

RESEARCH ACTIVITIES

In preparation for European legislation, research work has been conducted to assess various seat belt systems (Ref 1). 2-point and 3-point belt systems were considered and the influence of seat spacing. It was found that a 2-point belt could adequately restrain an occupant and the seat in front could be used as an impact surface for the head whilst the upper torso rotated forwards. By optimisation of the seat stiffness characteristics and the seat spacing, 2-point belts can provide a reasonable level of protection for a 50th percentile male. Complications can arise when children and large adults are considered, as the optimum seat spacing may be different for these cases. Also, an incorrectly fitted 2-point belt, for example one left loose, can result in abdominal injuries as the occupant is suddenly restrained once the slack in the belt has been taken up and the occupant's body is incorrectly positioned for the appropriate restraint around the pelvis.

Occupant protection studies at the Motor Industry Research Association, UK (MIRA) have focused on the use of 3-point seat belts that are integrated into the seats. Research has indicated a high demand for 3-point seat belts on replacement minibuses since 1994, although there has been no legislation requiring seat belts for such vehicles in the UK. An initial study was conducted at MIRA to establish current practice for the installation of seat belts in both new and used minibuses. The lack of any form of regulation had resulted in seat belt installations to a wide range of standards. The only common factor was that the majority of systems had not benefited from any form of testing and development. The following approaches were found to be common:-

- Seat belts attached to existing passenger seats
- Seat belts attached to the vehicle floor and passing over the original seats
- Additional tubular structures that provided seat belt anchorage points
- Replacement seats that contained integral seat belts installed in place of the original items.

The lack of quality standards in the evaluation of seat belt installations led MIRA to producing a report that outlined recommendations for the installation of seat belts in minibuses and coaches (Ref 5). The test standard recommended was the adoption of ECE 14.03 for M2 (minibuses) or M3 (coaches) vehicles. Also, advice was given on the practical aspects, outlining the correct engineering approach to the fitment of seat-belts. Different installation methods were listed and the advantages and disadvantages of each system were considered.

The recommended best practice was to fit replacement seats with integral 3-point seat belts.

DEVELOPMENT OF MINIBUS SEAT BELT SYSTEMS

In 1995 MIRA undertook a programme of work with a vehicle manufacturer to install seats and seat belts into new minibuses derived from vans, using the services of a number of authorised conversion specialists. There were three versions of the standard minibus with fixed seats, the fourth variant was specified to be wheelchair accessible. The concept of the project for the minibus variants is shown in Fig 1. The wheelchair accessible (mobility) minibus was to use proprietary aluminium tracking that would allow either seats or wheelchairs to be secured at any point along the rails using a proven clamping system.

Selection of Seats and Seat Belts

Seats would be chosen that had been developed by quasi-static load tests as described in ECE Regulation 14. Testing these seats on a rigid base did not ensure that they would perform adequately when fitted to a vehicle, but it provided a useful initial assessment of the integrity of the seat assembly. Two aspects were considered. Firstly, the strength of the seat was assessed; would it resist the loads without a major structural failure? Secondly, the geometrical aspects were assessed. This involved measuring pre- and post-test belt anchorage locations and comparing them with the requirements of the Regulation. The majority of seats sold with integral seat belts are untested and their performance under crash conditions was open to question. However, satisfactory seats were chosen from three principal UK manufacturers. All seat belts selected had been tested to ECE Regulation 16, which gave an acceptable level of confidence.

Seat Installation

The installation of seats into the van floor demanded certain criteria to be met:

- The resulting seat belt anchorage strength for the installed seats had to be sufficiently strong enough to meet the strength and geometrical requirements of ECE 14.
- Different seat layouts had to be provided, to cater for varying customer requirements
- Long and short wheelbase versions had to be accommodated
- Minimum additional weight
- Acceptable ergonomic layout
- Simple conversion that did not require welding to the vehicle (that would impair the vehicle's long term resistance to corrosion)
- A wheelchair accessible version was required, using a proprietary tracking system

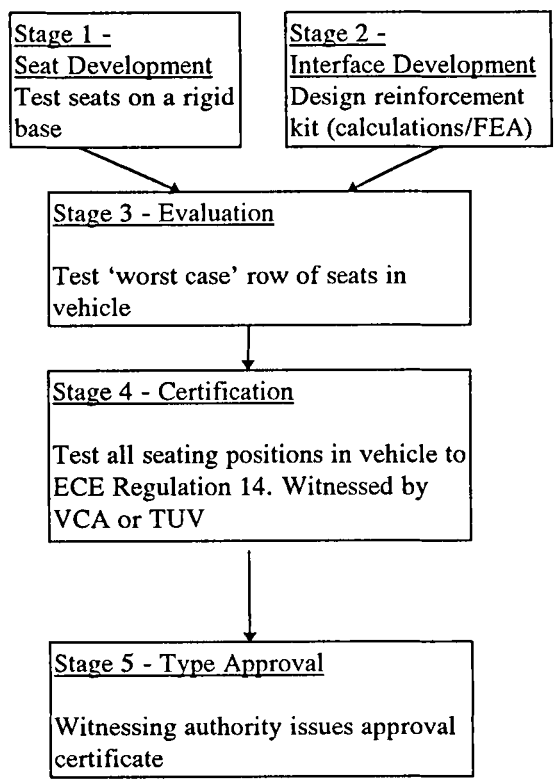

Figure 1 Development Concept

Re-engineering of Floor

Van floors are often constructed of longitudinally swaged thin sheet steel, spot welded to the main chassis rails and cross-members. The majority of minibus conversion companies rely on bolting the seats through the steel floor, spreading the load with large washers or plates. The loads experienced on seat mountings are illustrated in Fig 2 for a double seat, attached to the floor through two legs. The load case was taken from the European Regulation 14, for M2 vehicles. Although minibuses in the UK and Europe do not need to meet this (or any other) seat belt anchorage strength requirement, it was considered to be an appropriate design target. To re-engineer the floor, the initial step taken was to establish the strength and stiffness that was required to resist tearing the seats from the floor or cause excess deformation, when subjected to the test load.

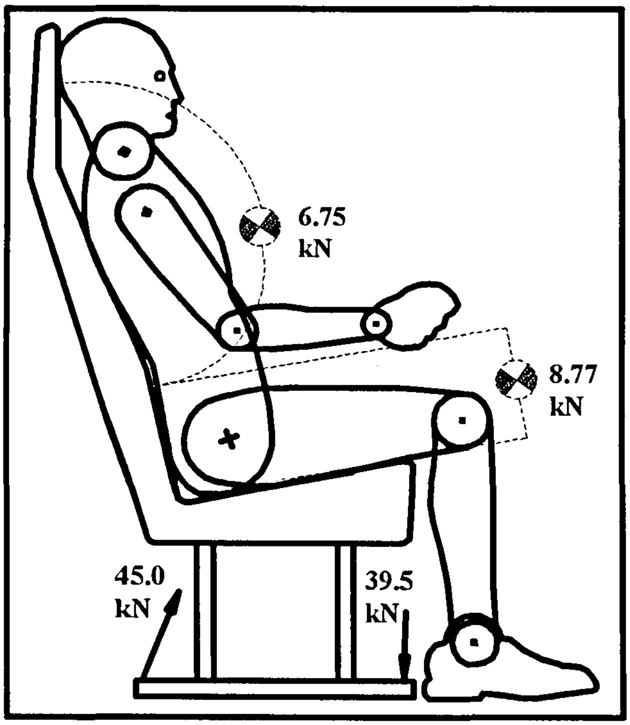

Figure 2 Loads on Seat and Floor Mountings

A simple method of providing adequate strength is to construct a framework of square section tube that provides pick-up points for all the seat mountings. This framework is placed over the existing floor and is bolted through the floor using straps to clamp around the main chassis rails. As well as raising the centre of gravity of the seats the disadvantage of this method is that the weight of the framework, typically 120 to 180 kg, results in a loss of seating capacity because the maximum vehicle weight is limited by legal considerations. To produce a weight optimised solution, it was necessary to make the maximum use of the existing structure. This was achieved by using new, additional under-floor structural members that were attached to the main vehicle chassis rails and cross-members.

Seat Layout

Four minibus versions were to be developed for this range of vehicles. To reduce the number of variations certain 'worst case' seat layouts were chosen for development. 'Worst case' was determined by assessing which floor areas were given the minimum level of support by the main chassis rails. Additionally, areas that featured a spot welded seam where two floor sections were joined together were identified as being prone to tearing. The other areas of the floor were left to be proven in final validation tests.

The spacing of minibus seats is partly governed by the minimum required seat pitch to give sufficient leg room. However, for this project the seats were placed such that the optimum load path could be achieved between the seat mountings and the main vehicle structure (Fig 3). In particular, it was necessary to avoid bolting into the closed box created by the chassis components, since the drilling of any large holes in these key members could weaken the structure.

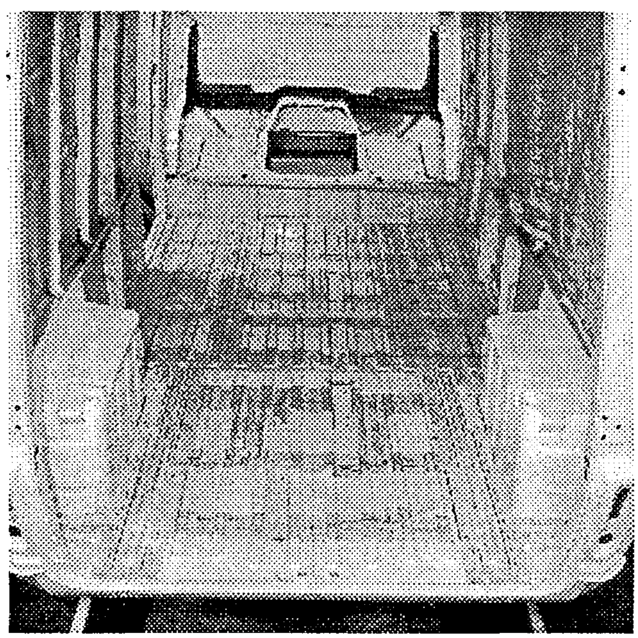

Figure 3 Position of Seat Mountings

Structural Analysis

Basic calculations provided an initial indication of structural requirements but as they indicated that the failure mode may involve buckling and tearing of the underfloor, finite element analysis was undertaken to account for the non-linear behaviour after the onset of plastic deformation. To minimise weight, plastic deformation of the seat mounting system was acceptable but a progressive failure was required, to prevent a sudden fracture and loss of load carrying capability. The non-linear dynamic finite element code OASYS DYNA-3D was used to analyse initial design proposals (Fig 4). Optimum section sizes, material specification and thickness were predicted and the solution that provided the best performance for the least weight was chosen for evaluation.

The vehicle manufacturer did not want the additional structural members to be welded in place because this process would disturb the PVC anti-rust coating on the underbody. Also, control of welding quality is difficult with fully built up vehicles and a quality assured design solution was essential. Zinc coated steel 'pop' rivets were selected to attach the new sections to the main structure working in a shear mode. These fasteners provided adequate strength with no corrosion problems and allowed rapid assembly. The front mounting of each seat is loaded downwards under test and the design of the floor reinforcements was optimised to account for this load case. To prevent fatigue failure around the seat mounting holes, load spreading plates were specified for the top side of the floor, bonded in place with polyurethane adhesive to provide a good shear connection.

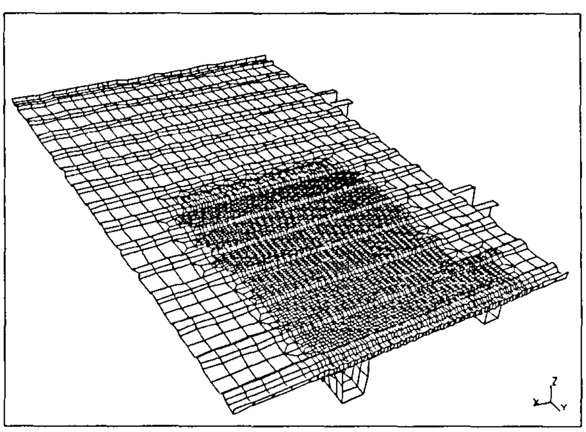

Figure 4 FE Model of Floor Structure

Wheelchair Accessible Minibus

The aluminium tracking specified for mobility minibuses is generally mounted directly to van floors using M6 set screws and large diameter washers under the floor to provide load spreading. Structural analysis of the track and floor indicated that under test conditions the floor would tear around the fixing holes. The aluminium section did not have sufficient bending stiffness to resist the anchorage loadings, thus allowing virtually all of the tensile force in the rear mount to be transmitted to a single set screw, for many of the clamp fitting positions. A further concern was that it was not possible to fit the tracking directly to the floor without having some of the fixing screws dropping into closed sections. When this occurs, general minibus convertor practice is to secure the rail with a 'pop' rivet. However, this solution is not suitable for a correctly engineered conversion, since it provides insufficient strength in this particular application. To overcome these concerns, larger, high specification attachment screws were specified and a thin wall box section tubular structure was used as a mounting surface for the proprietary tracking. The combined stiffness of the tracking and box section provided the appropriate level of stiffness and the new screws prevented failure of the attachment of the tracking.

Development Process

To validate the finite element model and prove the design of the re-engineered floor a range of development tests were conducted on prototype parts. Rather than using a new body-in white (BIW) as a basis for the tests, a floor 'mock up' was constructed. The mock up provided a geometrical representation of the floor and chassis rail layout and allowed multiple tests, so long as the floor was repaired between tests.

The problems that can occur when simple plates are used to spread the loads is illustrated in Fig 5. The fixing nut extrudes through the washer or load spreading plate and breaks free. Increasing the thickness of the plate to reduce this mode of failure can lead to problems, as the entire plate tends to act like a punch in a piercing operation. This results in the whole load spreading plate cutting a large hole through the minibus floor.

The form of the kit varied significantly from one area of the floor to another (Figs 6 and 7). This was because in well supported areas adjacent to a structural member, a small fixing was adequate. However, for the first two rows of seats, steel box sections were needed to stiffen the vehicle and provide adequate strength. The finite element analysis model indicated the areas that required a higher level of reinforcement.

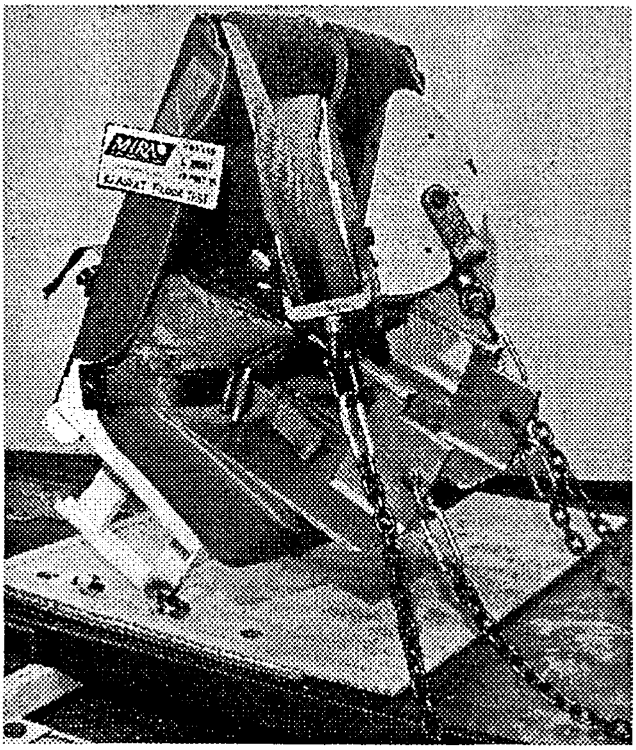

Figure 5 Effect of Lightly Reinforced Floor

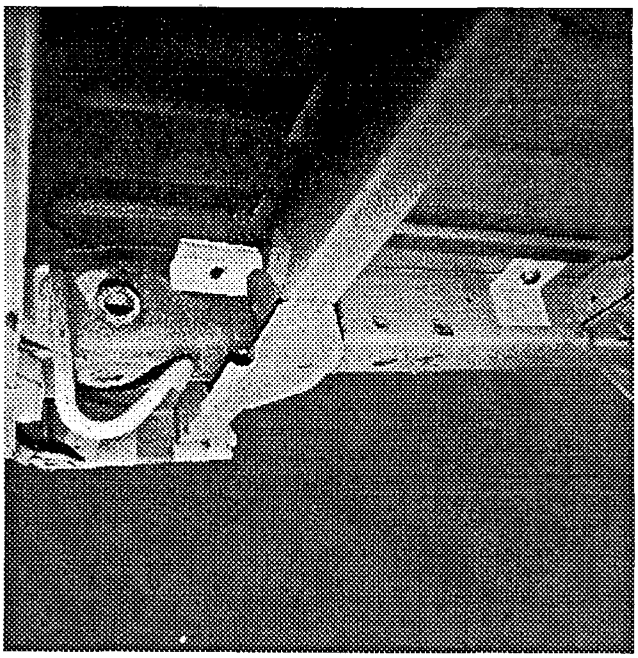

Figure 6 Small Reinforcement Plates

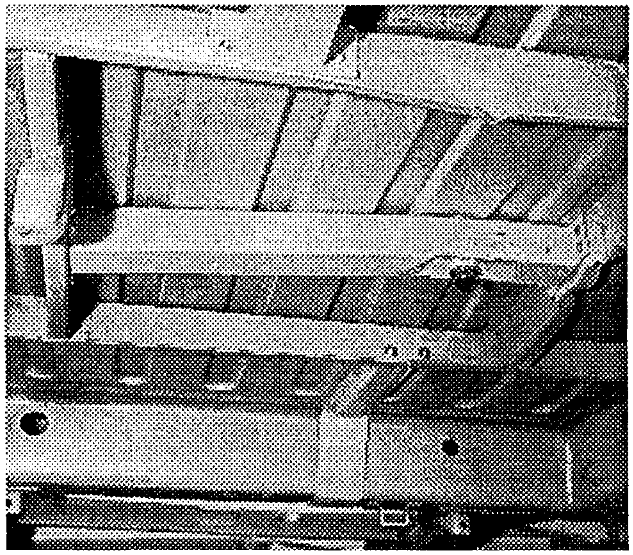

Figure 7 Box Section Reinforcements

Using these techniques the entire kit of parts needed to install seats in a 17 seat minibus weighed 45 kg, representing a weight reduction of over 60% compared to other proven conversion kits.

Tests on the mobility version (Fig 8) proved that the proposed mounting system resisted the seat-belt anchorage loads without failure, regardless of the position of the anchorage clamp along the rail.

Validation

To fully validate the seat-belt anchorage systems for the four minibus variants, tests were carried out on BIWs with fully representative re-engineered floors and seats. Fig 9 shows a typical test in progress. The tests confirmed the predictions of the finite element modelling and by using seats from three manufacturers, the system was proven for all three designs of seat. This gave the minibus manufacturer the range of options needed to satisfy the demands of different customers. During the tests, the under-body reinforcements suffered the initial stages of structural collapse but did not exhibit any catastrophic failures, such as tearing of the floor. This confirmed that the design was optimised, since if no plastic deformation had occurred this would have indicated that the design was over-engineered. A further benefit of having controlled structural deformation is that it produces a lower acceleration profile for occupants strapped into the seats during a frontal impact. However, it important that the level of deformation is controlled, to prevent belted occupants from striking an unoccupied seat in front. The tests confirmed that the seat rotation was adequately controlled to prevent such an occurrence

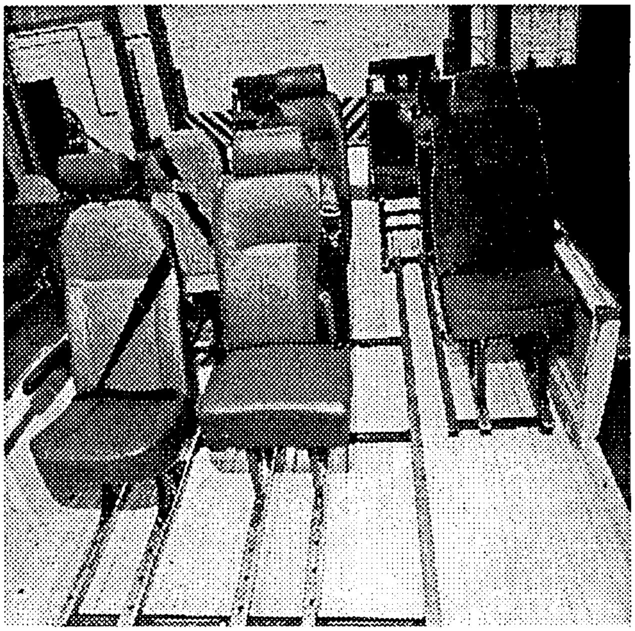

Figure 8 Mobility Vehicle Seating Layout

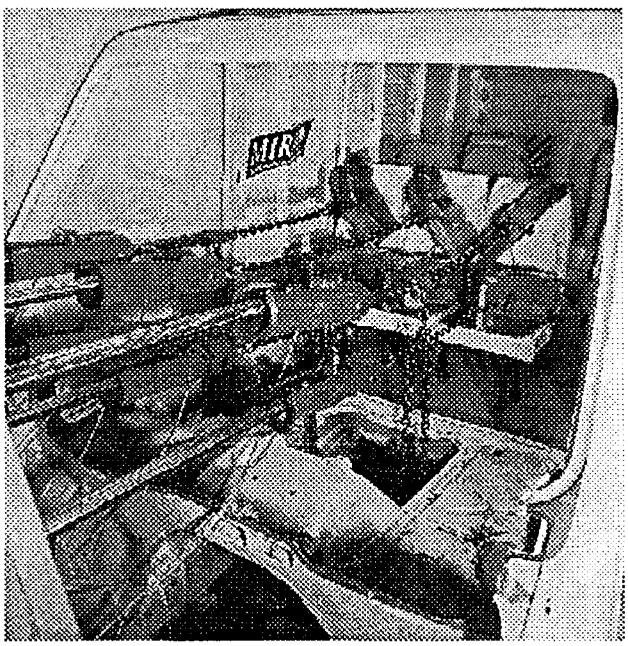

Figure 9 In-vehicle Seat Belt Anchorage Test

The tracking system for the wheelchair accessible (mobility) minibus was developed to transmit occupant loads into the vehicle structure. During crash conditions many wheelchairs are prone to collapse when high loads are transferred through them. Steps have been taken to improve wheelchair strength and the draft version of ISO/CD 10542-1 (Ref 6) describes certain requirements for wheelchair tie-downs and occupant restraint systems. Presently though, ECE Regulation 14.03 applies only to conventional seats.

The validation tests were carried out with the conventional removable seats in various 'worst case' locations specified by the Vehicle Certification Agency. The tests confirmed that the mobility vehicle also met all the requirements of ECE Regulation 14.03.

CONCLUSION

Many claims have been made in the minibus conversion industry regarding the installation of seat-belts into converted vans. Most agree that the optimum solution for passenger safety is to fit seats that have integral 3-point seat belts. Some convertors claim that it is not possible to fit suitably strong seat belt anchorages into van based minibuses. Conversion companies that do fit seat belts have usually chosen one of the following two methods. The first solution is to install seats onto a heavy framework mounted inside the vehicle, thus reducing the available head room and causing a weight problem. The secondly popular approach is to bolt the seats through the floor and use under-body plates to spread the load, using engineering judgement to specify the size, thickness and shape of the reinforcements. These designs have been shown to be generally inadequate.

The design and development work carried out at MIRA has proved that a low cost, relatively simple design solution is achievable for most vans of unitary construction. However, it has also demonstrated that it is necessary to conduct a thorough development programme, with extensive use of non-linear, dynamic finite element modelling, to avoid structural weaknesses and provide a solution that provides the optimum level of protection for passengers whilst keeping the overall vehicle weight to a minimum.

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IMPROVED VEHICLE DESIGN FOR THE PREVENTION OF SEVERE HEAD AND NECK INJURIES TO RESTRAINTED OCCUPANTS IN ROLLOVER ACCIDENTS

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Paper Number 96-S5-O-14

ABSTRACT

We analyzed the 1988 through 1992 NASS field accident data on rollovers and injuries to occupants in these crashes. The data show that more than 96 percent of all occupants in rollovers do *not* receive serious head or neck injuries. The authors discuss why most restrained occupants do not suffer serious head or neck injuries in rollovers and how that helps us understand the injuries that do occur. Based on these data, the authors further developed the rollover injury parameter "residual headroom" to identify the likelihood of severe head/face or neck injury and the vehicle design measures that can mitigate those injuries. A theory of rollover head and neck injury causation is proposed that is supported by all available evidence and observations. In particular, we will discuss how minor modifications of the roof structure and occupant protection systems of most contemporary passenger cars, light trucks, and vans can prevent severe injuries in rollovers.

INTRODUCTION

Single vehicle rollovers have more serious consequences than any other broad class of crashes involving light motor vehicles. Rollovers result in 23,900 severe or fatal injuries annually [Data Link, 1993]. At least 10,000 are to the head or neck. On the other hand, rollovers are potentially among the most benign accidents because the vehicle as a whole typically decelerates over a long distance and the gross accelerations of the occupants are low. The particularly dangerous aspects of rollovers come from whole or partial ejection of an occupant's body and from intrusion and local impacts in the roof area where the most vulnerable part of an occupant's body -- the head and neck – are located.

Automotive safety research is motivated by both prospective and retrospective factors. The first is to improve the safety of the motoring public and is conducted in support of government regulation, industry product improvement, and consumer information. The second is to support litigation to determine liability of injuries in motor vehicle crashes. Within the research community, the latter is often considered suspect and potentially biased because of the major financial interests in the outcome. However, because of the stakes involved, it often undergoes much closer scrutiny than is typical in peer review.

In this paper, we discuss our personally and privately funded research, some of which was conducted in the course of consumer litigation, and the research conducted and supported by automobile manufacturers that has been used extensively in their defense. Our data files on rollover crashes show patterns of occupant contact and injury that are similar to estimates involving serious head and neck injury resulting from rollover from the National Accident Sampling System (NASS), which we used extensively in this study. We will also discuss what appears to be a fundamental flaw in previous research conducted in support of manufacturers' litigation position.

ROLLOVER CRASH DATA

By 1980 sufficient data were available to show that rollover accidents resulted in approximately 20% of the total harm from injury in passenger cars [Malliaris, 1982] and 46% of the total harm from injury in light trucks and vans (LTV's) [GM, 1979]. NHTSA has described rollover accidents as the most dangerous type for light duty vehicles as measured by the number of fatal and incapacitating injuries to the number of occupants involved in tow-away accidents. It estimated that in 1989 there were 137,600 (±13,000) passenger car rollovers and 75,600 (±18,000) LTV rollovers. Kahane

[1991] reported 15,000 serious to critical injuries and 4500 fatalities in passenger car rollovers involving no other vehicles.

Table 1 shows the estimated number of rollover occupants (1,900,000) and those known to have had head or neck injuries (680,000) based on NASS data for the five year period from 1988 through 1992. Each year about 140,000 (36%) of all occupants are known to have received head, face or neck injuries. Of all occupants in rollovers, more than 96 percent were not coded as having been seriously injured. Two percent received serious to fatal head or face injuries and 1.0 percent received serious to fatal neck injuries. It should be noted that NASS underestimates the number of fatalities in rollovers (compared to FARS) and hence may also underestimate the number of serious injuries as well. Neck fractures, fracture dislocations and cord involvement were included as serious regardless of the AIS assigned to them.

Table 1 - 1988 to 1992 NASS estimated head, face and neck rollover injuries

ALL OCCUPANTS (5 years)	**Number**	**%**
(National Sampling adjusted; selected)	1,900,000	100%
Known Uninjured	630,000	33%
No HFN injury or Unknown Injury	570,000	30%
Known HFN Injuries NOT Serious to Fatal	630,000	33%
Head and Face Injuries only - Serious to Fatal	33,000	2%
Neck Injuries only - Serious to Fatal	17,000	1%
Both Head, Face and Neck Injuries	3,600 .	2%

The roofs of virtually all cars and LTV's currently on U.S. roads comply with FMVSS 216, but this does not prevent roof crush in a rollover. The available headroom in an undamaged automobile is generally 10 to 15 cm. Ineffective safety belts permit about this amount of excursion if a vehicle is rotated 180 degrees. Thus, the heads of occupants using these belts are likely to contact a roof which has deformed a few centimeters during a rollover. The force of the contact is increased with the rate and extent of roof deformation and the ineffectiveness of the belt.

Tables 2 and 3 separate the total occupants into unrestrained and restrained injury categories. While

Table 2 is the NASS estimated head, face and neck injuries to unrestrained occupants.

	Number	%
UNRESTRAINED OCCUPANTS	980,000	52%
Known NOT injured	270,000	14%
No HFN injury or Injury Unknown	310,000	16%
Known HFN Injuries NOT Serious to Fatal	360,000	19%
Head and Face Injuries only - Serious to Fatal	27,000	1%
Neck Injuries only - Serious to Fatal	14,000	.5%
Both Head, Face and Neck Injuries	3,100	.2%

about half the occupants were restrained (900,000 out of 1,900,000), there were 4 times as many serious to fatal injuries among the unrestrained (41,000 compared to 9,500). There were twice as many head and face injuries as neck injuries in total (33,000 to 17,000), unrestrained (27,000 to 14,000) and restrained (6,100 to 3,400). Since serious head/face and neck injuries occur together so infrequently (0.2%), the implication is that they are probably independent events caused by different mechanisms.

Table 3 is the NASS estimated restrained occupant's head, face and neck injuries.

	Number	%
RESTRAINED OCCUPANTS	890,000	48%
Known NOT injured	360,000	19%
No HFN injury or Unknown Injury	260,000	14%
HFN Injuries NOT Serious to Fatal	260,000	14%
Head and Face Injuries only - Serious to Fatal	6,100	.5%
Neck Injuries only - Serious to Fatal	3,400	.2%
Both Head, Face and Neck Injuries	430	0%

CRASH RECONSTRUCTIONS

The authors have investigated, analyzed, and often modeled over 45 rollover crashes involving serious to fatal head and neck injuries to restrained occupants. This work was conducted as experts for plaintiffs in product liability litigation. These cases were selected based on two further criteria. First, there was sufficient evidence available from each crash to permit detailed analyses of how injuries occurred. Second, it appeared likely that the severity of the injuries could have been reduced by relatively simple, inexpensive changes in the design and construction of the vehicle involved.

The authors typically obtained crash data including damage measurements, occupant impact points in the vehicle interior, belt performance data, and scene information. They estimated the force deflection characteristics of occupant contact surfaces using body part profiles applied to areas identified as occupant contacts or by actually testing a similar part. The roof strength characteristics used are either typical or are obtained from manufacturers' data.

Finally, restraint use was verified. Restraints are not typically loaded more than a few hundred pounds in rollovers. However, a latch plate with plastic in the belt slot will often show an indication of belt loading in a rollover. The characteristics of occupant protection systems were found from tests of typical components and manufacturer test data.

Using the computer simulation models such as ATB and DYNAMAN, the authors in most cases determine the kinematics and dynamics of the vehicles and their occupants to show how the injuries occurred. These models show forces on the occupant's head and neck that were sufficient to produce the observed injuries.

To assess the effect of changing occupant protection parameters, changes reflecting modified strengths of the vehicles' roof structures, the performance of their restraint systems, and interior padding without changing any other conditions of the crash are investigated. We modified the parameters in the computer models to represent inexpensive, practical improvements in roof strength, padding, and restraints that could have been made in these vehicles. Running the computer models with representations of these vehicle improvements confirmed the degree to which these changes could have ameliorated the injuries in these crashes.

The previously reported conclusions from these case studies, aside from those related to ejection and partial ejection, were:

1. The evidence suggests that four vehicle design factors affect head and neck injury to restrained occupants in rollovers: roof strength, upper interior padding, restraint performance, and interior compartment geometry.

2. Head injury is associated primarily with a lack of energy absorbing materials in the upper interior.

3. Neck injury is principally a function of roof crush (inadequate roof strength) exacerbated by poor interior compartment geometry and restraints that permit excessive excursion.

Figures 1 and 2 are photographs showing an example of how roof crush results in serious neck injury. The van's roof intruded 35 cm as a consequence of impact between the roof A-pillar/header/roof rail and the ground. The roof damage resulted from a frontal pitched impact at the end of the first half of a single roll in which the speed of the vehicle changed by roughly by 2.5 m/sec. The structure is particularly weak at the connection between the B-pillar and the front roof rail. The 160 cm tall, 95 kg female occupant received neck injuries that resulted in quadriplegia. She had 20 cm of headroom before the crash, and the restraint system permitted 20 cm of vertical excursion. The woman shown in Figure 2 is a model who is the same size as the injured woman.

FURTHER FIELD ACCIDENT DATA

In 1985, the authors examined estimates from NASS based on 1982 and 1983 files. We found that the probability of critical injury was four times greater in cases where roof damage was coded with a collision deformation index >3 in the proximity of the injured occupants' seating position, compared with cases where the corresponding deformation was less in the proximity of the occupant.

In 1995, Rains and Kanianthra used 1988 through 1992 NASS data to determine characteristic residual headroom for restrained occupants in rollover crashes (Table 4). The pre-crush headroom was in the order of 15 cm (six inches) for injured and uninjured cases using their methodology. The residual headroom is the space between the estimated pre-crush position of the occupant's head and the roof following the rollover. Their results indicate that the difference in average post crush residual headroom of 4.8 cm for injured, and 10.7 cm for uninjured, is only 6 cm (2.5 inches). They also discuss the sensitivity of this difference.

Table 4 - (From Table 1. of Rains, Kanianthra) Pre-crush headroom, and post-crush headroom for weighted vehicle and occupant cases.

	Avg. Pre Crush Headroom (cm)	Avg. Post Crush Headroom (cm)	% Headroom Reduction
Injured Cases	14.5	4.8	69
Uninjured	15.8	10.7	31

In the present study we analyzed cases involving head, face and neck injuries to identify the average residual headroom of unrestrained and 3pt manual belt restrained occupants. We looked at residual headroom where injuries were AIS 1-2 as compared with AIS 3-6 (including AIS 2 neck fracture, fracture/dislocation and cord involvement). The average of the interval reported for intrusion was used as the estimate of crush at the occupant location. The raw data results are shown in Table 5.

For unrestrained occupants with injurious upper interior contact, there seems to be little difference (-3.3 to -5.3cm) in negative residual headroom between minor (AIS 1-2) and severe (AIS 3-6) injury. For restrained occupants, however, the difference is dramatic. For head, face and neck injuries of AIS 2 or less, the average residual headroom is around 3cm (1 inch). For more serious injuries, the average residual headroom is minus 13 cm. (-5 inches).

Table 5 - HEAD/FACE/NECK (HFN) INJURY VS RESIDUAL HEADROOM (RHdrm)*

	UNRESTRAINED OCCUPANTS			
	AIS 1-2 HFN		AIS 3-6 HFN	
	No.**	RHdrm	No.	RHdrm
HFN contact NOT with Upper Interior	312	5.0 cm	30	2.2 cm
HFN contact with Upper Interior only	257	-3.3cm	79	-5.3cm
No injurious HFN contact	455	-0.6 cm		

*Residual Headroom is the average space between the occupants head and roof after the rollover.
** Number (No.) of occupants in assumed 38" seat to roof space in all cars and trucks.

	RESTRAINED OCCUPANTS			
	AIS 1-2 HFN		AIS 3-6 HFN	
	No.**	RHdrm	No.	RHdrm
HFN contact NOT with Upper Interior	228	3.2cm	7	3.3 cm
HFN contact with Upper Interior only	237	2.5 cm	41	-13.2 cm
No injurious HFN contact	619	6.6cm		

*Residual Headroom is the average space between the occupants head and roof after the rollover.
** Number (No.) of occupants in assumed 38" seat to roof space in all cars and trucks.

FACTORS AFFECTING ROLLOVER INJURIES

There are six factors that affect injury in rollover crashes. They are the occurrence of the rollover in the first place, intrusion of the roof into the occupant compartment, the energy absorption characteristics of the interior of the roof, occupant restraint, packaging, and containment of the occupant. The following is a summary of these factors.

- Rollover stability: In order to roll, a vehicle must get into a position where it is susceptible (yaw instability or partial rotation about a longitudinal axis such as might occur on an embankment at the side of a road). It then must be susceptible to being tripped or to tipping when in such a position. Yaw instability is commonly referred to as an oversteering tendency. Locking of rear brakes before front brakes on hard application will also produce yaw instability. A crude measure of a vehicle's tendency to roll is the ratio of the height of center of gravity to its track width. It is more accurately measured by the maximum angle a vehicle can sustain when rotated about its longitudinal axis on a tilt table. Rear anti-lock brakes were adopted on light trucks and vans primarily to reduce rollovers.

- Roof Strength: Deformation of a vehicle roof puts occupants at risk of injury. In addition to a roof's resistance to impact forces, the geometry of the vehicle may affect the ability of a vehicle roof to resist the forces of a rollover. We define the roof vulnerability ratio (RVR) as the ratio of (1) the distance from a longitudinal line through the center of gravity to the side corner of the roof to (2) the distance from the same longitudinal line to the lower outside edges of the wheels. A higher RVR will result in excessive roof loading and will make roof deformation in a rollover more likely. The current quasi-static test of roof strength ensures only minimal roof strength. Vehicle roofs that meet this standard routinely collapse in rollovers at low equivalent drop heights.

- Interior energy absorbing materials: Energy absorbing material in the roof, roof rails, and pillars can reduce acceleration loads to the head. A recent amendment to FMVSS 201 is intended to provide a substantial reduction in head and face injuries from direct impact with the roof and roof structures.

- Safety belt performance: Safety belts can reduce the likelihood that an occupant's head will make harmful contact with the roof and roof rails if the roof does not collapse excessively over the occupant. A related paper being presented at this conference, the *Ability of 3 Point Safety Belts to Restrain Occupants in Rollover Crashes,* discusses some potential improvements in safety belt performance.

- Door and window integrity: The doors, rear hatches and windows can protect against ejection of unrestrained occupants and of the head and limbs of a restrained occupant. The particularly weak point in virtually all vehicles in the U.S. is the side windows which are made of tempered glass and which are commonly broken due to roof crush or occupant contact in rollovers leaving no impediment to partial ejection through the window openings.

- Occupant Packaging: Proximity of the occupant to contact surfaces or ejection paths need to be consistent with intrusion, restraints and potential portals.

THE BIOMECHANICS OF NECK INJURY

In 1977 Snyder, Foust and Bowman investigated head and neck injury by examining human free fall data in a "Study of Impact Tolerance through Free-Fall Investigation." They found that head injury was unlikely to occur, and neck injury did not occur, from drop heights of less than five feet, corresponding to an impact velocity of 5.4 m/sec (12 mph). This was confirmed by parametric computer simulations.

The flexibility of the spinal column and its resistance to injury through bending has been demonstrated by Sances, Yogananda, Maiman *et al* in experiments in which they were able to produce flexion distraction type injuries.. These tests were from drop heights of .9 to 1.5 meters corresponding to an impact velocity of about 4.2 m/s to 5.4 m/s (9.5 mph to 12 mph). It is also clear from clinical studies, that people can drop from high heights without neck injury and that head injuries are twice as frequent as neck injuries.

This characteristic of the head and neck, is well known to all who watch American football on TV, especially the close-up replays of tackles. While most of us don't think about it, a football helmet is designed to deflect and cushion the head from the high speed impact acceleration of the tackle. There are very few neck injuries (although the helmet is not designed to minimize the potential), because players are admonished by their coaches to keep the head up (thereby keeping the neck from being aligned). There appear to be literally thousands of football helmeted head strikes at closing velocities of 4.4 to 6.7 m/sec (10 to 15 mph), without neck injury each year.

But studies have shown that the vertical velocity of the far side occupant and roof in a rollover are typically 2 m/s and rarely exceed 4.4 meters/sec (10 mph). Furthermore this usually occurred when the vehicle was rolling at 1 rev. / sec. or more, keeping the torso from maintaining its velocity vector.

To illustrate the injury mechanism to a restrained occupant resulting from roof crush, consider a driver seated and restrained. A typical seat back angle is 20 to 25 degrees, and the pelvis and thoracic spine is rounded and without lordosis in the small of the back. The lower portion of the neck continues the rounding until the middle neck lordosis curves toward an erect head. The angle of the neck at C-6 may naturally be about 20 to 30 degrees forward of the car vertical even assuming the driver doesn't hunch his shoulders and duck the head.

Now imagine that the vehicle yaws counterclockwise so that the passenger side is leading and begins to roll. The driver moves up, over and down head first with a horizontal velocity (relative to the earth) of at least 10 m/sec and 2 to 3 m/sec vertically. The car is rolling at about one revolution per second sliding and pivoting over its contacts.

The roof rail next to the driver hits the ground somewhat after one-half roll. Around this time, with current vertically ineffective restraints, the drivers head is likely to be in contact with the roof panel whether the roof deforms or not. Here it becomes important whether the roof deforms.

- If the roof retains its structural integrity, the occupant's vertical velocity will go to zero. If the roof panel is not in contact with the ground or there is a least 2 cm of padding, the head will decelerate at 10 to 20 g over a duration of 15 to 20 ms. This is easily survivable. It is unlikely that the neck will be aligned because of the dynamics of the rollover. The seat belt and flexibility of the spine will decelerate the effective mass of the body without serious injury as long as the neck flexion angle and moment limits are not exceeded.
- If the roof crushes substantially, and the B-pillar moves inward as a consequence, the torso belt anchor may be brought closer to the center of the vehicle reducing further the effectiveness of the lap belt. The torso belt anchor may also be brought closer to the center of the vehicle further reducing the belt's effectiveness. The reaction of the head on the roof, and the motion of the lower torso toward the roof brought about by the roof crush then can exceed the capability of the spine to flex without failure, and failure occurs through the angular deflection of the neck.

The Hybrid II and III dummies are designed for frontal impact to move in reaction to high acceleration during 0.1 sec. and do not appropriately represent the motion of a human during a 2 to 7 sec. rollover. In this respect they are poor surrogates for humans in rollover crashes. While the head, neck, and torso of a dummy remain relatively aligned in a rollover test, the head, neck and torso of a human will flex considerably and in various directions to accommodate the forces on it. As will be seen in the next section, dummy rollover tests can be misinterpreted.

EARLY ROLLOVER TESTS AND THEIR INTERPRETATION

In 1975, Moffatt published "Occupant Motion in Rollover Collisions," in which he postulated that if an occupant struck the roof at about the same time the roof struck the ground in a rollover, head injury was independent of subsequent damage to the roof. Specifically, he stated, "It made no difference whether he struck the roof before, after, or during the damage occurrence: he still hit his head with the same contact velocity."

This is correct for local injury to the head if it strikes at the same time and at a point where the roof has come into contact with solid ground. That is, in such an instance, the roof and the head come temporarily to rest (in the vertical direction) at about the same time. Further displacement of the vehicle towards the ground is irrelevant to subsequent accelerations *of the head.* It should be noted, however, that head injury from such contacts can be substantially reduced by energy absorbing padding or structure in the interior of the roof since the head is traveling at only a few m/sec at the time of such impact.

Later, Moffatt articulated the theory of "torso augmentation" which postulated that neck injury came from the force imposed by the inertia of the torso, not from the force of roof deformation on the head. The torso augmentation theory is that once the head has been stopped by impact with the roof which is in contact with the ground, (if the restraint allows the head to reach the roof), the body must be decelerated by forces on the neck. For an occupant who started on the side of the vehicle opposite that which was leading as the roll began, the combination of rotation and small vertical motion can produce a vertical velocity of several m/sec at the time the roof over the occupant strikes the ground. But the duration of time over which the velocity is vertical and the displacement of the body in that direction is dependent on the body's roll rate.

If the torso/neck is very stiff, as in a Hybrid III dummy, the force on the neck can be high and the duration short. But if it is as compliant as a human, energy is dissipated over some time, displacement and rotation so that the force is reduced. Given that most people are uninjured in this circumstance with 5 cm of roof crush, and only moderately injured with 10 cm, the musculature and bending capability of the human spine must be able to absorb this energy. The rotational velocity of the body, which limits the time during which the bending displacement occurs, is often or intentionally overlooked. In other words, a cylinder rolling on a bumpy surface is a better analogy than a falling elevator.

In the mid-1980s, General Motors conducted two series of eight rollover tests of 1983 Chevrolet Malibu sedans. In eight of the sixteen tests, rollcages were installed in the vehicles to increase their roof strength. Half of the tests of the standard and rollcaged sedans had restrained dummies and half had unrestrained dummies. Orlowski, Bundorf and Moffatt analyzed these tests. They found "The rollcaged vehicles did not have any increased level of protection over the standard roof vehicles in these tests."

Because of the stiffness and alignment of the dummy neck, it bends very little, so they measured axial neck compression loads in the range of 2,000 to 7,000 N at a sensor just below the rigid head. They defined as "potentially injurious impacts" those that produced more than 2,000 N force on the neck. They claimed that the PII's occur before the roof crushed. They went on to say that there was no significant difference in the injury frequency or severity for the two types of roof.

The Hybrid III dummy was supposedly designed to represent severe axial compression injury at above about 4000 N, and at these levels, in the unrestrained tests, the driver (far side) dummy in the four production cars received 15 impacts, while in the rollcaged car it received only 8, making clear that a stronger roof has some beneficial effect. In the tests with restrained dummies, the ratio was 11 to 2, making clear that a combination of increased roof strength and restraint is even more beneficial. Furthermore, lateral and flexion neck injuries for restrained dummies were not reported although the ratio of such injuries using selected criteria appears to have been 10 to 1 and 1 to 0 in production versus rollcaged vehicles respectively. The 10 photographic analyses of A-pillar, roof contact was also generally not reported, but in cases preceded the PII's.

However, an experiment conducted with an unrestrained or loosely restrained Hybrid II or Hybrid III dummy cannot provide a realistic simulation of the potential for neck injury for four reasons:

First, the orientation and motion of a human torso during rolling and before roof touchdown would result in a pre-flexed real human spine/neck, and the necks of these dummies are far too stiff to permit such bending. Measurements of the forces on the neck would be similarly unrepresentative because the neck would not be realistically flexed at the time the measurements were taken.

Second, Sances' experiments show that in these circumstances, it is highly unlikely that significant axial compression loads could be imposed on the neck. And Sances indicates that clinically, flexion distraction, or flexion compression, not axial compression is the most frequently occurring neck injury. Our own investigations are consistent with this finding.

Third, the body of a human is not rigid as is the body of the dummy. It is inclined to flex and compress in a way that further limits loading of the neck when the roof strikes the ground in a rollover.

Fourth, Yogananda, Sances and Pintar have shown that the axial stiffness of the human neck structure is much lower than the neck structure of the Hybrid III dummy.

In their analysis of the unrestrained and restrained rollover tests, Orlowski, Moffatt and Bundorf measured a potential for injury an average of about three times for each dummy in each of the 16 rollover tests (about 90 dummy PII contacts). It would require more than say 6000 real world rollovers to produce that many neck injuries (to represent real world statistics). Conversely, these tests do not represent or explain the 96% of occupants who are not seriously injured in rollovers.

In every one of the unrestrained production roof vehicles, there was at least one dummy axial neck compression load of over 4000N (Orlowski, Bundorf, Moffatt, 1985, Figure 21). In every one of the restrained production roof vehicles there were at least two axial neck compression loads over 4000N (Bahling, Bundorf, Kaspzyk, Moffatt, Orlowski, Stocke, 1990, Figure 11). However, in accidents, we can see that neck injuries occur with far less frequency, and when they do occur they typically are not the burst fracture type injuries associated with axial compression type loading.

Conclusions by Orlowski and Moffatt that roof crush in real world accidents is *only* an indication of the severity of a rollover and not a cause of neck injury are therefore not valid. Injury measurements made on the dummy neck do not correlate with the forces that would be experienced by a human in the same circumstances. The Hybrid II and III dummies lack biofidelity for measuring injury in rollovers where the acceleration loads are much lower than in frontal crashes. The measurements of injury taken from the dummy indicate injury at a rate two orders of magnitude greater than occurs in actual crashes. This alone calls the validity of injury measurement in these experiments into serious question.

To the degree that a person is not ejected, has adequate survival space, and reasonable head impact protection, he or she is unlikely to be seriously injured in a rollover. Improved occupant restraints that limit torso movement toward the roof could further reduce the potential for head and neck injury.

A NEW THEORY OF HEAD AND NECK INJURY ROLLOVERS

The following observations provide a foundation for understanding head and neck injuries in rollovers:

Most rollovers involve a number of short low velocity impacts which incrementally scrub off the initial horizontal trip speed and support the rolling vehicle vertically. Occupant contact with the interior of a vehicle as it rolls are also typically benign. As a consequence, the vast majority of occupants in rollovers are uninjured or receive only minor injury.

The human head can withstand most such impacts that occur in rollovers without major injury. The remainder of the body – particularly the neck and torso – are highly flexible and resilient. Thus, they can accommodate most of the contact forces and even some intrusion in a rollover.

Rollover tests using Hybrid II and III dummies produce misleading results because of the lack of biofidelity in the flexibility of the neck and torso of these dummies.

Our theory is that there are two primary mechanisms of injury to restrained occupants from rollover, and that they occur relatively independently of each other.

- Severe head injuries typically occur in rollovers where the head strikes either a rigid part of the interior (such as an unpadded roof rail) or a rigid surface outside the vehicle (such as the road) where the relative speed of the roof and the road is relatively high (7 to 12 m/sec).

- Severe neck injuries typically occur in rollovers in which there is a substantial reduction in headroom. However, the head, neck and torso, considered as a unit, can flex and foreshorten while absorbing energy. Exceeding the spine's tolerance for such energy absorption may result in injury to the neck -- the most vulnerable component of this unit -- as it is bent beyond its limits. The bone and ligament structure of the neck is mechanically damaged and consequently can damage the spinal column.

Rollovers are not inherently violent crashes: that is, the forces generated by most rollovers are within human tolerance limits. Relatively minor modifications to vehicles could substantially reduce the probability of these types of injury to restrained occupants in rollovers. The next section will discuss those modifications in more detail.

VEHICLE IMPROVEMENTS

Neck injury to restrained occupants would be dramatically reduced if roof crush in most rollovers were limited to no more than 10 cm. (or an amount equal to the headroom for the range of occupants to be protected plus 6 cm., whichever is less). Improved belt systems that reduced vertical excursion in rollovers would further reduce neck injury.

We recently learned that Opel, for their FMVSS 208 dolly rollover tests, set an internal standard that "roof crush shall not exceed 10% of the internal height of the compartment," which is about 10 cm. The amount of allowed crush needs to be consistent with the restraint capabilities, the headroom provided and the field situations expected.

A simple, repeatable dynamic test of roof crush resistance could be developed based on the rollover tests, that have been conducted over the years. It would take into account the vertical and horizontal forces that are typical of a rollover. Those forces are dependent on the RVR in that a higher RVR will result in a higher ratio of horizontal to vertical forces on the roof. The degree to which the roof extends beyond the roll radius from the center of gravity of the vehicle can make it more vulnerable to lateral loads. This means that the roofs of light trucks and vans are likely to be more vulnerable in a rollover.

Vehicle Structures. Specific changes in a vehicle's design and materials would reduce the seriousness of head and neck injuries in contemporary vehicles at little cost or weight penalty. These include increased sheet metal thickness', improved component cross sections, and the use of structural foams. Monocoque roof structures are the most efficient current designs.

Occupant Restraint. The protection provided by many safety belt systems in cars is compromised by many characteristics. These restraint systems inadequately manage vertical excursions because of, for example, long buckle stalks, inappropriate anchor points, belt to seat cushion interactions, and window-shade devices. Door-mounted "automatic" belts are particularly poor for controlling vertical motion. Well-designed belts with anchors integrated with seating systems can substantially reduce an occupant's vertical excursion, and therefore reduce the opportunity for injury-producing head contacts with vehicle roof structures. Belt pretensioning, which has become common in European vehicles, could further reduce occupant excursion. However, current pretensioning systems lack a rollover sensor which means that they do not provide this important additional protection. Mercedes-Benz and BMW have demonstrated practical roll sensors to deploy a rollbar in a production convertible model.

Occupant Packaging. In the design of a new vehicle, its interior can be configured to increase the distance between an occupant's head and parts of the roof structure to be consistent with the amount of intrusion expected and the amount of excursion allowed by the restraint over the range of occupant sizes to be accommodated. It should also be consistent with preventing partial ejection. This, particularly in conjunction with the other countermeasures discussed here, reduces the likelihood of injury-producing head contacts.

CONCLUSION

Significant reductions in the number of severe and fatal injuries to unrestrained and particularly restrained rollover occupants are possible using currently available materials and technologies, and feasible designs that would add little cost or weight to new vehicles. The benefit in reduced catastrophic injury would far outweigh any added manufacturing and operating costs.

ACKNOWLEDGMENTS

The authors wish to express their appreciation to Dr. Rolf Eppinger of NHTSA, for his suggestion in 1990, to study non-injury accidents, in order to gain insight into how injury accidents occur and for his pre-publication comments on this paper. And to Dr. Carl Nash, recently retired from NHTSA, for his editorial suggestions and assistance. And to Dr. Joseph Kanianthra and Glen Rains of NHTSA for following up on our 1982-83 NASS study of injury risk under a collapsing roof, and publishing the residual headroom concept. And to Dr. Anthony Sances of the Biomechanics Institute for his insightful comments on diving neck injury.

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SHOULD CAR ROOF PILLARS BE EPOXY-FILLED FOR INCREASED ROLL-OVER STRENGTH?

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ABSTRACT

Some vehicle manufacturers have recently used lightweight epoxies to fill hollow thin-walled roof pillars in car subframes to increase their strength and energy dissipation capacity in roll-over crashes. It is questionable, however, whether epoxy filling is more efficient than increasing the wall-thickness of a member to improve strength to weight ratios. This paper examines vehicle roof pillars and how their breadth to thickness, b/t, ratio and epoxy filling effects their structural performance under gross bending deformation. A review of theoretical and experimental work carried out to date on void-filled versus hollow sections and an analysis of some typical vehicle roof pillar sections are presented. Conclusions indicate that with present epoxy fillers and roof pillar dimensions, there is no gain in peak strength to weight efficiencies when void-filled sections are compared to their equivalent in weight thicker-walled hollow members, unless compressive strengths of fillers can be increased and/or density reduced.

INTRODUCTION

Roll-over accidents account for around 19% of fatal vehicle crashes in Australia and are the major contributor to serious vehicle related spinal injuries resulting in paralysis [MUARC 1994]. Evidence from accidents indicates that a noticeable proportion of cars, especially four wheel drive vehicles (Fig. 1), fail to maintain a survivable occupant space during adverse roll-over accidents and therefore fail to protect the occupant [Dayawansa & Grzebieta 1990, Grzebieta & Dayawansa 1987, Rechnitzer & Lane 1994].

Australian Design Rules for vehicles offer no clear guidance for design against roll-over crashes for passenger vehicles. USA and Canada are the only countries that have a roof crush standard; CMVSS 216 which is identical to US FMVSS 216. However these standards are not very demanding and promote the use of glass and windscreen bonding glue as structurally adequate materials [Grzebieta 1995, Murray 1994]. Clearly the high occurrence of fatalities and severe injuries, and the lack of clear guidance to designers justifies research into improved framing design against roll-over crash loading.

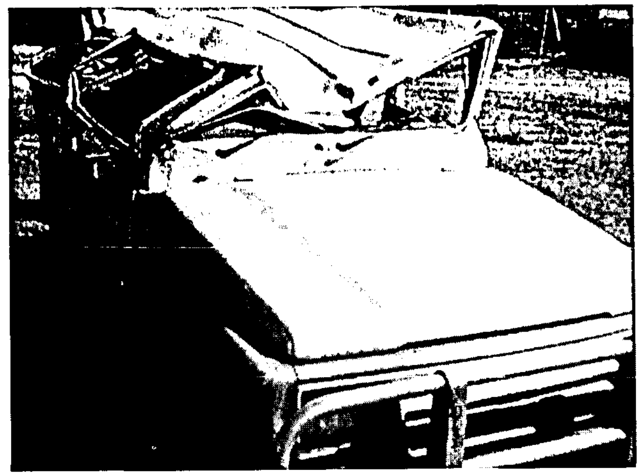


Figure 1. The occupant cell of a car acts as a survival space.*

The intention of this paper is to investigate whether there are any advantages of epoxy filling hollow thin-walled members as a means of strengthening car pillars and roof rails against flexural loading resulting from roll-over crashes. Epoxy filling is currently being used by some car manufacturers to strengthen structural body components, but on what theoretical or experimental basis, is unknown. After examining the structure of a typical vehicle roof pillar the theory of epoxy filling is investigated and some parallels in the building industry are mentioned. Finally the potential benefits of epoxy filling versus the use of thicker-walled sections are discussed and conclusions are then drawn.

VEHICLE ROOF STRUCTURES

Today's vehicle structures are typically composed of thin-walled steel elements connected by spot welding or glue, with an inner "occupant cell" composed of a more rigid 3-D framework of thin-walled closed sections. Part of the "occupant cell" is made up of roof rails and side pillars, and achieves its strength and stability through its fixed jointing between these members. The bonded windscreen glass acts as lateral bracing to this roof system

* Photo courtesy of N.W. Murray from his book "When it comes to the crunch"

until of course the glass shatters and the laminate then acts as the bracing! Under roll-over conditions, large bending moments are generated at these joint locations as can be clearly noticed in the case of the four wheel drive vehicle shown in Figure 1. The framework typically fails through flexure of the side pillars at the joint regions and at the points of loading. Figure 2 shows some typical cross-sections through A and B pillars (windscreen and middle pillar respectively) of a range of current sedan vehicle models. These pillars vary enormously in shape and plate make up along their length, generally decreasing in size from the vehicle base.

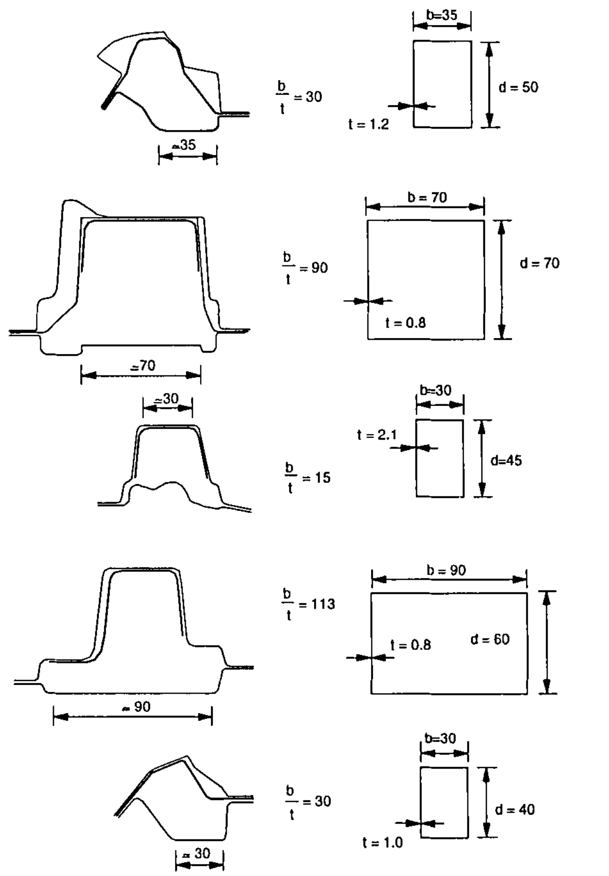


Figure 2. Some typical A and B pillar cross-sections of passenger vehicles with simplified rectangular equivalent.

When designing vehicles for roll-over crashworthiness, the peak bending strength, residual strength and energy dissipation capacity of the pillar sections must be considered. As thin-walled sections these pillars display certain characteristics which are a direct result of being "thin", i.e having a large unsupported plate width-to thickness (b/t) ratio. The effect of b/t ratios is demonstrated in the example of a rectangular hollow section with a high b/t ratio subjected to a large bending moment. Under this loading, the compression flange of the section buckles and a local plastic collapse mechanism forms [Kecman 1983]. As collapse proceeds, cross-section distortion causes a reduction in the member stiffness and a subsequent sudden decrease in the moment carrying capacity (Fig. 3, curves 1, 2 & 3). This also results in a reduction in the members capacity to dissipate energy, being the area under the moment-rotation M-θ curve. If, however, the b/t ratio is low enough, no buckling occurs and the section continues to carry its peak load until failure by strain rupture (Fig.3 curve 4). In this context sections that fail through a local buckling mechanism are described as "thin-walled" and those that fail by strain rupture are described as "thick-walled" [Murray 1984].

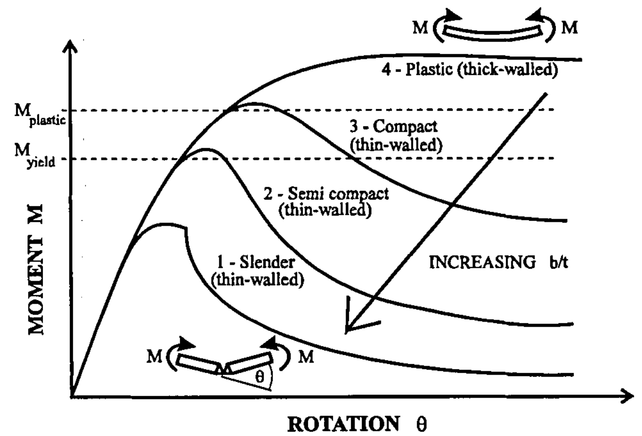


Figure 3. Moment-rotation curves for rectangular sections of varying b/t ratios.

Generally the higher the b/t ratio of the section the greater the reduction in the full plastic capacity due to local plate buckling effects. There are also secondary effects apart from b/t, which contribute to the local buckling of a section. Three major secondary influences are, the manufacturing and welding procedure of the section, initial out of plane imperfections and localised out of plane loading positions.

REVIEW OF THEORY FOR ANALYSING EPOXY FILLED TUBES

Clearly it would be preferable to use "thick-walled" sections in vehicle roof pillars because of their extended moment capacity with rotation, resulting in greater energy absorbing characteristics. However, material saving and weight efficiency, which leads to greater fuel efficiency, typically dictate the use of thin-walled profiles as the more economical section.

One means of improving the characteristics of a thin-walled section is to reduce the b/t ratio to that of a "thick-walled" section, i.e. use thicker plating. Another means is to prevent the local buckling mechanisms from forming by the addition of a suitably stiff material. In the case of closed sections, such as car pillars, this can be achieved by filling the section with a light weight material. This has the effect of providing sufficient lateral stiffness to the section walls to prevent local buckling, and therefore prevent local plastic collapse mechanisms from forming. The section then behaves like a "thick-walled" member with no strength or energy absorption reductions when subjected to large bending deformations.

A number of analogous studies on the axial and flexural behaviour of steel tubes filled with concrete have been reported in structural engineering research literature [Bradford 1991, Zhang & Brahmachari 1994, Lu & Kennedy 1994]. Flockhart and Murray (1994) looked at axially compressed hollow and epoxy filled spot-welded thin-walled column tubes. Grzebieta and White (1994) and Grzebieta et al (1995) described the behaviour of epoxy filled simple beams and cantilever beams subjected to bending loads.

Grzebieta et al (1995) looked in detail at the bending capacity and the rate of energy absorption for filled and unfilled square sections of varying b/t ratios. They used a non-expansive two part epoxy, of approximate density of $0.5 g/cm^3$ to fill the sections. Minimal adhesion between the steel and filler and the relatively low compressive and tensile yield stress of the filler resulted in the epoxy contributing negligibly to the section strength in bending. However, Flockhart (1994) in his paper on axially loaded columns noted a contribution to the compressive capacity of the section from the filler using a similar epoxy. Grzebieta et al (1995) subjected epoxy filled and unfilled square hollow sections of varying b/t ratios to a pure bending moment until failure. The testing procedure and results are described in their paper in detail. A comparison of energy absorption and moment capacity between the filled and unfilled sections were then plotted and are shown in Fig. 4 & 5.

Theoretical maximum bending moments for filled and unfilled sections as well as plastic residual load collapse curves were calculated by Grzebieta et al (1995) using equations developed by Kecman (1979) and Grzebieta and White (1994). It was verified that this theory could used to predict the behaviour of thin and thick-walled, hollow and epoxy filled square closed sections.

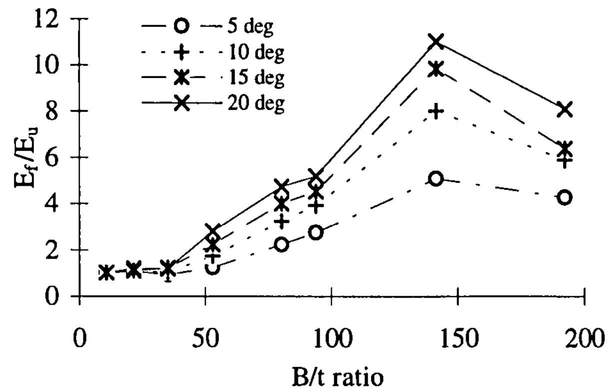


Figure 4. Ratio of energy absorption of filled beam over energy absorption of unfilled beam E_f/E_u versus b/t ratio ($B \equiv b$) [after Grzebieta et al 1995].

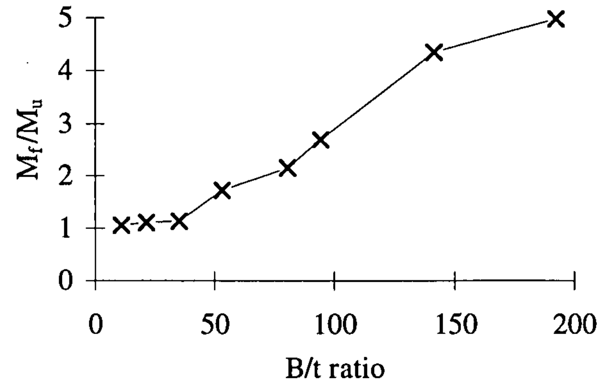
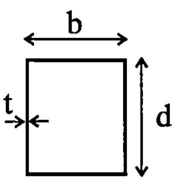
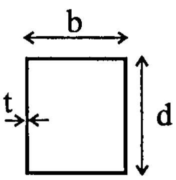


Figure 5. Ratio of peak moment of filled beam over peak moment of unfilled beam (M_f/M_u) versus b/t ratio ($B \equiv b$) [after Grzebieta et al 1995].

Grzebieta et al (1995) concluded that, for square sections where b/t ≤ 40, epoxy filling provides negligible increases in strength and energy absorption characteristics. Above this limit the benefit of epoxy filling increases with b/t ratio, to an upper limit, in this case b/t = 150, above which they concluded that the ductility of the material becomes the critical failure mode. This upper limit varies and is a function of the section material properties of yield stress and material ductility. Figures 4 & 5 indicate up to a fivefold increase in bending moment strength , and a tenfold increase in energy absorption for rotations of 20°. This clearly points to benefits in epoxy filling when b/t ratios are high.

Table 1. Limiting width to thickness ratios, $\varepsilon = \sqrt{275 / \sigma_y}$

(Elements which exceed these limits are to be taken as slender cross-sections.)
[Extract from Table 7, BS5950 Part 1 1985]

Type of element	Type of section	Plastic	Compact	Semi-compact
Internal element of compression flange	Built-up by welding	$\frac{b}{t} \leq 23\varepsilon$	$\frac{b}{t} \leq 25\varepsilon$	$\frac{b}{t} \leq 28\varepsilon$
	Rolled sections	$\frac{b}{t} \leq 26\varepsilon$	$\frac{b}{t} \leq 32\varepsilon$	$\frac{b}{t} \leq 39\varepsilon$
Web, with neutral axis at mid-depth 	All sections	$\frac{d}{t} \leq 79\varepsilon$	$\frac{d}{t} \leq 98\varepsilon$	$\frac{d}{t} \leq 120\varepsilon$

PARALLELS IN THE BUILDING INDUSTRY

It is interesting to note that knowledge of how varying b/t ratios effects the behaviour of structural members has existed in the building industry for many years. In structural building design, there is a general member classification based on b/t criterion for cross-sections subject to various loading regimes, as shown in Fig. 6 [Owens 1989]. Figure 3 further illustrates this criterion. Three of the four b/t classifications, slender, compact and semi-compact , show varying degrees of moment collapse after experiencing peak moment in Fig. 3, making them by definition "thin-walled". A "plastic" section in Fig. 6, is by definition a "thick-walled" section, showing no reduction in moment capacity with rotation.

Table 1, extracted from BS5950-British Standard Structural Steel Design code, indicates the limiting b/t ratios for member classification for closed rectangular sections. b/t ratios which exceed these limits are slender cross-sections, and the design strength is then reduced by the use of further equations in the code, which basically relate the reduced design strength to the inverse of the b/t ratio. Table 1 also shows that for a rectangular section under pure bending, with d/b < 3 (i.e. 79/23), it is only the b/t ratio, not d/t ratio, that governs the section behaviour. This allows typical (d/b < 3) rectangular sections to be treated in a similar manner to square sections. All

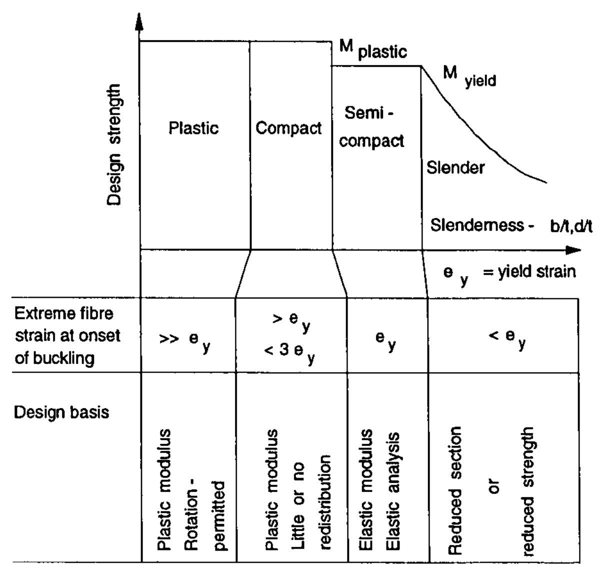
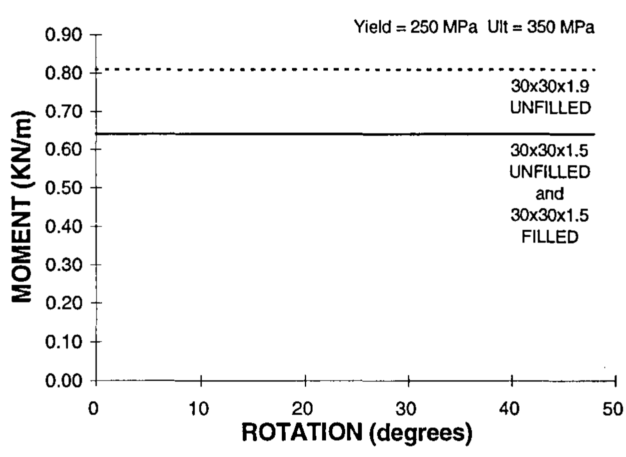


Figure 6. Classification of slenderness limitations for local buckling and their implications on design strength in flexure [after Owens 1989].

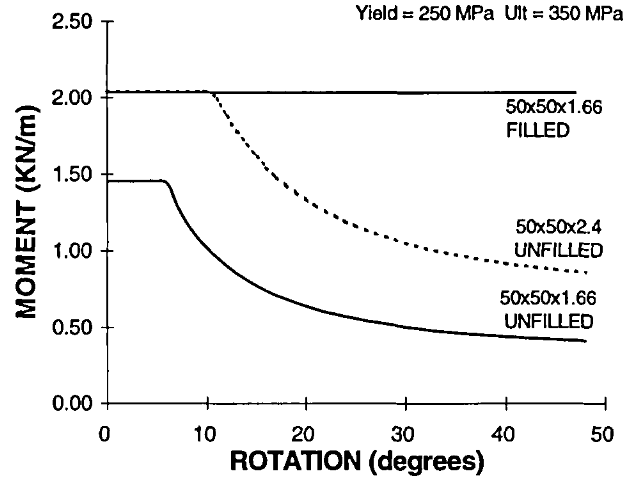
this information could be applied to any structural element including those making up a vehicle subframe, simply by estimating a b/t ratio and knowing the material yield strength. Therefore we can conclude that similar concepts to those used in the building industry are also relevant when addressing the structural makeup of car components, and can be used to assess the likelihood of early local plastic mechanism failure and the level of strength reduction as a result of early local mechanism failure.

EPOXY FILLING VERSUS THICKER WALLED SECTIONS

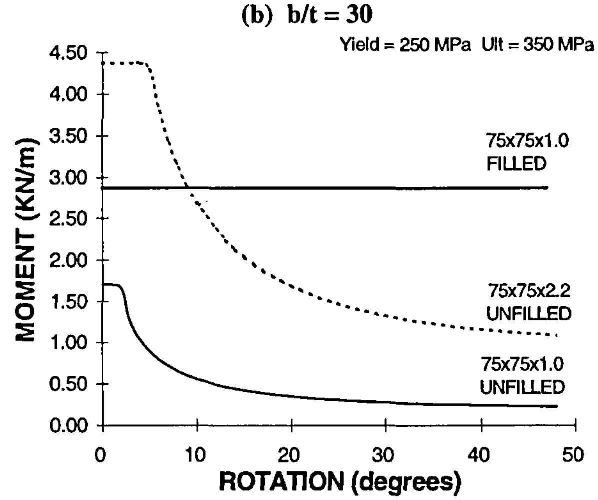
In many structures such as vehicles, however, weight is of primary importance. When assessing the viability of epoxy filling with respect to vehicle strengthening one should also consider the weight economics of the design and contemplate increasing plate thickness as an alternative to epoxy filling. To illustrate this point with an example, four b/t ratios are considered; namely b/t = 20, 30, 75 & 150, as shown in Table 2. Fig. 7 shows M-θ plots based on theory described in Grzebieta et al (1995), for 3 cases of unfilled, epoxy filled and thicker-walled sections. In each plot the profile has the same breadth and depth. The thickness of the thicker-walled section in each plot was calculated based on a cross-section equivalent in weight to the thinner epoxy filled member. The filler density was assumed as 0.5g/cm3 which is typical of the density currently being used by manufacturers. When calculating the full plastic moment capacity of the epoxy filled members, the ultimate strength was used in place of the yield strength to allow for rapid strain hardening at large deformations. Also the elastic behaviour at the onset of loading was ignored in all cases to simplify energy calculations.



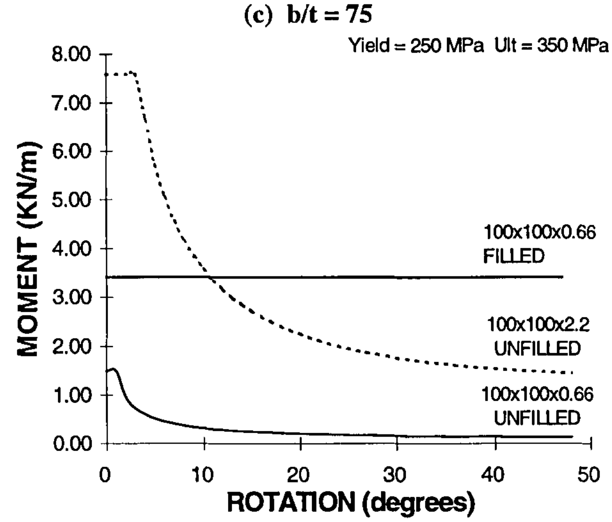
(a) b/t = 20



(b) b/t = 30



(c) b/t = 75



(d) b/t = 150

Figure 7. Moment-rotation curves for filled, unfilled and thicker walled sections (based on equivalent cross-section weight of the filled section).

Table 2. Energy absorption (at 20°) and peak moment ratios for different cross-section proportions

Section size	30x30x1.5*	50x50x1.66	75x75x1	100x100x0.66
b/t Ratio	20	30	75	150
Section classification (to BS5950)	plastic	compact	slender	slender
Equivalent weight steel section	30x30x1.9	50x50x2.4	75x75x2.2	100x100x2.2
Ratio of peak moment of filled to unfilled	1.0	1.4	1.7	2.3
Ratio of peak moment of thicker steel section to unfilled	1.2	1.4	2.6	5.0
Ratio of energy absorption of filled to unfilled (at 20°)	1.0	1.9	4	7.4
Ratio of energy absorption of thicker steel section to unfilled (at 20°)	1.2	1.7	4	9.3

* 30x30x1.5 section is "plastic" or thick-walled, and therefore experiences no local plastic mechanism failure and accordingly shows no benefit from epoxy filling.

Table 2 gives the ratios of peak moments and energy absorption of filled to unfilled sections, and thickened to unfilled sections. At very low b/t ratios (plastic/thick-walled classification) there are no strength or energy improvements from epoxy filling, implying epoxy filling is not a strengthening option in this b/t range. As b/t increases into the thin-walled/slender range, however, the increased peak moments and energy absorption resulting from increasing the plate thickness exceed those of epoxy filling, increasing as b/t increases. It can therefore be concluded that if weight is a critical issue, based on a filler density of 0.5g/cm3, increasing the section steel plate thickness is the better alternative of the two options for increased peak strength performance. However, Figures 7 (b), (c) and (d) also show that filled profiles provide an increase in residual force or ductility at large distortion angles.

Further work is currently under way to investigate the effects of varying (lowering) filler density.

GENERAL DISCUSSION AND CONCLUSIONS

Vehicle A & B pillars can be very complicated elements with irregular section shapes and sizes and varying plate makeup along their length, as well as localised holes for fitting and weight efficiency (refer Fig. 2). Loading profiles are always non-uniform under crash conditions and end fixing restraints are a function of the complex jointing between pillars and roof rails and base fixings. Pillars are typically spot-welded and failure of the section may be a result of spot-welding failures rather than of buckling. There are also issues of production cost, ease of manufacture and weight efficiency, and other possible beneficial effects such as sound/vibration reduction. All these variables also require careful consideration when assessing the possible advantages of epoxy filling or plate thickening as a means of strengthening.

In order to assess possible strengthening options of pillars against pure bending, based purely on b/t ratios, the section must first be simplified to a rectangular equivalent. Examples of this can be seen in Fig. 2. Based on a study of roll-over crashes, it was found that side pillars typically fail in the top half of their length where sections are of reduced taper. b/t ratios in this region (over the top half) of A & B pillars, of standard passenger vehicles, were found to lie within the range of 15-110. Based on the discussion above, this places side pillars outside and at the lower end of the b/t range that experiences optimum strength and energy absorption increases as a result of epoxy filling. This implies that though some strength and energy benefits may be gained from epoxy filling, they would not lie within the optimum range. If however sections reach a b/t range of 110-200, which may be the case in larger vehicles such as agricultural and earth-moving machinery, clearly greater improvements would be gained, and epoxy filling could be considered as an efficient means of strength and energy absorption improvement.

Taking section weight efficiency into consideration, as detailed in Table 2, increasing the section plate thickness as an alternative to epoxy filling (based on an epoxy filler density of $0.5g/m^3$) typically yields improved peak strength and energy absorption characteristics at all b/t ratios, increasingly so at higher b/t ratios.

Overall we can conclude that epoxy filling car pillars, as a means of strengthening against gross bending deformations resulting from roll-over, is not an efficient means of improving the structural performance of the side pillars due to the low effective b/t ratios of the sections. Based on a typical filler weight of $0.5kg/m^3$, a simpler and more weight efficient means of improving pillar peak load and energy dissipation performance overall, regardless of b/t ratios, is to increase the section plate thicknesses. However, void filling does improve residual load capacity at large deformation angles and is a suitable means of improving the performance of existing thin-walled members where increasing plate thickness is not a viable design option.

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EFFECTIVENESS OF AIRBAGS IN AUSTRALIA

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ABSTRACT

General Motors - Holden's Automotive (Holden) was the first Australian manufacturer to introduce a package of new safety features with the release of the VR Commodore, including a driver's side airbag. This was followed two years later with a passenger airbag, released in the VS model. These airbags, in conjunction with an improved seat belt system, have the distinction of being specifically designed for Australian driving and accident conditions and as a consequence are different to those found in vehicles designed overseas. To determine the effectiveness of these systems the investigation of a number of field accidents has been conducted. The preliminary results of this work, although not all statistically significant, are very encouraging and suggest that the airbags have had a positive effect on reducing occupant injury with few of the negative side effects that are now being identified with some foreign airbag systems.

INTRODUCTION

There has been a marked increase recently in the safety technology of Australian passenger cars aimed at improving occupant protection. The VR Commodore, released in 1993, heralded a number of significant advances in local automotive safety. This primarily consisted of the driver's airbag (SRS system), the first to be released on a locally built vehicle and the incorporation of webbing clamps on the seat belts fitted at the front seating positions. This was followed in 1995 with the VS Commodore which featured a front passenger SRS system, again the first for a local vehicle. The development of these restraints represented a major challenge for Holden's Advanced Engineering group as the primary design objectives were to produce systems that would best suit Australia's driving and accident environment, not simply duplicate American or European designs, and act to supplement the protection provided by the seat belts.

To gauge the effectiveness of the GMH airbags, the Monash University Accident Research Centre (MUARC) was commissioned to inspect all crashes in the Eastern Sates of Australia where the airbag(s) was deployed. Similar severity non-airbag VR Commodore crashes and a sample of previous VN and VP models were also inspected as controls. Inspections have been continuous since the release of these vehicles and there are currently 64 airbag cases, 40 non-airbag cases and 54 previous model controls. The data collected from these crashes have been analysed to determine the success of the restraint systems and the results so far have been very positive.

Design Philosophy of Restraint Systems

Australia has a unique driving and accident environment. Two points in particular characterise the differences between the local situation and that experienced overseas: (i) there is a high incidence of frontal collisions, and (ii) there are high rates of seat belt wearing. As a result of these differences, an automotive restraint system designed specifically to suit the local environment will be different to that found in vehicles from other countries and the restraints employed in the Commodore are a good example of this.

The high rate of frontal impacts experienced in Australia, in comparison to other collision modes, is the result of a less sophisticated road network. In contrast to the road systems found in some other countries, such as the US, there are vast amounts of undivided roads and more roadside hazards, such as trees and poles. This results in a high incidence of frontal accidents, including offset and oblique impacts and is well illustrated by the fact that in the state of Victoria 56% of all collisions in which one or more of the vehicle occupants are injured results from a frontal crash while in the US only 39% of such collisions are attributed to frontal impacts. This trend is even more pronounced when looking at serious (at least one occupant admitted to hospital) and fatal collisions where over 62% of such injuries are attributed to frontal impacts.

Australia also has a high rate of seat belt wearing; over 95% for front seat occupants and over 80% for rear seat occupants. This is a result of legislation that came into effect in the state of Victoria on 22 December 1970 which mandated the use of belts for the first time in a state with a substantial vehicle population. It is estimated that this legislation resulted in a 12% reduction in the number of driver and front seat passenger deaths by 1971 when the overall usage rate was only 50%. This legislation was soon adopted by the other Australian states and territories and today some 40 countries world-wide have mandatory wearing laws.

These two points have had a major influence on the design of the restraint systems employed in the Commodore and allowed two important assumptions to be made early in the development of these systems: (i) the airbag trigger threshold would be set such that they only deployed when an accident was of such severity that the seat belts alone could not offer complete protection, and (ii) the airbags would be tuned to inflate as less aggressively as possible. As a consequence, fundamentally different restraint systems have been implemented in the Commodore than are typically found in many overseas vehicles, especially those developed for unrestrained occupants. In particular, the airbags have been specifically designed to offer supplementary protection to that provided by the seat belts.

Seat Belt Design

Given that seat belts are so frequently utilised in this country it is important that they provide the primary means of protecting vehicle occupants by offering the maximum possible protection. The webbing clamp seat belts employed in the front seating positions represent a significant improvement over the conventional Emergency Locking Retractor (ELR) design of belt. They incorporate a metal clamp on top of the retractor which reduces both the payout and spooling of the webbing as the belt is loaded in an accident. This restricted payout has the benefit of providing more controlled occupant kinematics and reduces the risk of the occupants striking the steering wheel or dash. In comparison to ELRs, the webbing clamps release around 100 mm less webbing in a standard ADR 69/00 (48 km/h 0° frontal) barrier test.

SRS Sensing System

The airbags in the Commodore are triggered by a single point Sensing and Diagnostic Module (SDM). This unit provides a centralised, self-contained sensing and triggering system that is capable of distinguishing between a minor parking bump and a potentially injurious collision. It also performs a diagnostic role by continually testing the SRS system and assisting service personnel by indicating the cause and location of any faults. It is fitted with an energy reserve capacity such that its function is temporally preserved in a collision if power to the module is lost and most importantly provides superior collision sensing with the Commodore's crash pulse. Through a range of impacts the SDM is capable of triggering the airbags within acceptable time limits to ensure that the occupants received the maximum benefits of the airbags. This is an important point as an SRS system that cannot deploy within the necessary time will not only be of much less benefit but may present an injury risk to the occupants if they contact the airbag cushions while they are still inflating.

The concept behind the sensing operation of an SDM is relatively straight forward. Once a vehicle becomes involved in a collision, an accelerometer measures the resulting deceleration as the vehicle's structure crushes. This signal is mathematically processed and compared with predetermined thresholds based on jerk, acceleration, velocity and energy criteria. At the same time, a simple mechanical deceleration switch is checked. If the requirements of both the sensing algorithm and the mechanical switch are met, indicating that the vehicle is involved in a severe frontal collision, the airbags are deployed. If only the accelerometer algorithm or the mechanical switch are activated then the airbags are not fired. The purpose of the mechanical switch is to act as a guard against accidental firing if the vehicle is subject to intense electro-magnetic interference as is often the case near communications towers and airport radars, etc. Vehicles fitted with both driver and passenger SRS systems will deploy both airbags simultaneously when involved in severe collisions.

The thresholds with which the accelerometer signal is compared are derived from vehicle barrier tests. Each test produces a distinct 'crash pulse' (deceleration profile) as the various impacts cause the body of the vehicle to deform in different ways. Hence a range of tests must be performed to produce a full SDM calibration, including: frontal, oblique, off-set, pole and under-ride impacts. A series of 'non-deploy' tests must also be performed to ensure that the SDM will not trigger in situations where airbag inflation would be of no benefit to the occupants, including low speed impacts or when the vehicle is being exposed to severe driving conditions, such as through pot-holed roads or during emergency braking over rough surfaces. Approximately 30 barrier tests were conducted in the development of the VR Commodore's sensing system and another 15 in the development of the VS Commodore's system.

Given that the Commodore's SRS systems have been developed for belted occupants, the airbag deployment thresholds have been set relatively high. The system's no-fire limit (the equivalent frontal barrier impact below which the airbags should not deploy) is around 20 km/h while the all-fire limit (the equivalent frontal barrier impact above which the airbags should always deploy) is approximately 28 km/h; between these limits the airbags may deploy depending upon the circumstances of the impact. This is significantly higher than the thresholds employed in some other restraint systems; some vehicles have no-fire thresholds of 12 km/h.

Airbag Design

The airbag modules in the Commodore are designed to deploy as less aggressively as possible while still providing the necessary protection to occupants of different size, weight and sex whom will be potentially involved in a variety of collisions. Great efforts have been taken in the development of the inflators and cushions to ensure they present as little risk as possible to the occupants during inflation. Since the airbags have been designed to operate in conjunction with the seat belts, they are only required to decelerate the occupant's head and upper torso as the primary retardation is provided by the belts. This is fundamentally different to many other airbag designs especially those used to protect unrestrained occupants. Such systems typically utilise high performance inflators in conjunction with cushions with low venting rates. This combination ensures that the airbags are sufficiently stiff to decelerate unbelted occupants. While such systems operate well in standard accidents they can present an increased risk to occupants who are close to the airbags when they deploy, such as small female drivers. Such occupants are increasingly being identified as suffering inflation induced injuries (I^3) which are injuries directly attributed to the deployment of the airbags.

Both the driver and passenger airbags in the Commodore employ sodium azide inflators. The driver's unit uses a moderate performance inflator which yields a peak pressure of 300 kPa in a standard 1 cubic foot tank test. A fully coated, full-sized cushion is employed which has a volume of 65 litres when completely inflated. Four 275 mm tethers and an innovative folding pattern are used to control the shape of the cushion during deployment and prevent it from inflating directly towards the driver. The tethers consist of strips of material that connect the front face of the cushion to the module housing and prevent it from deploying beyond a predetermined limit. The folding pattern helps ensure that a flat surface is presented to the driver such that the cushion has a lower tendency to balloon around their neck. Two 45 mm diameter vents allow the cushion to rapidly deflate as the driver contacts it which decelerates the occupant's head and upper torso as gently as possible.

The passenger airbag is significantly larger and displaces a volume of 120 litres when fully inflated. However it also employs a relatively gentle inflator which produces a peak pressure of 240 kPa in a standard 100 litre tank test. Again, tethers panels and a specific folding pattern are used to control its shape during deployment. Two 30 mm vents allow the gas to escape from the cushion as the passenger contacts the bag.

Approximately 150 Hyge sled tests were conducted in the development of the restraint systems employed in the VR and VS Commodores. These tests, as with the barrier tests, were performed at Holden's Lang Lang Proving Ground, south east of Melbourne and utilised the then recently purchased 'family' of Hybrid III dummies which consists of several 50th percentile male, a 5th percentile female, a 95th percentile male and several child dummies. The performance of all these dummy types was evaluated to ensure that the maximum protection would be offered to all occupant not just those represented in federal certification crash tests.

ANALYSIS OF EFFECTIVENESS

An analysis was undertaken for GMH by the Monash University Accident Research Centre to determine the effectiveness of the GMH airbag fitted in Australian Commodores. Data were collected on ***three*** versions of Holden Commodores involved in frontal crashes of minimum tow-away crash severity, namely (1), baseline cars (models VN & VP, the predecessor to the first airbag model); (2), airbag models VR & VS, where the optional airbag was fitted; and (3), non-airbag VR and VS models where the airbag option was not taken up. A total of 158 eligible crashes were inspected comprising 54 baseline, 64 airbag and 40 non-airbag cars.

Overall Findings

There were no noticeable differences between the two samples in terms of type of frontal crash and breakdown of driver age and sex. There were also no marked differences in seat belt wearing rates either between airbag or non-airbag cases which is reassuring as it suggests that the presence of airbags does not mean that drivers are less likely to wear their seat belts in these cars. The number of kilometres travelled was, however, considerably higher for the baseline cars than the more recent VR and VS models, simply because they were an older fleet at the time of inspection with higher exposure. However, this was not felt to be a major problem for this analysis. Given these findings, it was concluded that combining the baseline and non-airbag cases to form no airbag controls was appropriate in the subsequent analysis.

Figure 1 shows the delta-V distributions for the airbag and control (baseline plus non-airbag) cases where delta-V was either known or could be calculated (69% of cases). While there were differences between the two distributions, the modal values were similar (41-50km/h) and there was no appreciable differences in mean impact velocity between both categories up to 61km/h (Airbag mean = 38.6; Control mean = 38.4). The delta-V values above 61km/h were grossly different as was the outcome severity of these occupants. For the nine control occupants, their delta-V values ranged from 61km/h to 102km/h and two-thirds were hospitalised and the rest only required A&E department treatment while the one severe airbag crash was at 108km/h and the driver was killed. Because it was felt that the airbag was less likely to be effective at these high crash severities and the bias that including these few high delta-V cases was likely to have on the results, the analysis was confined to crash severities below 60km/h. Thus, the outcome for drivers in 63 airbag and 85 non-airbag control vehicles was compared as a measure of airbag effectiveness among GM Holden Commodores.

Injury Analysis

The body region injury outcomes of the airbag and non airbag drivers involved in tow-away frontal collisions is shown in Table 1. Drivers in airbag deployed Commodores had significantly fewer chest injuries of all severities (c^2=5.8, p<.05) and there was a trend towards fewer head, face and abdomen-pelvic injuries, albeit not statistically significant. Head injuries of moderate severity and all upper extremity injuries did approached significance (c^2=2.8, p<.10 for both comparisons). The increase in upper extremity injuries, in particular, was most noteworthy where airbag occupants were about 14% more likely to sustain these injuries than their non-airbag counterparts. However, this was confined exclusively to minor (AIS 1) injuries. The increase in the percent of spinal injuries for airbag occupants, while not significant, was of some concern. On closer examination, the two severe spinal injuries to the airbag occupants were fractures to T8/9 in the thoracic region with unknown sources of injury while the one severe spinal injury was a fractured L2 vertebrae to the lower back, also of unknown source. Because of the low number of cases, these findings should be taken as indicative only at this stage.

Tables 2 further shows the mean ISS, probability of injury and average Harm sustained for these drivers in frontal crashes. Of particular note, drivers in airbag deployed Commodores generally had a lower ISS score and corresponding lower probabilities of injury at each level than did drivers of non airbag crashed Commodores. The Harm savings to the driver for airbag equipped vehicles was A$20,000 per crash in A$1995 prices (Fildes, Digges, Carr, Dyte & Vulcan 1995).

Source of Injury

The source of injury for drivers in airbag and non airbag GM-Holden Commodores involved in frontal crashes is shown in Table 3. There were only a few differences in the source of injury patterns between airbag and control cases. Airbag occupants had slightly fewer contacts with the steering assembly, especially those that resulted in severe injuries but they had more contacts with the windscreen and header rail. They also had more contacts with the roof but fewer with the door panel. Of particular note, airbag occupants had slightly fewer seatbelt induced injuries. 14% of their injuries, , albeit of minor (AIS 1) severity, were from contact with the airbag itself. There were too few injury cases available to break these findings down any further to examine specific body region injuries by contact source.

DISCUSSION

Australian passenger cars have recently seen a significant increase in the level of protection afforded to their occupants. Holden was the first vehicle manufacturer to develop and introduce advanced restraint systems that would best suit Australia's driving and accident environment. As seat belt wearing rates in Australia are relatively high, the Holden airbags have been specifically designed to supplement the protection provided by the belts.

While limited in the amount of data available, the GMH Commodore airbag effectiveness analysis was encouraging. There was a significant reduction in chest injuries and an indication of a reduction in serious head injuries for those injured in airbag Commodores, compared to similar non-airbag controls. The reduction in chest and possibly head injuries for drivers with deployed airbags confirmed the beneficial effects of these

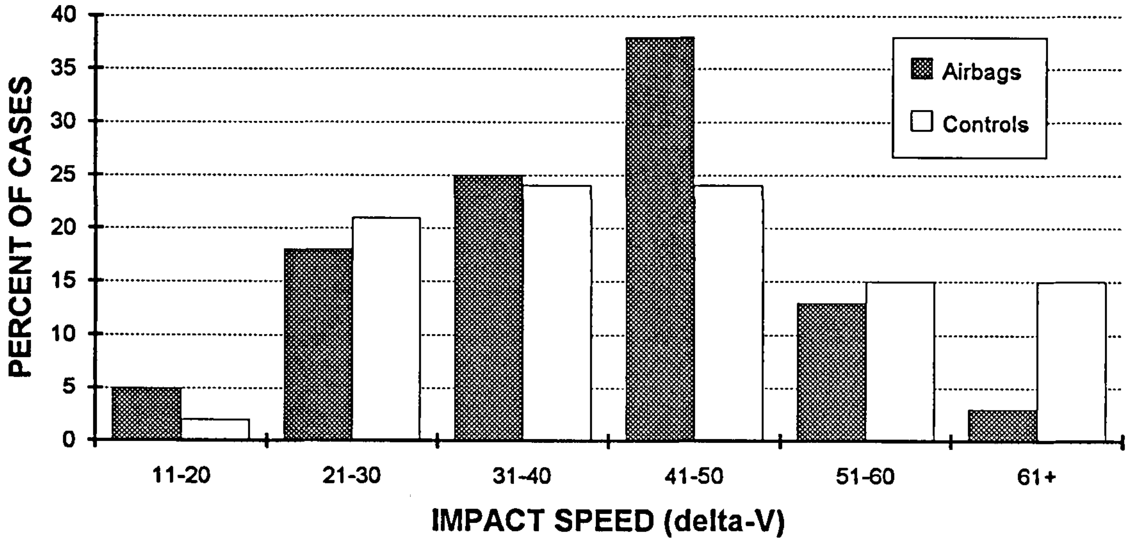

Figure 1 *Delta-V distribution for airbag and control Commodores where change of velocity during impact could be calculated (69 percent of cases)*

Table 1 - Percent of injuries to drivers of airbag and control Commodores involved in tow-away frontal crashes

Body Region Injured	Airbag Cases (n=63)			Non Airbag Controls (n=85)		
	All AIS	AIS 2+	AIS 3+	All AIS	AIS 2+	AIS 3+
Head	12.7%	4.8%	1.6%	14.1%	12.9%	2.4%
Face	15.9%	nil	nil	22.4%	2.4%	nil
Neck	23.8%	nil	nil	27.1%	3.5%	1.2%
Chest	25.4%	6.3%	1.6%	44.7%	14.1%	5.9%
Abdomen/pelvis	28.6%	nil	nil	30.6%	2.4%	2.4%
Spine	6.3%	3.2%	nil	2.4%	1.2%	nil
Upper extremity	54.0%	4.8%	nil	40.0%	7.1%	nil
Lower extremity	39.7%	4.8%	nil	32.9%	4.7%	nil

AIS scores range from 1 (minor) to 6 (untreatable). Multiple body regions included (MAIS per body region)

***Table 2** - Mean Injury Severity Score (ISS), probability of injury and Harm sustained by drivers of airbag and non airbag control Commodores involved in tow-away frontal crashes*

Body Region Injured	Number of cases	Mean ISS	Mean Harm ($ 000s)	Probability of injury		
				AIS 2+	AIS 3+	ISS 15+
Airbag cases	63	2.6	9.2	0.19	0.03	0.02
No airbag controls	85	5.4	29.2	0.31	0.07	0.05

ISS is the sum of the 3 MAIS body region scores squared.

***Table 3** - Sources of body region injuries to drivers of airbag and control Commodores*

Source of Injury	Airbag Cases (n=63)		Non Airbag Controls (n=85)	
	All AIS	AIS 2+	All AIS	AIS 2+
Front screen & header	6.3%	nil	3.5%	1.2%
Steering assembly	15.9%	1.6%	21.2%	7.1%
Instrument panel	23.8%	3.2%	22.4%	3.5%
Side window & frame	3.2%	nil	2.4%	1.2%
B-pillar	nil	nil	1.2%	1.2%
Roof surface	4.8%	1.6%	nil	nil
Door and fittings	nil	nil	4.7%	1.2%
Floor & toepan	14.3%	4.8%	12.9%	2.4%
Seat belts	49.2%	7.9%	52.9%	11.8%
Airbag	14.3%	nil	nil	nil
Exterior contacts	nil	nil	1.2%	1.2%
Other & unknown	55.6%	4.8%	49.4%	10.6%

AIS scores range from 1 (minor) to 6 (untreatable). Multiple body regions included (MAIS per body region) units for occupants of these Australian vehicles. The reduction in Harm for airbag occupants was around A$20,000 per crash for crash severities up to 60km/h (the expected range for which airbags offer maximum protection).

In North America, Dalmotas (1995) compared the performance of restrained drivers in US airbag cars that crashed with a control sample of restrained but no airbag crashed vehicles. It should be noted that the Holden Commodore airbag is similar in size to US airbags, although has different firing thresholds and generally a lower, softer deployment rate. He found a reduction in severe head injuries (AIS 3+) of between 42% and 96%, depending on crash severity which is slightly better than the 33% reduction observed here, a difference that can probably be attributed to variations in seat belt design between the two countries. Involvement rates for these severe head injuries were similar in both studies (in the Canadian study, there were 1.3% serious head injuries for delta-V < 32km/h compared with the 1.6% severe head injury rate observed here for delta-Vs from 11 to 60km/h). However, Dalmotas (1995) reported some conflicting results with those found in this study. He noted a substantial increase in chest and abdominal injuries of AIS 3+ severity of over 250% among his airbag sample. In this study, there were reductions observed in both these body region injuries compared with control cases (73% fewer chest and no severe abdominal injuries). Unfortunately, it was not possible to segregate the low impact severity cases in the Canadian analysis. Dalmotas also reported a substantial increase in AIS 3+ upper extremity injuries for airbag deployed cases (an increase of between 8 and 90 times over that of his controls, depending on crash severity). This is also in contrast to the findings reported here; there were no cases of AIS 3+ upper extremity injuries reported for either airbag or control cases and for the AIS 2+ injuries, airbag occupants sustained 32% fewer upper extremity injuries than similar control cases.

These differences could be explained in one of two ways. First, it is not clear whether Dalmotas excluded high delta-V cases in which case most of his findings could be influenced by high speed impacts where the airbag may have less effect on injuries. Alternatively, as the US airbag is designed as a primary restraint unit, it is more aggressive than its Australian counterpart. As noted earlier, the Commodore airbag is designed as a secondary restraint system and has a firing threshold of around 28km/h (>17mph) and a less aggressive deployment rate. The fewer severe chest, abdominal and upper extremity injuries, therefore, might be a reflection of a superior performance of the Australian unit. Given the importance of these findings for the design of optimal airbag protection, it is imperative to examine these results further with more cases than were available here. It could be that US airbags could offer significant improvements in occupant protection if they were redesigned as a secondary restraint mechanism and used in conjunction with seatbelts (higher firing thresholds and less aggressive deployment rates). For a limited number of cases investigated in the Commodore study where the airbag was fitted but not deployed (delta-V values up to 30km/h), there were no instances of occupants sustaining injury, which suggests that the Commodore firing threshold levels are not set too low.

While small in number, there were two thoracic fractures to the spine among these occupants compared to only one lower lumbar fracture in the control cases. Minor spinal injuries included bruising and abrasions. As it was difficult to assign source of injury to these, it is unclear what may have caused them. Unfortunately, there were no other results to compare this finding with (Dalmotas did not report on spinal injuries in his study). It would be important to continue to monitor these injuries in future.

There was a relatively modest 20% increase in lower limb injuries among airbag occupants, albeit of a minor nature. Dalmotas, too, reported an increase in severe pelvic and lower extremity injuries of 58% among airbag occupants. These findings suggests that there is scope for further improvements in lower limb protection in passenger cars involved in frontal crashes.

CONCLUSION

The results of this preliminary analysis are encouraging for occupants of Australian passenger cars. Although not all statistically significant, there were substantial benefits to occupants involved in frontal crashes in Holden Commodores in terms of reduced injuries (especially those of moderate to serious AIS 2+ severities) across the range of delta-Vs where airbags are expected to be of benefit. Spinal injuries among airbag occupants warrants further investigation. In general terms, the findings from this study compared favourably with similar studies overseas, suggesting that the supplementary designed airbag used in this popular Australian passenger car may be superior in performance to its US primary restraint counterpart. There would seem to be considerable advantage internationally for extending this study to include many more cases to ensure that the findings reported here are robust.

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Technical Session 6

Side Impact and Upper Interior Head Protection

Chairperson: Dennis McLennan, Australia

FIELD STUDY ON THE POTENTIAL BENEFIT OF DIFFERENT SIDE AIRBAG SYSTEMS

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Paper Number 96-S6-O-01

SUMMARY

In the opinion of accident researchers, in side-impact collisions top priority should be given to protecting the head. However, the side airbags so far offered and installed in production cars are systems primarily dedicated to protecting the upper part of the occupant's body. Introducing a side airbag system for the head area proved to be very much more difficult. For this reason systems such as the ITS presented by BMW are appearing on the market with something of a delay compared with the thorax airbag. In order to assess the various approaches adopted for this specific form of protection system, we must define the requirements, describe the test procedures and draw up a form of "shopping list" to permit comparisons to be made.

INTRODUCTION

The development of side airbags has occupied the attention of a large number of development engineers in recent years throughout the industry which produces airbags worldwide. In 1994, Volvo introduced a seat-integrated airbag which, after being activated by a impact igniter, provides a degree of additional protection in the event of a side-impact collision, but only in the thorax area. Automobiles from Mercedes-Benz and BMW have been equipped with similar systems since the end of 1995, though in this case with larger airbags integrated into the doors. From the outset, however, most accident experts were agreed that protection of the head should be given greater priority in the event of a side impact. The initial studies and concepts, although in some cases actually shown to the public at automobile exhibitions, were soon consigned to the wastepaper basket. Two main problems proved impossible to master: the aggressive character of systems which inflate an airbag from a packed position right next to the occupant's head, could represent a high hazard risk if the head were tilted even slightly. The second problem: even if there were space for the airbag between the side contour of the vehicle's body and the occupant's head, it could not be provided with the lateral support needed to improve the occupant kinematics resulting from this type of impact.

For these reasons preference was given to developing side airbags which mainly protect the thorax area. However, publications dealing with the ITS (Inflatable Tubular Structure) head-level side airbag system from BMW has steered development once again in the direction of providing greater protection for the head. In the meantime, various head-level side airbag designs have been put forward. The purpose of these remarks is to describe the requirements which a head-level side airbag system must satisfy, in order to make a comparative assessment easier.

WHAT HAPPENS IN AN ACTUAL ACCIDENT?

The following statistical information is based on material collected by BMW Accident Research, with accidents in which at least one BMW vehicle was involved. In view of the prevailing accident report system, the accidents investigated are normally severe ones. This explains why there may be slight deviations from the results of other surveys. However, this circumstance does not have any effect on the optimization of protective measures, since it is precisely the accidents involving occupant injury which are of importance in this case.

One fifth of the accidents investigated were of the side-impact type.

Deformation Highth in Side Collision

In 39.7% of those events with another vehicle clearly identifiable as involved in the accident, the other vehicle was an automobile. In 46% the vehicle struck a tree, a substantial post or a mast. In 14.2% the other vehicle involved in the accident was in the category comprising off-road vehicles, trucks and buses (Fig. 1).

Table 1: Opponents in Side Crashes

Opponent	Percent
Car	39,7
4x4	1,8
Truck/Bus	12,4
Tree/Pole	46,1

In almost two-thirds of the accidents investigated, the other vehicle therefore intruded at a height which included the zone above the door capping.

Injury patterns in side-impact collisions

Of the severe side-impact collisions investigated which injuries were clearly attributable to the nature of the collision, only 25 % did not involve head-area injuries. In 52.7 % of collisions occurring on the side where an occupant was seated, the thorax remained uninjured. Moderately severe (AIS 3-4) thorax injuries predominate. Head injuries occur more frequently at the lower end of the AIS scale (AIS 1-2) and - particularly evident in this case - at the upper end also (AIS 5-6). Among the AIS 6 injuries, more than three times more concerned the head than the thorax.

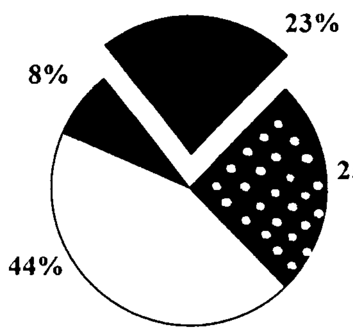
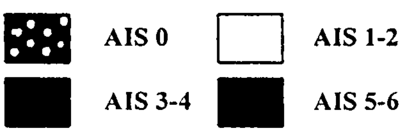


Fig. 1: Head Injuries in Side Collisions



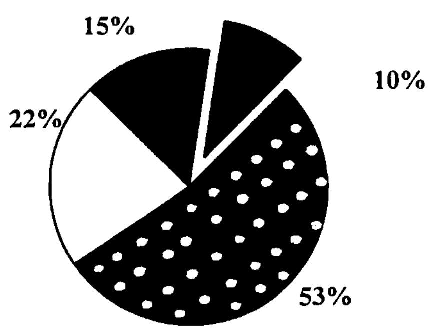


Fig. 2: Thorax Injuries in Side Collisions

These results clearly confirm the necessity for measures aimed at protecting the thorax, but also show that there is a very large and so far unsatisfied demand for efficient restraint systems to protect the head in the event of side impacts.

It is also evident that the total number of thorax and head injuries is greater than the number of cases investigated: in this upper injury zone in particular, multiple injuries occur frequently and cannot be reduced sufficiently with a single restraint system which protects either the head or the thorax.

Roll-over situations

A side airbag system capable of sealing off the side window aperture reliably, at least in part, could help to reduce the frequent tendency for the occupants to be thrown out if the car rolls over.

Position of occupants in car

Before the correct side airbag position can be decided upon, it is essential to know the occupants' actual position. In about 8% of the cases the seat was in the front third of its forward-and-back adjustment range, in about 19 % it was found to be in the rear third of the adjustment range. Most of the seat positions (72%) were in the central third of the available range of adjustment (Fig.3). If one bears in mind that details of the accident can normally only be recorded after the accident victims have been removed from the car, we can assume that in at least 80 % of all cases the seat must have been somewhere in the front and center ranges of adjustment. Furthermore, some of the rearward seat positions are certainly due to movement of the seat during rescue work.

Table 2: Seating Position after Crash

Position	Percent
Front 1/3	8,2
Mid 1/3	72,5
Rear 1/3	19,3

WHAT REQUIREMENTS CAN WE DERIVE FROM THIS SITUATION?

Top priority in protection against side-impact collisions must be given to bodyshell structural measures. The more rigid the structure remains when subjected to intrusion loads, the greater the chances of loads on the occupants remaining low. This is the only precondition in which additional protection systems such as side airbags can be utilized in a worthwhile manner. As with the driver's airbag and the frontal collision, we can say that the airbag is only of benefit if the vehicle itself possesses a high protection potential. Installing an airbag in itself is not sufficient to make a safe vehicle out of one that is less safe.

If the vehicle itself satisfies the necessary preconditions, the next step is to decide on the design and rating of the side airbag system. It is important to note that neither a single head protection system nor a system which protects the thorax area exclusively can be regarded as an ideal solution, since accident research tells us that there have been many cases of extremely severe multiple and even fatal injuries in both the head and the thorax areas. The only systems which can accordingly offer the prospect of success are those which protect both the head and the thorax. The following remarks will however discuss only the head-level airbag.

The head-level side airbag has differing and at first glance positively contradictory tasks to perform:

- It should provide an energy-absorbing cushion between the occupant's head and an intruding object such as a tree.
- It should act as a form of retaining strap and protect the head against excessive "whiplash" acceleration when an impact (mainly in the lower area of the side of the vehicle) occurs.
- It should if possible maintain its restraining function for several seconds, for example in order to offer adequate head protection in a roll-over or multiple collision situation.
- It should take up its active position with a minimum of energy being expended, so that this does not represent a hazard for the vehicle's occupants.

The airbag as an energy-absorbing cushion

In accidents of the type in which, for example, the vehicle skids and strikes a tree sideways on, the greatest risk for the occupant is for his or her head to strike the intruding object directly. Similar injury mechanisms are to be anticipated if the occupant has adopted a seat position which leads to a direct impact against part of the vehicle's interior structure in the event of a side-impact collision, for example the B-post or the upper seat belt loop. In such cases the airbag must act as an energy-absorbing "cushion", which prolongs the head acceleration path, reduces the resulting rate of head acceleration and increases the contact area over which the impact takes place. This enables both the danger of skull injuries and the risk of brain damage as a result of violent deceleration to be reduced.

In order to be able to perform these tasks, the head-level airbag must be of a certain minimum thickness and be inflated to a given internal pressure. The thickness of the inflated "cushion" is restricted by the proximity of the side body contour to the occupant's head. A value of 100 to 130 millimeters would appear to be suitable. The internal pressure is determined by the force needed to convert the energy and must be regarded as dependent on the specific airbag system; however, it should not be less than 1 bar.

The airbag as a retaining strap to reduce loads on the neck

Side-on collisions between two passenger cars normally involve one vehicle penetrating the side of the other at an impact height which does not extend above the door capping. In such cases, not only thorax injuries as a result of direct impact contact but also problems in the head and neck areas are known to occur. The high inertial mass of the head causes the upper part of the body to be "pushed away" from under it. Accident physicians have registered high loads on the necks of accident victims. In certain circumstances the angle to which the neck is bent can be so severe that the occupant's head strikes the door capping or even the engine hood of the intruding automobile.

In view of this accident pattern, a solution must be sought which prevents excessive neck angles from occurring. For just this purpose, an airbag is not absolutely essential: in the past, pyrotechnically activated curtain nets have been demonstrated which also might perform well in this single situation. It does, though, give a poor effect in other, intruding situations. It is important for the protective element itself to be mounted in a sufficiently stable manner, since the vehicle cannot provide a suitable reaction point. In normal circumstances the side windows break very early, and a system attached at one end only would be forced outwards by the occupant's head without undue force being necessary. In other words, the restraining effect must be provided without recourse to additional supports. Provided that a small gap between the head and the restraint system is assured (the minimum thickness referred to above), the ability to absorb energy can in this case be ignored, since no excessively high relative speed can build up.

The airbag as long-term protection in roll-over or multiple collision situations

If the vehicle rolls over it would be most useful to have a protection system which covers the side window area to the greatest practicable extent This would counteract the risk of being thrown out of the car, this being - in particular in the case of occupants not wearing seat belts - a frequent source of extremely severe or even fatal injuries. To satisfy this requirement, two important peripheral conditions must be fulfilled: the sensors for the dynamic system must be triggered off in a roll-over situation, and the system itself must remain active for an extended period.

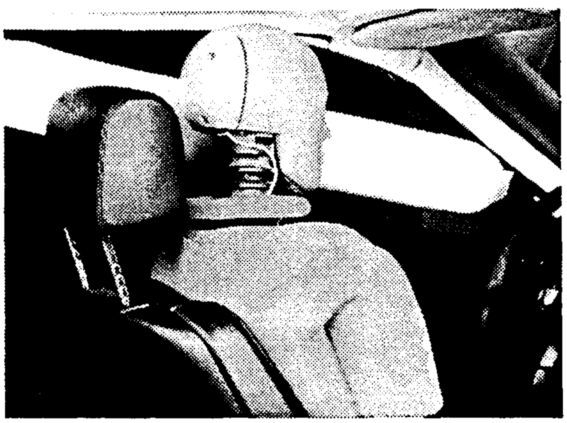


Figure 3: Side Airbag for Ejection Prevention

Existing roll-over sensor systems, for instance those used to activate automatic roll protection hoops, do not meet the reliability standards needed to operate airbags. Whereas the accidental release of a reversible roll hoop could be tolerated when the car is driven over a hum and the suspension rebounds fully, the triggering off of an airbag in this situation would certainly not be acceptable. Many teams of experts are currently working on the detection of roll-over situations at an adequate level of reliability, but there suggested solutions have no place in these remarks. The BMW accident data base contains several cases in which it is highly probable that side airbags would have been activated in a roll-over situation as a result of the lateral acceleration incurred by the car. After detailed analysis of 120 roll-overs without a subsequent collision, the accident researchers concluded that in about one-third of these roll-overs either the tipping action of the vehicle would have resulted in an increase in moment sufficiently sudden to be detected by the side airbag sensors as a side-on collision, or else severe lateral deceleration in the course of the roll-over would in all probability have triggered the system off.

These theoretical assumptions were confirmed coincidentally by an actual accident in which a 7 Series BMW equipped with two thorax side airbags rolled over several times. During this accident, both side airbags were triggered off. The sensors are designed in such a way that only the side airbag on the side where the impact occurs is activated in the event of a collision. This indicates that during the multiple roll-over movement the lateral accelerations which occurred on both the left and right sides of the body< were sufficiently high to ignite both side airbags in succession. Further work must be done on a reliable roll-over sensing system, though even with the existing acceleration sensors side airbags would be able to demonstrate their beneficial effect in certain roll-over situations.

However, in order to do this they would have to exert a restraining effect for a sufficient length of time. A roll-over lasts up to 7 seconds. We know this from reconstructing an accident in which a car overturned at a speed of 160 km/h and was not braked by any form of external obstruction. It took about this time for the vehicle to come to a standstill. Nor do multiple collisions normally last any longer than this. If a side airbag is to provide protection throughout this occurrence, therefore, we must be certain that it still provides sufficient restraining action 7 seconds after it has been ignited. This requirement cannot be met by conventional front airbags, so that neither a ventilated construction nor a permeable woven material is suitable.

The airbag must suit occupants of all sizes

The occupants of the vehicle are not always seated in a standardized position. Whereas on the driver's side we can assume that the situation is fairly normal, since the car has to be driven and the driver must therefore sit within reach of the steering wheels and pedals, more unfavorable combinations of occupant stature and seated position must be reckoned with on the passenger's side. There are clearly certain limits to what any system can achieve, and a head-level side airbag is evidently without effect if the person it has been installed to protect has the seat in a fully reclined position. However, in conjunction with the thorax airbag even a small occupant with the seat towards the rearward limit of its adjustment should still receive adequate head protection.

Inflation at the lowest possible energy level

The aggressive character of airbag systems is currently the subject of fierce discussion, particularly in the USA. When a new airbag system is developed, as this discussion implies, it is necessary to devote particular attention to reducing its aggressive character. If, furthermore, this airbag system is triggered off immediately adjacent to the head of the person it is designed to protect, the need to keep the violence of the inflation process to a minimum becomes even more evident.

The aggressive effect of an airbag can be described primarily with the aid of two measured values: unfolding speed and the distance covered as the airbag unfolds. The relationship between the position of the packed and inflated airbag is an indicator of the system' aggression. The greater the difference between these values, the worse the risk of injuries caused by being struck. The aim is therefore quite simply for the airbag to move as little as possible while it is being activated. To achieve this, one has no choice but to depart from the conventional concept of a tightly packed, spherical airbag.

WHICH TESTS SHOULD THEREFORE BE CONDUCTED?

In order to assess and test the function and performance of side airbag systems, both stationary and sled or actual crash tests can be used. The aim should be to test the above requirements in the simplest and easiest possible way, and one which is easily reproducible. Stationary tests are particularly suitable for this. The use of sled tests calls for relatively high effort and expense, for example because the intrusions represent an important factor when simulating impacts against a post. However, simulating intrusions successfully during sled tests is no simple matter. When developing head-level side airbags, it is normally better to use a complete vehicle from the outset and perform an actual crash.

Stationary tests

A new statutory test procedure has currently reached the Advanced Notice of Proposed Rulemaking (ANPRM) stage. This test, which is part of Safety Standard 201, is intended in particular to allow for the risk of head injuries, and specifies that defined targets on the vehicle contour should be struck by a free-moving head form (FMHF) at a speed of 15 mile/h. The aim of this safety standard was to reduce the risk of head injuries in collisions and roll-overs. To comply with it, energy-absorbing panels are needed on all structural elements in the side area of the vehicle. However, compliance with the standard cannot provide any protection against direct head contact with objects intruding into the vehicle or excessive neck loads as a result of hyperflexion, such as a dynamic airbag system might be expected to provide. Furthermore, it must be acknowledged that the additional padding conflicts with the need to accommodate the head-level side airbag. In awareness of this the National Highway Traffic Safety Administration (NHTSA) has expressed its readiness to test proposals for vehicles equipped with dynamic systems. One of these proposals is that the head impact speed be reduced to 12 km/h if the benefit of the dynamic system can be demonstrated sufficiently effectively.

In addition, certain requirements must be subjected to a static test when the function of a protective system is being tested. In this way, individual or combined aspects of the system's performance such as inflation time, operational period or airbag strength can be tested very rapidly and reproducibly.

Axial force

In order to achieve head restraint without any form of reaction support being available, the airbag must be highly stable. This stability is lacking in conventional airbag systems unless the force input vector coincides precisely with the mounting point. In the ITS system the structural factor is achieved by clamping the unit with a force of up to 5 kN between its two attachment points. This high axial force is easily able to withstand the resulting lateral forces.

Internal pressure

The ITS system operates at an internal pressure of approximately 1.5 bar. This high value ensures that even when struck directly by a post, and despite the short retardation distance which is available, the loads on the car's occupants can be expected to remain low.

Diameter

The head protection device should be brought as close as possible to the occupant's head, in order to keep relative speeds as low as possible. The ideal cushion thickness has proved to be 120 - 150 mm, which still enables the airbag to be accommodated easily between the head and the B-post. When inflated, it fills the gap between shoulder and head.

Long-term performance

To ensure that effective restraint is still obtained in roll-over situations, the protective system must remain available for a period of at least 7 seconds. In order to test its efficacy during this period, a method was chosen in which 7 seconds after ignition of the dynamic system the free-moving head form (as laid down in FMVSS 201) is propelled directly onto the activated protection system at a speed of 15 mile/h. The resulting HPC values are required to be well under the biomechanical limits.

Aggressive character

The active current discussion on the aggressive character of airbag systems must be taken into account by all new developments. While being activated, new systems should not lead to any additional occupant hazard. As an aid to estimating the potential danger, the unfolding speed and the distance which has to be covered while the airbag is unfolding can be measured. Whereas in the case of airbag systems designed to protect the car's occupants against injury in frontal collisions, unfolding speeds of up to 185 mph are not unknown, side airbag systems must operate at far lower speeds because of their proximity to the seat occupant's head. The absence of experience prevents a limit value for the speed of movement as the airbag unfolds from being stated, but tests with the ITS system indicate that at an unfolding speed of 30 mph no problems involving direct impact against the dummy have occurred. The distance covered as the system reaches its active position is only about 180 mm. This further reduces the risk of occupant injury.

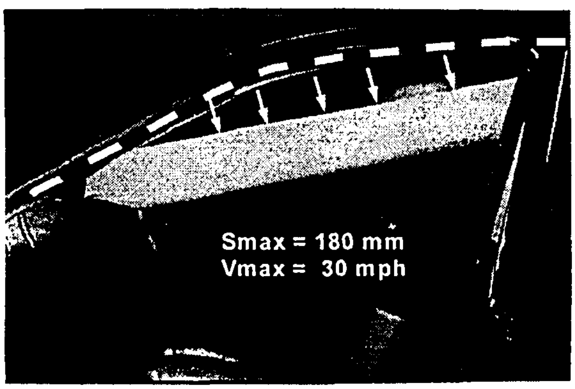


Fig. 4: Inflation Distance and Velocity of ITS

In stationary deployment tests instrumented dummies were placed under the ITS system in such a way that when it was activated it struck the dummy's head directly. In a second series of tests the dummy was inclined by a considerable amount to the side, in order to measure forces and moments at the neck when the side of the head was struck. The results of these tests are shown in the table. In no cases were results obtained which could give any cause for anxiety.

Table 3: OOP-Test Results Head leaning against B-Post, ITS Deployment against head

max Head acceleration	ay = 16,7g
max Neck moment	Mx = 15,2 Nm
max Head force	Fz = 1,1 kN

Substitute tests with sled

Here too, sled tests are the link between stationary simulations and more complex actual crash tests. It is of course necessary here to take deformation of the side frame into account, since this not only influences the unfolding of the airbag system but also determines the load on the occupants. A typical application for sled tests is for instance checking the influence of various occupant statures and seat positions. The aim is to ensure that even if restraint does not primarily take effect at the head's center of gravity, satisfactory kinematics are none the less achieved.

Fullscale crash, statutory requirements

If we look for available test criteria referring to side protection measures, we rapidly encounter relevant statutory requirements. Within the European Union, the EU95 test criterion will be applicable in the 1999 model year: it calls for a barrier weighing 950 kg and equipped with a deformation element to be forced against the test vehicle at an angle of 90° and at a point 300 mm from the ground, at a speed of 50 km/h. In the United States of America a procedure has been valid for some years in which a barrier weighing 1,368 kg with deformation element moves at approx. 54 km/h along an inclined path. Both test methods mainly measure loads on the occupants' upper bodies, with less attention being devoted to loads on the head. A further test, namely FMVSS 301, also calls for a side impact, but is intended solely for checking freedom from fuel system leaks, the vehicle's occupants being disregarded.

Fullscale crash, internal requirements

In addition to the side crash requirements laid down by law, which do not permit the performance of head-level side airbags to be assessed adequately, tests have been carried out in conditions approaching reality, and clearly confirm the advantages of the systems concerned. In the statutory side crash method, only situations involving a passenger car striking the side of the test vehicle are simulated. If a dummy validated for this procedure is used, it is possible to assess the benefit of a head-level side airbag in the form of improved head and neck kinematics.

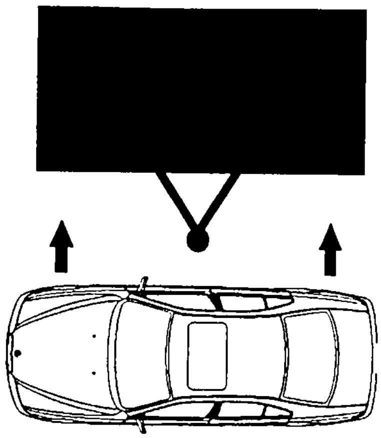


Fig. 5: Car to Rigid Pole Crash

In order to simulate direct contact with an intruding object, a new crash test procedure was employed; it is defined in a manner similar to the SAE standard (ISO/TC 22/SC 10/WG3 N100). The vehicle is pulled laterally against a fixed steel post with a diameter of 250 mm (Fig.6). The dummy is positioned such that its head cannot make contact with any internal elements such as the B-post. The post is positioned to make contact at the head's center of gravity. Vehicle speed in the region of 30 kph.

Without a head-level side airbag, the head is struck by the post at almost the full impact speed, since the vehicle's speed can only be reduced to a very small extent before this contact takes place. This basic test is complemented by a crash with head-level side airbag in use. The aim is evidently to comply reliably with the HPC head injury criterion in the vehicle's technical requirement specification.

In the case of roll-overs too, an internal company procedure is adopted in addition to the one laid down by law in the USA. It imposes new requirements in terms of the permissible loads: a helical-pattern roll-over also has to be taken into account. Whereas in the case of a "flat" roll-over the vehicle initially performs at least one rotation round its longitudinal axis, in the case of the helical-pattern roll-over it is driven onto a ramp at one side, so that a combined rotational and translational movement takes place.

"SHOPPING-LIST" FOR ASSESSING HEAD-LEVEL SIDE AIRBAG SYSTEMS

At the most recent ESV conference in May 1994, the ITS head-level side airbag system which BMW is developing was presented for the first time, whereupon a series of reports was received from the automobile industry concerning new and effective alternatives to the ITS system. They included a rebirth of the combined airbag, intended to protect both the head and the thorax. Pyrotechnically activated trap nets, aimed at preventing the head from suffering extreme rotation, "air mattresses" inflating out of the roof lining or door capping, pyrotechnically inflated roof linings and other such devices were all mentioned.

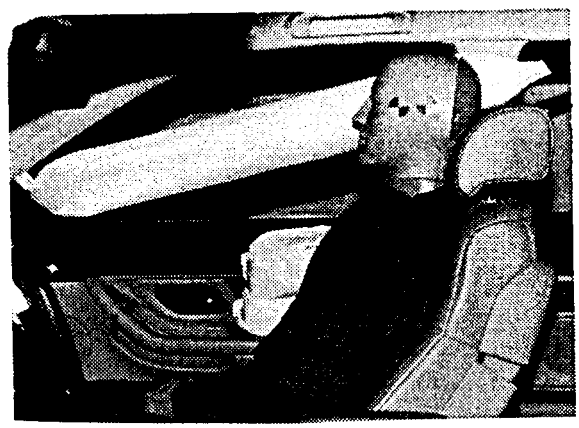


Fig. 6: BMW Side Airbag System ITS + Thorax

In order to assess these different approaches, it was clearly necessary to allow them to compete with each other and to compare the extent to which they fulfilled the tasks described above. It is important in such cases to maintain an overall view of the situation, since such protective systems contribute little to vehicle safety if they only achieve full results in a standardized situation. The effect of the various principles can be assessed on the basis of the tasks described. It is clear that the relative importance of the requirements is a more or less subjective matter, which can and indeed should be assessed critically by every observer. Furthermore, additional requirements will certainly be introduced in the course of time. This assessment should therefore be regarded as a first step towards separating the wheat from the chaff.

CONCLUDING REMARKS

The development of a restraint system to protect the occupant's head in side-on collisions is an evident necessity. Even the finest conventional side airbag cannot reduce the number of very severely or fatally injured occupants to zero. The benefit attainable from such systems will certainly be lower than with frontal airbag systems. However, their development is essential as the next important step in enhancing vehicle safety.

In view of the special situation applying to them (probability of the occupant being struck directly by an intruding object, close spatial proximity to the occupant, lack of support for the airbag and the particular sensitivity to injury of the human head), head-level side airbags must be subjected to particularly stringent selection criteria. In a direct comparison of various system principles, the ITS system would appear to come out a clear winner:

* It has the lowest initial contact force
* It stabilizes the head excellently when the impact against the vehicle takes place below the window line
* It offers optimum protection against direct head contact when an object penetrates the upper part of the vehicle
* It retains its protective effect for an unequaled period of time
* It protects a large range of occupant statures and seated positions, and
* It would appear to be the **only** system which can satisfy **all** the stated requirements in an optimum manner.

ANALYSIS OF TEST RESULTS OF SIDE COLLISIONS USING ACTUAL VEHICLES

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Paper Number 96-S6-O-02

ABSTRACT

Test procedures for evaluating the performances of a vehicle for occupant protection. during side impact have been applied in the United States since 1993, and are expected be applied in Europe beginning in 1996. In Japan, in the report of the Council for Transport Technology presented in 1992 on this subject, side impact occupant protection was adopted as a middle term study item and studies are in progress for finding an appropriate test procedure to be adopted in Japan.

This paper summaries the results of a series of full-scale side impact tests using Japanese passenger vehicles according to the US and European side impact test procedures.

In addition to tests carried out in accordance with the conditions of FMVSS214 and ECE/R.95, tests were implemented with the parameters such as crab angle, MDB height, seat position etc., were changed. The test vehicles were selected from Japanese-made 1994 models in a variety of body types, weights, etc. Dummy responses were measured in the tests, specifically, TTI and Pelvis G in the US procedure, and HPC, RDC, V*C, APF, PSPF, in the European procedure. Intrusions of the vehicle side structure, changes in velocity, were recorded.

Analysis of the test results included dummy responses, vehicle responses, and intrusions of the vehicle under a variety of tests and test conditions. In addition, analyses were made for repeatability of the full-scale tests, and repeatability in the dummy itself. The advantage and disadvantage of the US and European side impact testing procedures were studied based on the results of these analyses.

INTRODUCTION

The reduction of occupant injury to during side impact is a major subject worldwide. Extensive research into this subject is now being carried out, mainly in the US, Europe, and Japan. As a result, vigorous discussion has come out during various types of international conferences promoting international conformity such as WP29/GRSP, ISO, ESV,. However, international harmonization has not yet been attained, and in the US a new test FMVSS214 in which an actual vehicle dynamic test method has been added to the conventional static side door strength test method FMVSS214 came into effect in October 1990. On the other hand, in Europe, WP29 has included the ECE standard ECE/R.95, which incorporates an actual vehicle test method, and the EC commission is also studying an EEC directive with almost the same content as the ECE standard, with a goal of practical application in October 1998. The US and European methods use different MDB, the dummy, and the injury values.

In Japan, research into the reduction of injuries to vehicle occupants during side impact reported in a transportation technology inquiry commission report was adopted as an item for research in the intermediate term. It is expected that a proposal for a appropriate test method will be made in Japan within a few years.

This report is a consolidation of results of existing, well-known tests using actual vehicles, based on such existing conditions, as one link in the basic research for studying effective test methods relating to Japanese conditions, with the ultimate goal of reducing injuries to vehicle occupants during side impacts.

Five types of representative Japanese passenger vehicles were used in the tests, differentiated according to the vehicle dimensions and weight. These types are a medium model four-door vehicle (1.5 ton class, vehicle A), a small model four-door vehicle 1 (1 ton class, vehicle B), a small model four-door vehicle 2 (1 ton class, vehicle C), a small model two-door vehicle 1 (1 ton class, vehicle D), a small model two-door vehicle 2 (0.8 ton class, vehicle D), all of which were front engine type vehicles. A total of 24 cases of actual vehicle tests were studied using these automobiles, as shown in Table 1.

The following items were investigated.

(1) Comparison with European-US test methods
(2) Repeatability of tests
(3) Effect of MDB crab angle
(4) Effect of difference in Individual dummy
(5) Effect of seat position
(6) Effect of MDB height above ground
(7) Performances of rear-seat dummy

Table 1.

Test condition

Test Vehicle	Vehicle A		Vehicle D		Vehicle E	
Test method	ECE/R. 95	FMVSS214	ECE/R. 95	FMVSS214	ECE/R. 95	FMVSS214
Front Dummy	EUROSID-1①	SID①	EUROSID-1①	SID①	EUROSID-1①	SID①
Rear Dummy	none	SID②	none	SID②	none	SID②
Impact Velocity (km/h)	50.0	53.9	50.0	53.9	50.0	53.9

Test Vehicle	Vehicle B					
Test method	ECE/R. 95			FMVSS214		
	No. 1	No. 2	No. 3	No. 1	No. 2	No. 3
Front Dummy	EUROSID-1①	EUROSID-1①	EUROSID-1①	SID①	SID①	SID①
Rear Dummy	none	none	none	SID②	SID②	SID②
Impact Velocity (km/h)	50.0	50.0	50.0	53.9	53.9	53.9

Test Vehicle	Vehicle B					
Test method	ECE/R. 95					
	No. 4	No. 5	Front Most	Rear Most	H=300mm	Rear Dummy
Front Dummy	EUROSID-1②	EUROSID-1③	EUROSID-1①	EUROSID-1①	EUROSID-1①	EUROSID-1①
Rear Dummy	none	none	none	none	none	EUROSID-1②
Impact Velocity (km/h)	50.0	50.0	50.0	50.0	50.0	50.0

Test Vehicle	Vehicle B	Vehicle C				Vehicle D
Test method	FMVSS214	ECE/R. 95				ECE/R. 95
	Crubbed angle 0°		Front Most	Rear Most	H=300mm	H=300mm
Front Dummy	SID①	EUROSID-1①	EUROSID-1①	EUROSID-1①	EUROSID-1①	EUROSID-1①
Rear Dummy	SID②	none	none	none	none	none
Impact Velocity (km/h)	48.3	50.0	50.0	50.0	50.0	50.0

CONFIGURATION OF TESTED MATRIX

Tests with US Method

__Baseline Test__ - Tests conducted Based on FMVSS214 procedures. Four types of automobiles were used - vehicle A, vehicle B, vehicle D, and vehicle E. In addition, three test cases were studied under identical test conditions to confirm the repeatability of the tests. Each time the test vehicles were changed the tests were repeated with the same dummies in the same seat positions. A Japanese-made MDB which meets FMVSS214 requirement was used in these tests.

Effect of Crabbed Angle - The vehicle B was used to investigate the effect of crab angle, and tests were carried out, based on FMVSS214, with except crab angle. 30mph (48.3 km/hr), which is the perpendicular element for tested vehicle component in FMVSS214, was taken as the velocity of MDB. The loaded dummy had the standard configuration, and the same dummy was set in the same seat position.

Tests with European Method

Baseline Tests - Tests were conducted based on ECE/R.95 procedures. Five types of automobiles were used. In the same manner as in the US method, three test cases were studied for the vehicle B to confirm the repeatability of the tests. Also, when the test vehicles were changed the tests were repeated with the same dummies in the same seat position. An aluminum MDB made in Japan, which has same performance with UTAC (triangular pyramid-shaped) was used in these tests. The MDB was set at a height above ground of 260 mm.

Effect of Difference in Individual Dummy - In order to confirm the difference in individual dummy, the test was carried out using three EUROSID-1s. The vehicle B was used in the tests, and the test conditions were the same as in the baseline tests.

Effect of Seat Position - The test was carried out in the medium position as well as the most forward position and the most rearward position, to confirm the effect of seat position. Two types, the vehicle B and the vehicle C, were used in the tests. The dummy loading and test conditions were the same as in the baseline tests.

Effect of MDB Height above Ground - To confirm the effect of the MDB height above ground, the test was carried out at a height of 300 mm. Three types, the vehicle B, the vehicle C, and the vehicle D were used in the tests. Except MDB height, the conditions in this test were also the same as in the standard configuration.

Dummy Loaded in Rear Seat - The position in which the dummy is loaded is restricted to the front seat in the European system. However, tests were carried out using the vehicle B to confirm the performances of the dummy when loaded in the rear seat. All conditions in this test were the same as in the baseline tests except that the loading of the dummy in the rear seat was added.

TEST RESULTS

Comparison between European Test Method and US Test Method

There are many differences between European test method and US test method such as MDB configuration (dimension, structure, weight, front stiffness), dummy and injury criterion. Here, the impact performances of the test vehicle and the dummy are compared using the same vehicle model with European and US test method.

Table 2 gives comparisons of the amount of deformation for the interior and exterior of the vehicle compartment at the dummy's thoracic and pelvic regions.

This table shows that, when observing the deformation at the interior and exterior of the vehicle compartment at the front seat dummy's thoracic and pelvic regions, the European system is seen to provide a larger deformation than the US system in the thoracic region, while the same value is indicated at the pelvic region in both systems. The deformation at the interior and exterior for both the dummy's thoracic and pelvic regions in the back seat is larger in the US system than in the European system.

Table 2.

Results of intrusion

Test Vehicle			Vehicle A		Vehicle D		Vehicle E	
Test Method			ECE/R 95	FMVSS214	ECE/R 95	FMVSS214	ECE/R 95	FMVSS214
Exterior Intrusion (mm)	Front Seat	Thorax level	335	335	285	235	330	265
		Pelvis level	395	420	355	320	380	380
	Rear Seat	Thorax level	220	175	175	185	220	290
		Pelvis level	325	385	230	280	370	375
Interior Intrusion (mm)	Front Seat	Thorax level	295	240	195	145	280	155
		Pelvis level	300	300	215	180	300	230
	Rear Seat	Thorax level	130	120	90	15	90	180
		Pelvis level	160	190	135	135	245	280

Test Vehicle			Vehicle B							
Test Method			ECE/R 95					FMVSS214		
			No 1	No 2	No 3	No 4	No 5	No 1	No 2	No 3
Exterior Intrusion (mm)	Front Seat	Thorax level	350	325	305	324	307	295	280	290
		Pelvis level	400	380	380	389	382	385	375	395
	Rear Seat	Thorax level	185	165	150	189	155	215	210	220
		Pelvis level	305	295	275	312	299	380	365	380
Interior Intrusion (mm)	Front Seat	Thorax level	270	255	245	260	235	205	190	215
		Pelvis level	265	260	245	255	245	220	215	235
	Rear Seat	Thorax level	85	80	70	95	70	120	105	115
		Pelvis level	100	90	75	100	75	145	140	155

The deformation from the effect of the configuration and the stiffness of the MDB and the crab angle in the US method was almost the same from the front door to the rear door of the test vehicle. In the European method on the other hand, since the MDB is divided into six blocks, and the stiffer blocks are located close to the position of the occupant in the front seat, the deformation of the test vehicle is such that only the middle section of the MDB adjacent to the front door is intruded. However, because the stiffness in the other blocks is low, the deformation of the test vehicle is small in comparison with the middle section and, in particular, the deformation close to the position of the occupant in the back seat is small. Due to the MDB performance difference, there were some differences in the amount and mode of deformation in test vehicle.

Figure 1 shows a comparison of injury data from the dummies in the European and the US method. In this figure, a comparison is shown when the dummy injury standard values for the respective European and US methods are taken as 100%. In addition, the data for the B vehicle uses the average value of three test samples. The dummy injury value for a four-door vehicle in the European method tends to be more severe than in the US method. For a two-door vehicle, when the MDB in the European method is 260 mm above ground, the results of the US method tend to be more severe than those of the European method. However, in a later-described test in which the MDB is 300 mm above the ground, the injury value for both methods is the same, or higher in the European method in same cases. The percentage of the dummy injury values exceeding the standard value. and the percentage of each injury value exceeding the standard value in the European and the US tests are shown in Figure 2. From this figure the results of this study indicate that the percentage of thoracic injury values which exceeds the standard value is higher in the European method than in the US method. Also, when the injury values are observed individually, it is showed that an extremely high percentage of the thoracic injury values in the European method exceeds the standard value.

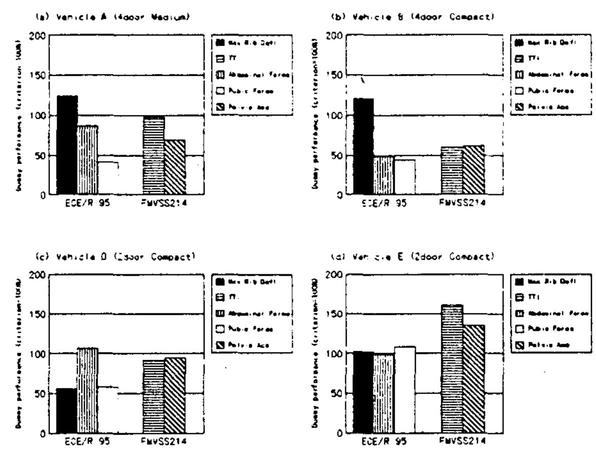


Figure 1. Comparison of dummy performance.

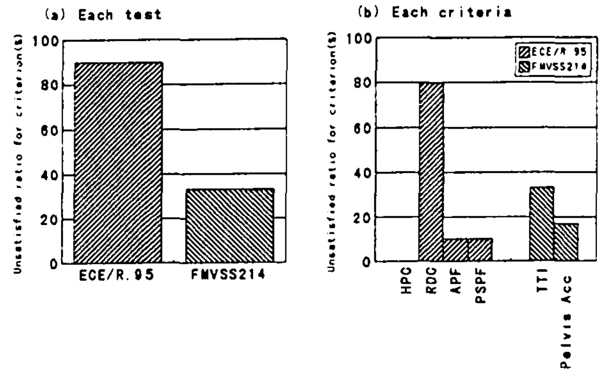


Figure 2. Comparison of unsatisfied ratio for criterion.

Investigation of Repeatability

Table.3 shows the variation in the collision velocities and in the impact. For the variation in the target impact velocities, the C.V. value at the maximum was 0.66% (where C.V. value is defined as the standard deviation divided by the average of the test results). Also, the variation in the target impact positions is within plus-minus 20 mm in the European method and within plus-minus 45 mm in the US method. These are in a range which adequately satisfies the acceptable ranges required by both the European and the US test methods.

Table 4 gives the deformation at the interior and exterior of the vehicle compartment for both the dummy's thoracic and pelvic regions. Table 5 gives the acceleration of the vehicle body, the maximum acceleration of the MDB, and their degree of variation. From this table , the variation in the deformation at the interior and exterior of the compartment of the car body was small at the thoracic and pelvic regions of the seated dummy in both the European and US methods. The car body acceleration had a C.V. value of 1.40% in the European method and 1.99% in the US method, while the maximum acceleration of the MDB had a C.V. value of 0.59% in the European method and 5.30% in the US method. The maximum acceleration on the test vehicle and the MDB shows a larger variation in the US method than in the European method. Figure 3 shows the variation in the dummy data. The following observations were made from this figure.

Table 3.

Impact velocity and discrepacy of impact point.

(a) Impact velocity and discrepacy of impact point

Test Vehicle		Vehicle A		Vehicle D		Vihicle E	
Test Method		ECE/R 95	FMVSS214	ECE/R 95	FMVSS214	ECE/R 95	FMVSS214
Impact Velocity (km/h)		50.4	54.2	50.3	53.4	50.7	54.0
Discrepacy of impact point (mm)	Horizontal	0	Front45	0	Front5	0	0
	Vertical	0	0	Upper10	0	0	0

Test Vehicle		Vehicle B					
Test Method		ECE/R 95			FMVSS214		
		No 1	No 2	No 3	No 1	No 2	No 3
Impact Velocity (km/h)		50.8	50.4	50.5	53.7	53.7	53.9
Discrepacy of impact point (mm)	Horizontal	Rear20	0	0	Front5	Front10	Front25
	Vertical	Upper10	Upper5	Upper5	0	0	Upper10

Test Vehicle		Vehicle B					
Test Method		ECE/R 95					
		No 4	No 5	Front Most	Rear Most	H=300mm	Rear Dummy
Impact Velocity (km/h)		50.0	50.0	50.1	49.9	50.0	50.0
Discrepacy of impact point (mm)	Horizontal	0	0	Rear5	0	0	Rear10
	Vertical	0	Upper5	0	Upper5	Upper5	0

Test Vehicle		Vehicle B	Vehicle C				Vehicle D
Test Method		FMVSS214	ECE/R 95				ECE/R 95
		Crabbed angle 0°		Front Most	Rear Most	H=300mm	H=300mm
Impact Velocity (km/h)		48.5	50.3	49.9	50.2	49.9	50.1
Discrepacy of impact point (mm)	Horizontal	0	Rear20	0	Rear10	Rear5	0
	Vertical	Upper5	Upper5	Upper10	Upper10	Upper5	Upper5

(b) Variance of impact velocity and impact point

	Impact Speed (km/h)		Impact Point Discrepacy (mm) Horizontal		Impact Point Discrepacty (mm) Vertical	
Test Method	ECE/R 95	FMVSS214	ECE/R 95	FMVSS214	ECE/R 95	FMVSS214
x	50.2	53.8	4.2	15.0	4.2	1.7
σ	0.29	0.25	6.54	15.55	3.72	3.73
3σ	0.87	0.76	19.63	46.64	11.16	11.18
C.V. (%)	0.58	0.47	-	-	-	-

Table 4.

Results of intrusion (Vehicle B, Repeatablity)

Test Method			ECE/R.95 No. 1	No. 2	No. 3	No. 4	No. 5
Exterior Intrusion (mm)	Front Seat	Thorax level	350	325	305	324	307
		Pelvis level	400	390	380	389	382
	Rear Seat	Thorax level	185	165	150	189	155
		Pelvis level	305	295	275	312	299
Interion Intrusion (mm)	Front Seat	Thorax level	270	255	245	260	235
		Pelvis level	265	260	245	255	245
	Rear Seat	Thorax level	85	80	70	95	70
		Pelvis level	100	90	75	100	75

Test Method			FMVSS214 No. 1	No. 2	No. 3
Exterior Intrusion (mm)	Front Seat	Thorax level	285	280	290
		Pelvis level	385	375	395
	Rear Seat	Thorax level	215	210	220
		Pelvis level	380	365	380
Interion Intrusion (mm)	Front Seat	Thorax level	205	190	215
		Pelvis level	220	215	235
	Rear Seat	Thorax level	120	105	115
		Pelvis level	145	140	155

Table 5.

Acceleration of Vehicle and MDB C.G. of vehicle B test (Repeatability)

	Vehicle Acceleration (G)		MDB C.G. Acceleration (G)	
Test Method	ECE/R.95	FMVSS214	ECE/R.95	FMVSS214
No. 1	12.3	17.0	-13.8	-14.1
No. 2	12.2	18.9	-13.9	-14.8
No. 3	11.9	19.2	-14.0	-14.4
No. 4	11.2	-	-12.8	-
No. 5	11.9	-	-13.1	-
X	11.9	18.4	-13.5	-14.4
σ	0.38	0.97	0.48	0.29
3σ	1.15	2.92	1.44	0.88
C.V. (%)	3.23	5.30	-3.54	-1.99

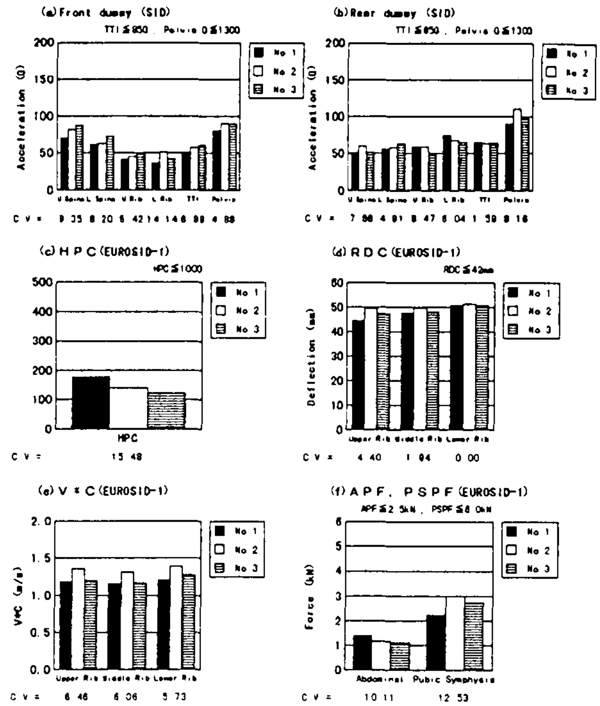


Figure 3. Comparison of dummy performance (Repeatabiliry).

<u>European Method</u> - The thoracic displacement V*C shows a maximum C.V. value of 6.46% or less, while the HPC, PSPF, and APF exceed 10%.

<u>US Method</u> - The spine (lower, upper), rib (upper), TTI, and pelvis G from the front seat dummy show a maximum C.V. value of 9.35% or less, while that for the rib (lower) exceeds 10%. From the rear seat dummy, the C.V. value was 10% or less.

When the European and US method are compared, the variation in the thoracic region was smaller in the European method while the variation in the pelvic region was smaller in the US method.

Effect of Crabbed Angle

Table.6 shows a comparison of the amount of deformation at the interior and exterior of the vehicle compartment for the dummy's thoracic and pelvic regions, Figure 4 shows a comparison of the injury values of the dummy, in the case of witn and without a crab angle.

The amount of deformation at the interior and exterior of the vehicle compartment for the dummy's thoracic and pelvic regions tends to be larger when the test is conducted without crab angle than with crab angle. However there is a tendency that the intrusion of body is about same in both cases.

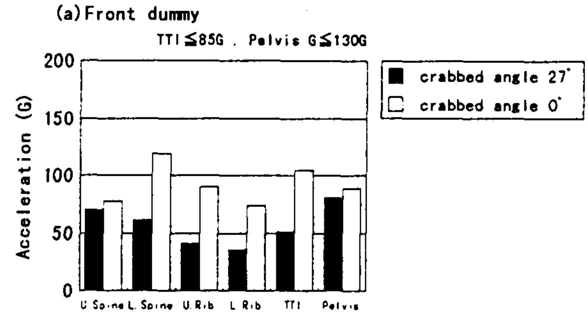
Front Seat Dummy - The acceleration and TTI for the lower spine and ribs (upper, lower) are greater in the case of without crab angle than with crab angle. The acceleration at the upper spine and pelvis was the same in both cases.

Rear Seat Dummy - The acceleration values for the spine, rib, and pelvis, and the TTI were smaller in the condition without crab angle than in the case with crab angle.

Measurements showed that the injuries to the front seat dummy were more severe in the condition without crab angle than in the case with crab angle. This indicated that comparing European and US method, even there are differences in the MDB, the dummies, only the crab angle is considered, European method without crab angle is more severe condition for front dummy than US method with crab angle.

Table 6.
Results of Intrusion
(Vehicle B, Difference for crab angle)

Test Method			FMVSS214			
			No. 1	No. 2	No. 3	Crabbed angle 0°
Exterior Intrusion (mm)	Front Seat	Thorax level	285	280	290	285
		Pelvis level	385	375	395	400
	Rear Seat	Thorax level	215	210	220	165
		Pelvis level	380	365	380	350
Interior Intrusion (mm)	Front Seat	Thorax level	205	190	215	210
		Pelvis level	220	215	235	235
	Rear Seat	Thorax level	120	105	115	90
		Pelvis level	145	140	155	110



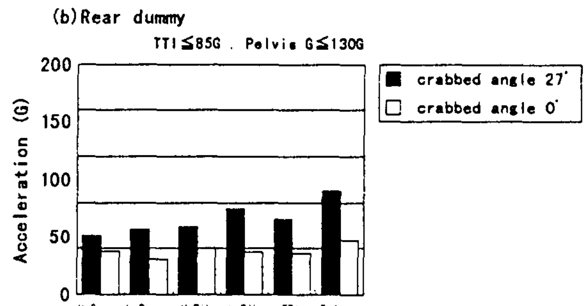


Figure 4. Results of dummy performance (Difference for crab angle).

Effect of Dummy Individuality

The variations of three test case are +0.8 km/h in the impact speed, 20 mm rear ward and 10 mm upper ward. These results adequately satisfy the variation range specified in the test method, therefore these conditions for comparing differences in each individual dummy can be considered adequate.

Figure 5 shows the variation of dummy injury values. In the test results from the effect of different dummies, the C.V. value for the HPC shows a small variation of 1.66%, while the APF and PSPF show a large variation exceeding 10%. In addition, the variation in the RDC for the thoracic is a maximum of 7.8% while the maximum V*C exceeds 10%.

The degree of variation in the dummy data from repeated confirmation tests was compared with the variation of the dummy data from comparative tests for different in dummies are shown in Figure 6. In the HPC and PSPF the variation for repeated tests was large, while for other regions the variations in all tests for different dummies were large.

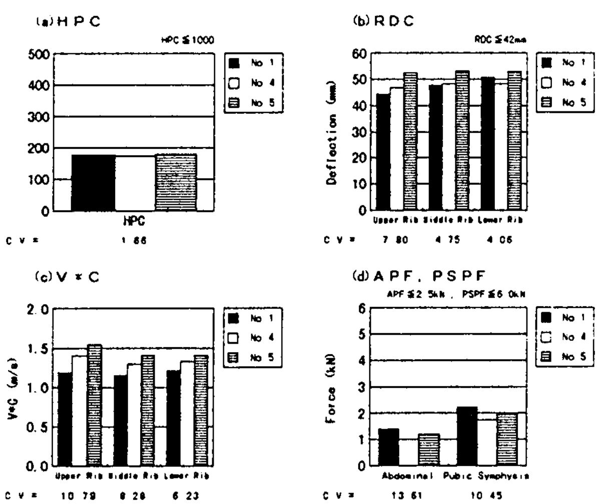


Figure 5. Comparative dummy performance (Dummy individuality).

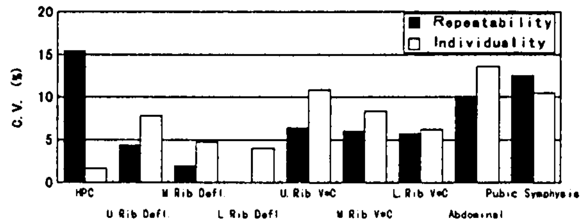


Figure 6. Comparison of C.V. between repeatability and dummy individuality.

Effect of Seat Position

The results for dummy injury values for each seat position of the B vehicle are shown in Figure 7, and C vehicle are shown in Figure 8. When the seat position is at the most forward end, the injury values in the thoracic region and the RDC and V*C are clearly low. At the most rearward and the neutral positions, most severe seat positions for each injury measuring point is different. There are also cases which show opposite result in other vehicle model. The clear conclusion for seat position was not defined due to those reason. From the results above, the dummy injury values are clearly low in the most forward seat position. in the case of most rearward position, even compared with mid position, most severe seat position was not defined.

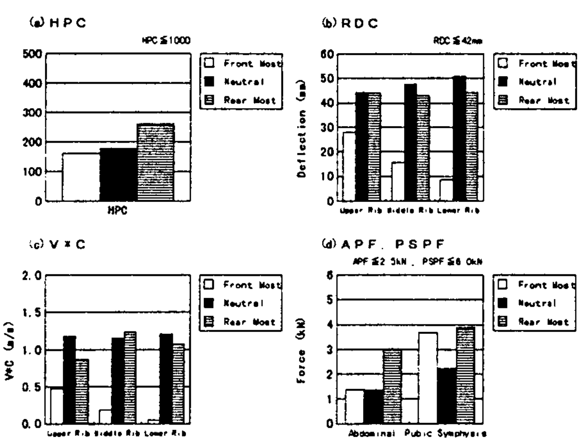


Figure 7. Comparison of dummy performance (Difference for seat position, Vehicle B).

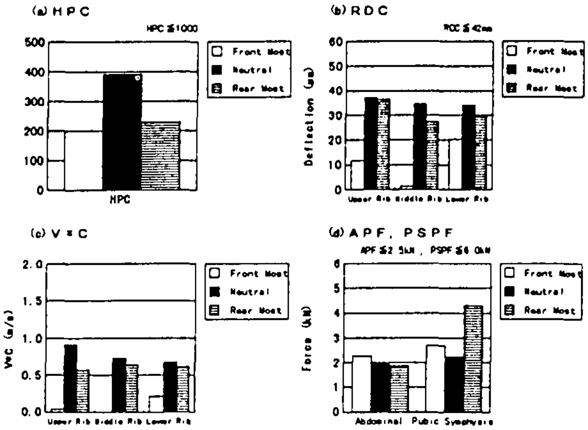


Figure 8. Comparison of dummy performance (Difference for seat position, Vehicle C).

Effect of MDB Height above Ground

The amount of deformation at the interior and exterior of the vehicle compartment for the thoracic and pelvic regions for each dummy is given in Table 7. This table shows that in the case of a four-door vehicle the deformation at the interior and exterior of the vehicle compartment for the thoracic and pelvic regions is larger when the MDB is 300 mm above ground. In the case of a two-door vehicle, the amount of deformation for the pelvic region shows same value in both cases, and the deformation is larger for the thoracic region when the MDB is 300 mm above ground.

A comparison of the results for dummy injury values at each height of the MDB are shown for the B vehicle in Figure 9, while the results for the C vehicle and the D vehicle are shown in Figure 10 and Figure 11 respectively. In these figures, for both the four-door and the two-door vehicles, the values are larger in all regions when the MDB is 300 mm above ground than when the MDB is 260 mm above ground, with the exception of the APF. The APF showed the same value for both heights.

Consider the amount of displacement in the vehicle and the injury value of the dummy, it is conducted that the condition for the MDB at 300 mm above ground is more severe than that at 260 mm.

Table 7.

Results of intrusion (Difference for MDB height)

Test Vehicle			Vehicle B		Vehicle C		Vehicle D	
Test Method			ECE/R 95		ECE/R 95		ECE/R 95	
MDB Hight (mm)			260	300	260	300	260	300
Exterior Intrusion (mm)	Front Seat	Thorax level	350	351	329	353	285	319
		Pelvis level	400	401	411	395	355	354
	Rear Seat	Thorax level	185	215	94	97	175	210
		Pelvis level	305	356	320	270	230	242
Interior Intrusion (mm)	Front Seat	Thorax level	270	295	250	280	195	240
		Pelvis level	265	270	280	300	215	200
	Rear Seat	Thorax level	85	115	65	65	90	115
		Pelvis level	100	125	190	150	135	135

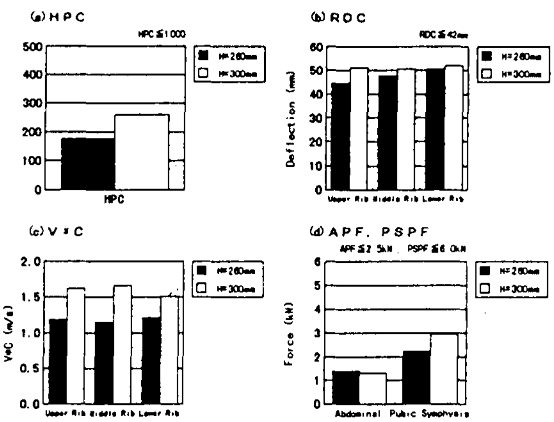


Figure 9. Comparison of dummy performance (Difference for MDB height, Vehicle B).

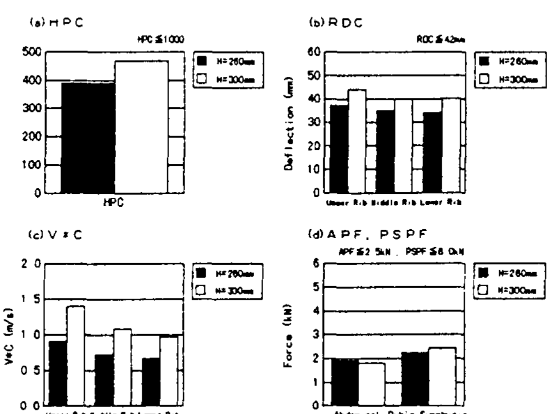


Figure 10. Comparison of dummy performance (Difference for MDB height, Vehicle C).

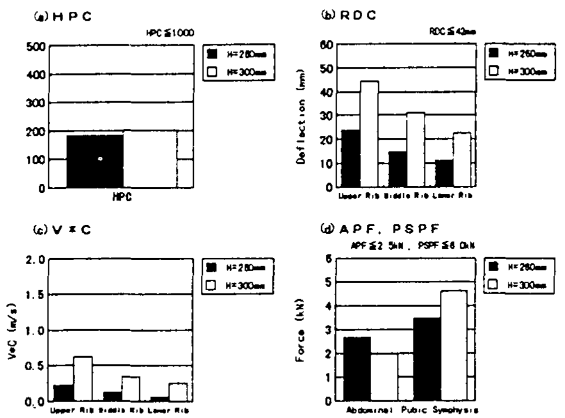


Figure 11. Comparison of dummy performance (Difference for MDB height, Vehicle D).

Loading of Dummy in Rear Seat

Comparisons of dummy injury values when the dummy is in the front seat and in the rear seat are shown in Figure 12. The dummy injury values, with the exception of the PSPF, show a tendency to be lower for the rear seat dummy than for the front seat dummy. In particular, this tendency is seen to be very clear at the thoracic and pelvic regions. The PSPF values for front and rear seat dummies are about same and much lower than the standard value.

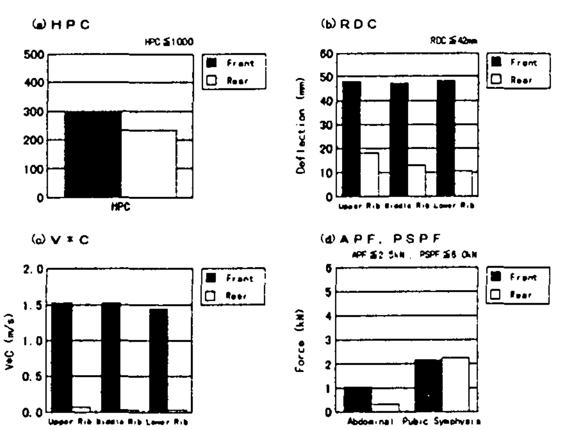


Figure 12. Comparison of dummy performance (Front and rear dummy, Vehicle B).

CONCLUSION

When the European and US test methods are compared, no great difference can be seen in the repeatability of the two methods. The dummy injuries show a tendency to be more severe in the European method than in the US method. Also, the thoracic region injury values in the European system are the most severe. The amount of deformation close to the seated position of the front seat dummy in the European method was larger than US method.

As the results of tests carried out with crab angle, seat position, height of MDB above ground, repeatability of tests, and difference in the each Individual test dummy as parameters which can affect the test results, the following can be stated. Test method without crab angle causes severe injuries to the front seat dummy, while the test method with crab angle causes severe injuries to the rear seat dummy. The neutral seat position or the most rearward seat position causes severe injuries to the dummies, but it cannot defined which position causes the most severe injuries since result is different in the type of the test vehicle. The MDB at a height of 300 mm causes higher injury values and more deformation than at a height of 260 mm. Also, the injury value level for the rear seat dummy is extremely small compared to that of the front seat dummy, and all injury values are extremely small with respect to the standard value in the European method. The repeatability of the test results from the test vehicles is about the same for both methods. The variation in the difference in each Individual dummy is larger than the variation in the dummy injury values in repeated test results.

REFERENCE

NHTSA, "Federal Motor Vehicle Safety Standards; "Side Impact Protection"". 49 CFR Part 571.

NHTSA, "Side Impact Testing Procedure".

ECE/R.95, "Uniform Provision Concerning the Approval of Vehicles with Regard to the Protection of the Occupants in the Event of a Lateral Collision".

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SIDE IMPACT PROTECTION OPPORTUNITIES

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ABSTRACT

The level of side impact protection achievable by passenger vehicles is examined in a series of crash tests performed following US FMVSS 214 dynamic testing protocols, but substituting EuroSID 1 and BioSID crash test dummies in place of the US SID. The vehicles included a Honda Accord modified by Transport Canada, a production Ford Contour and production versions of the Volvo 850 with and without the Sipsbag.

In addition to satisfying the criteria referenced in US and European regulations, the modified Honda Accord, the Ford Contour and the Volvo 850 fitted with the Sipsbag, all showed high levels of protection to the abdomen based on the responses measured with the BioSID. All three vehicle models limited abdominal penetration on the driver side to under 39 mm and the peak VC value to well below 1 m/s.

The results obtained in this series of tests suggest that innovative padding schemes represent viable alternatives to side-mounted air bag systems as a means of further improving occupant safety in side impacts.

INTRODUCTION

Regulatory developments in both the US and in Europe in the area of side impact protection have been followed closely by Transport Canada. To assist Transport Canada in deciding its course of regulatory action on occupant protection in side impacts, a major research programme was started in 1988. To date, over 40 vehicle crash tests have been carried out as part of this programme.

The initial objective of this programme was to assess the appropriateness of test procedures developed in the US and Europe in the context of the Canadian vehicle mix and side impact accident problem. The Canadian tests showed vehicle performance rankings provided by the European test procedure differed from those provided by the US test procedure and that these differences could be attributed largely to the different loading induced by the two different moving deformable barrier (MDB) designs [1]. Of the two moving barrier designs, the US version was found to produce vehicle deformation patterns and dummy responses which were more consistent with those observed in vehicle-to-vehicle tests.

With the selection of a suitable test procedure and MDB, the focus of Transport Canada's research efforts turned to identifying what would constitute the most appropriate combination of test dummy and performance criteria. In a cooperative research programme with NHTSA, a series of vehicle crash tests was conducted by Transport Canada to generate comparative dummy response data using both the production version of the EuroSID, designated as the EuroSID 1 and the BioSID. Assessments of vehicle performance, provided by both the EuroSID 1 and BioSID, and based solely on regulated US performance requirements, were not significantly different from those provided by the US SID dummy [2]. This finding was in agreement with that of an earlier test programme conducted by NHTSA [3]. The Canadian test programme, however, also showed that all of the vehicle tests, even those which produced exceptionally low TTI values, exceeded biomechanical injury thresholds based on deflection and force. Other researchers observed that assessments of vehicle performance derived using acceleration-based criteria often conflicted with those derived using deflection-based criteria[4]. This was particularly true with respect to issues such as armrest design[5,6].

To address the various issues raised above, Transport Canada initiated a modified vehicle programme using vehicle models tested earlier as part of the joint Transport Canada/NHTSA research programme. The possibility that assessments of vehicle performance using the US SID could lead to inappropriate armrest designs was confirmed early in the programme. However, the programme also indicated that all acceleration criteria referenced in the US regulation and all deflection and force criteria referenced in the European regulation could be satisfied through the use of interior padding schemes which controlled the timing and phasing of the load paths developed between the occupant and vehicle [3]. Consequently, the deficiencies of the existing US regulation could be overcome by substituting a dummy capable of supporting deflection and force measurements in the Canadian side impact regulation.

The scope of Transport Canada's modified vehicle programme was subsequently expanded to include the testing of production vehicles that incorporated innovative padding schemes or other technologies such as side-mounted air bag systems. This was done to quantify the levels of side impact protection achievable using current technology. The initial findings of this on-going programme are summarized in the present paper.

ACHIEVABLE PERFORMANCE PROGRAMME

In Phase I of Transport Canada's modified vehicle programme, the front and rear door assemblies of three vehicle models were modified. The modifications took the form of padding added to the surface of the interior door panel. A 100 mm thick pelvic pad was added to each door assembly. In addition to providing force attenuation, the purpose of the pelvic pad was to control the timing and location of load transfers between the occupant and the intruding door structure. The objective was to achieve very early engagement of the pelvis, when the door velocity is greatest and to sustain this loading until the door is decelerating. At that point, the loading is partially transferred to the shoulder and then to the thorax. Depending on the vehicle, shoulder and chest pads were also added. When added, the shoulder pad was stiffer than the chest pad. The stiffness differential between the shoulder pad and chest pad served to create a buffer zone to minimize the likelihood of any secondary "punch" to the thorax or abdomen.

Table 1.
US and EEVC Regulatory Limits
Transport Canada Target Values

Criterion (Units) / Filter Class	Peak Values: Suggested Regulatory Limits	Peak Values: TC Target Levels
Thoracic Rib Deflections (mm) / SAE 180	42	< 35
Thoracic Rib V*C Values (m/s) / SAE 60	1.0	< .50
Abdominal Rib Deflections * (mm) / SAE 180	39	< 20
Abdominal Rib V*C Values * (m/s) / SAE 60	1.0	< .25
Sum of Abdominal Forces (N) /SAE 600	2,500	< 1,250
Pubic Force (N) / SAE 600	6,000	< 6,000
Pelvic Acceleration (g) / FIR	130	< 130
TTI ** (g) /FIR	85/90	< 85

* BioSID only (substitute for EEVC abdominal force criterion). Currently not a regulatory requirement.
**85 g limit applies to 4-door vehicles, 90 g limit applies to 2-door vehicles.

The base test condition consisted of a BioSID dummy in the driver position and a EuroSID 1 in the rear left passenger position. Except for the substitution of dummies, crash testing followed US FMVSS 214 testing protocols. All tests were performed with each dummy's arm in the down position.

The results obtained in this initial series of tests, previously presented at the 14th ESV conference, were very encouraging[3]. In addition to satisfying all European and US regulatory limits, a reduction in the average value of each injury index was achieved by the modified vehicles. The reductions ranged from a high of 84 percent in the case of the abdominal VC value (BioSID only) to a low of 9 percent in the case of the average pubic force value. In terms of US requirements, the reductions in the average TTI and average peak pelvic lateral acceleration were 14 and 25 percent respectively.

Modified Honda Accord Series

Among the vehicles modified in the above programme was a 1992 Honda Accord. This vehicle model was chosen to serve as a platform to further refine the padding scheme under Phase II of the modified vehicle programme. The primary objective of the Phase II programme was to reduce the level of encroachment on interior space by the added padding while still achieving the targeted levels of performance described in Table 1. The modifications were carried out in stages. First, the thickness of the pelvic pad was reduced from 100 mm to 75 mm in both seating positions and a 25 mm thick shoulder pad was added. The second test of the vehicle produced peak response values very close to those observed in the first test. As expected, the reduction in the thickness of the pelvic pad produced modest increases in a number of the peak response values observed in the second modified vehicle test. All thoracic rib deflections remained below 35 mm, while abdominal deflections remained below 30 mm. Slight increases in the peak pelvic acceleration and peak pubic load values were also observed on the driver side. However, all US and proposed EEVC requirements were satisfied.

Subsequent modifications were carried out on 1994/95 versions of the Honda Accord. This was prompted by the fact that the 1994 Accord had been redesigned by Honda. In addition to styling and structural changes, the production version now incorporated a number of design features similar to the modifications implemented by Transport Canada in the 1992 modified Accord series. The changes included the addition of a pelvic pad in the form of a partitioned thin-walled plastic shell and a flatter armrest design. This redesign facilitated implementation of several planned changes. Accordingly, the new production Honda Accord was restested and additional tests were carried out to adapt the 1992 modifications over to the 1994 version.

The final modifications made by Transport Canada to the 1994 Honda Accord consisted of replacing the plastic pelvic shell with a pad formed from low density expanded polypropylene (EPP) foam, the substitution of a foam armrest which was hollowed out to collapse under direct loading, and the addition of a shoulder pad, again using EPP foam. All of the foam pads were recessed and integrated into the original interior door contours of the production Accord. The density of the foam used for the pelvic and shoulder pads was reduced from that used in earlier tests. The modified front door assembly is illustrated in Figures 1 and 2, while the modified rear door assembly is illustrated in Figures 3 and 4.

Table 2.
Driver ATD (BioSID) Response Summary
Modified Honda Accord

ATD Response / Filter	Peak Value
Rib Deflections (mm) / SAE 180:	
Upper Thoracic Rib	28.7
Middle Thoracic Rib	21.5
Lower Thoracic Rib	16.7
Upper Abdominal Rib	12.6
Lower Abdominal Rib	12.0
Rib V*C Values (m/s) : / SAE 60	
Upper Thoracic Rib	0.25
Middle Thoracic Rib	0.12
Lower Thoracic Rib	0.08
Upper Abdominal Rib	0.07
Lower Abdominal Rib	0.09
Pubic Force (N) / SAE 600	3,290
Pelvic Acceleration (g) / FIR	79.0
TTI (g) / FIR	70

Note : V*C values based on a half-thorax dimension of 140 mm

The responses measured on the driver side are summarized in Table 2. In addition to all US and proposed EEVC requirements being satisfied, all of Transport Canada's target values were met also. The targeted levels of performance were achieved in the rear seating position as well. Consequently, the initial objectives of the Phase II programme had been achieved.

Although initially it was planned to carry out similar modifications on the remaining two vehicle models tested in Phase 1 of the programme, this was not pursued. The need for such a programme was eliminated by the introduction of production vehicles which already incorporated similar padding schemes. Accordingly, it was felt that the Phase II programme objectives could be served equally well, if not better, by testing vehicles with "advanced' padding schemes and vehicles fitted with other innovations such as side-mounted air bag systems.

Ford Contour Test Series

The 1995 Ford Contour was selected for testing as it featured a padding scheme very similar to that employed in Transport Canada's modified vehicle programme. The padding scheme employed in the Contour also makes use of a pelvic pad and a shoulder pad (Figure 5). In addition, it incorporates a "pusher" pad mounted on the inside of the metal door skin at the height of the interior pelvic pad. As the door is crushed, the pusher pad comes into contact with the pelvic pad, pushing the latter inwards to facilitate early engagement of the pelvis. It should be noted that this padding scheme applies only to the front doors.

Three tests were carried out on the Ford Contour. In the first two tests, the BioSID was seated in the driver's position and the EuroSID 1 was seated in the left rear passenger's position. The seating order was reversed in the third test. Other than for dummy differences, testing followed FMVSS 214 testing protocols.

From the data presented in Appendix A, it can be seen that all US and EEVC requirements were satisfied in the driver position in all three tests. Although the padding arrangement places the pelvis in the primary load path developed between the vehicle and occupant, the pelvic accelerations and pubic forces were all exceptionally low. While the peak driver abdominal deflections values measured in the BioSID tests (24.0 to 37.3 mm) can be seen to be higher than those achieved in the above modified Honda Accord test (under 13 mm), they are among the lowest recorded by Transport Canada in production vehicles.

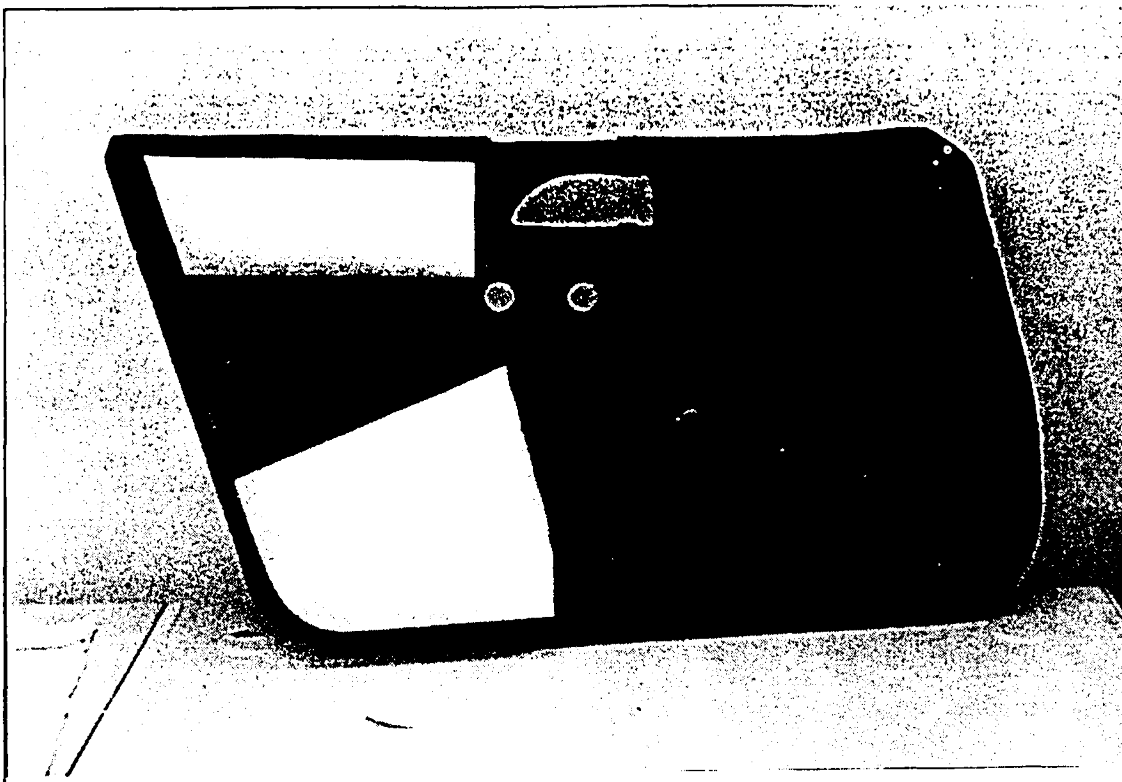

Figure 1. - Modified Front Left Honda Accord Door Assembly (Viewed From Inside)

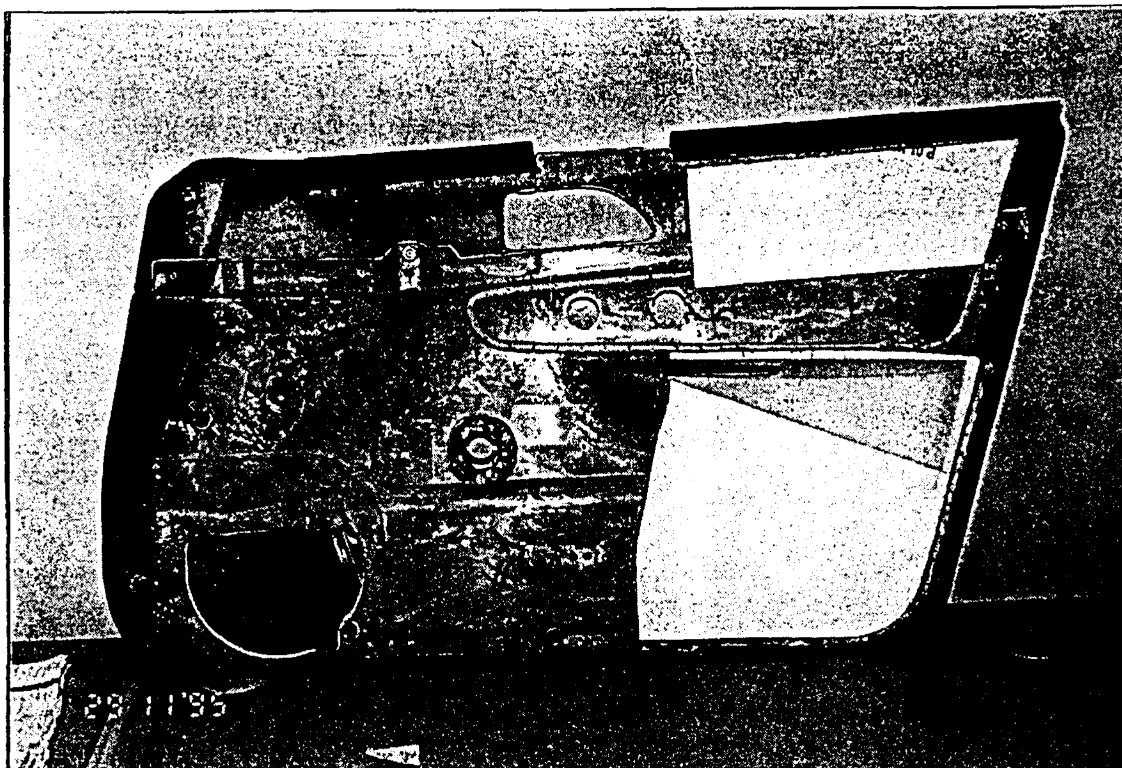

Figure 2. - Modified Front Left Honda Accord Door Assembly (Viewed from Outside)

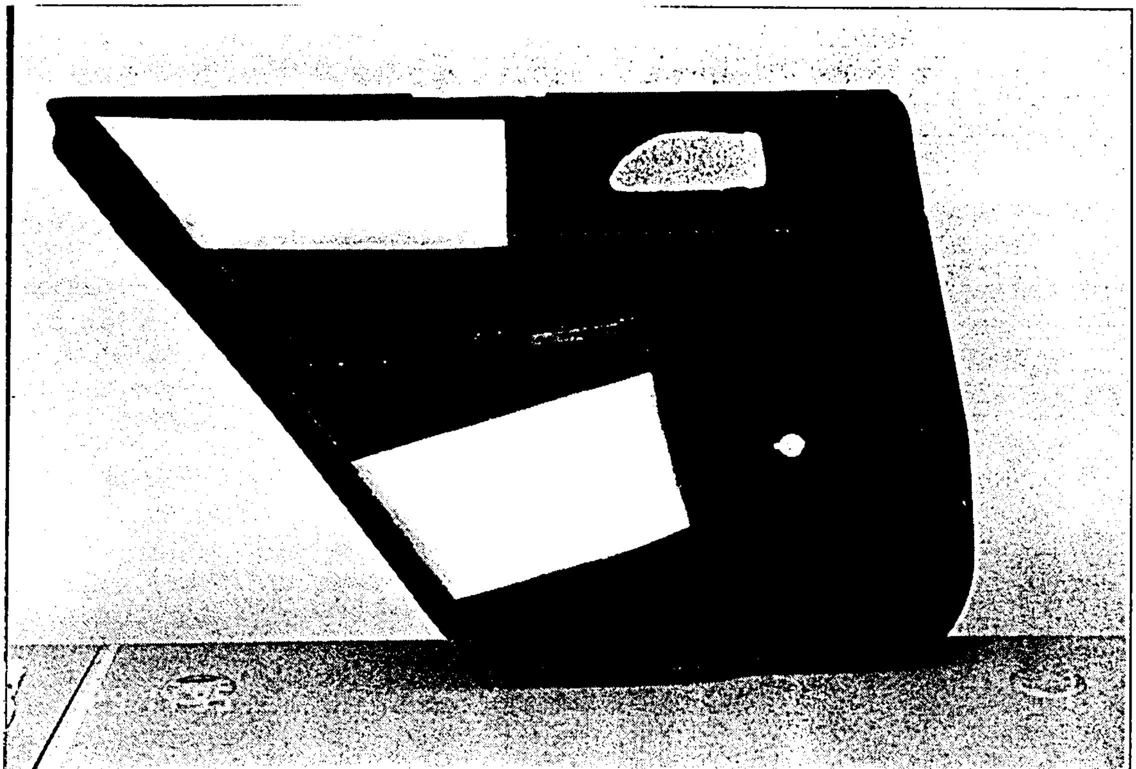

Figure 3. - Modified Rear Left Honda Accord Door Assembly (Viewed From Inside)

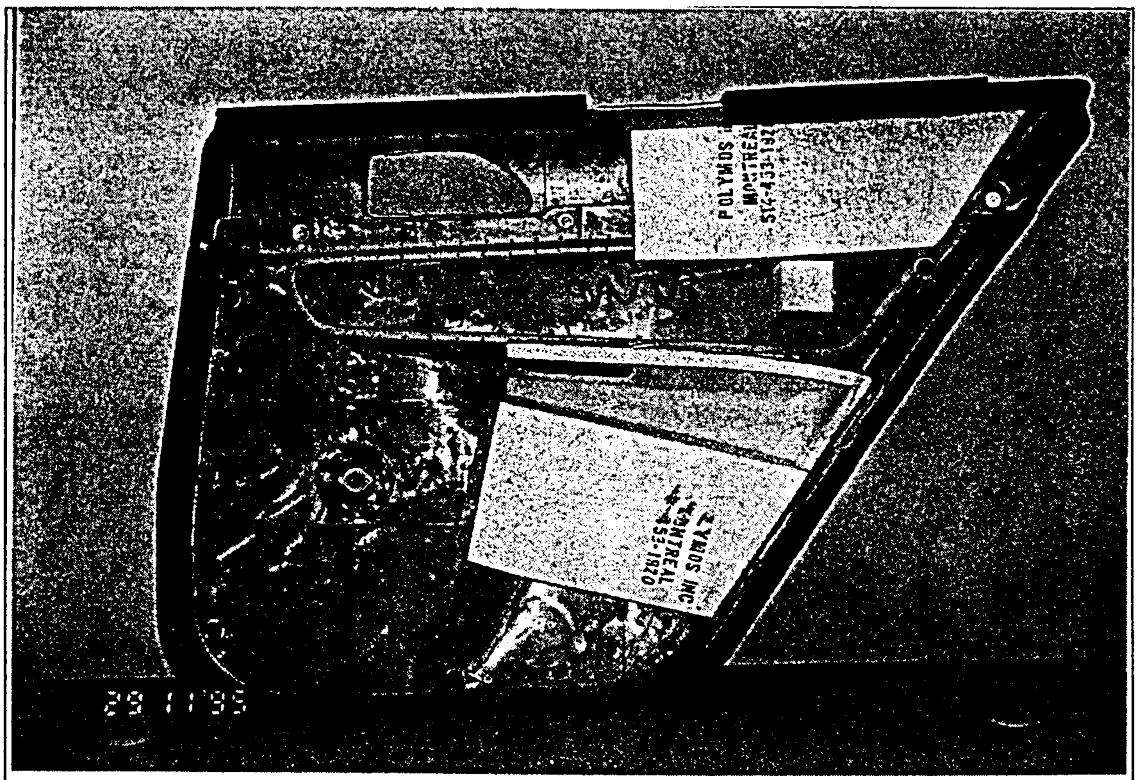

Figure 4. - Modified Rear Left Honda Accord Door Assembly (Viewed from Outside)

Table 3.
Comparison of Driver ATD Responses
1995 Ford Contour : BioSID vs. EuroSID 1

ATD Response / Filter	Peak Values BioSID (TC95-203)	Peak Values EuroSID (TC95-204)
Rib Deflections (mm) / SAE 180:		
Thoracic Rib (Any)	29.4	31.7
Abdominal Rib (Any)	37.3	#N/A
Rib V*C Values (m/s) / SAE 60 :		
Thoracic Rib (Any)	0.20	0.41
Abdominal Rib (Any)	0.42	#N/A
Abdominal Force (N)/ SAE 600	#N/A	<2,000
Pubic Force (N) / SAE 600	1,954	2,701
Pelvic Acceleration (g) / FIR	55.3	64.7
TTI (g) / FIR	72	78

Table 3 presents a comparison of the driver responses obtained with the BioSID and EuroSID 1 dummies on the driver side. As two tests with the BioSID were conducted, the BioSID results presented in Table 3 reflect the values obtained for the test which most closely matched the EuroSID 1 test in terms of barrier alignment at time of impact. There was close agreement between dummies in terms of US performance indices (TTI and peak y-axis lateral acceleration) and rib deflection. Differences were more pronounced for VC and pubic force values. However, the latter peak values were far removed from the associated limit values.

From Appendix A, it can further be seen that there was also general agreement in peak dummy response values between the two BioSID tests. In the driver's seating position, the most significant difference was in the peak abdominal deflection (25.4 mm vs. 37.3). The difference is likely attributable to the deformation pattern of the B-pillar. The lower value was observed in the test where the displaced B-pillar maintained a near vertical profile. The higher value was observed in the test where the B-pillar was bent inwards resulting in approximately 25 mm greater intrusion into the passenger compartment.

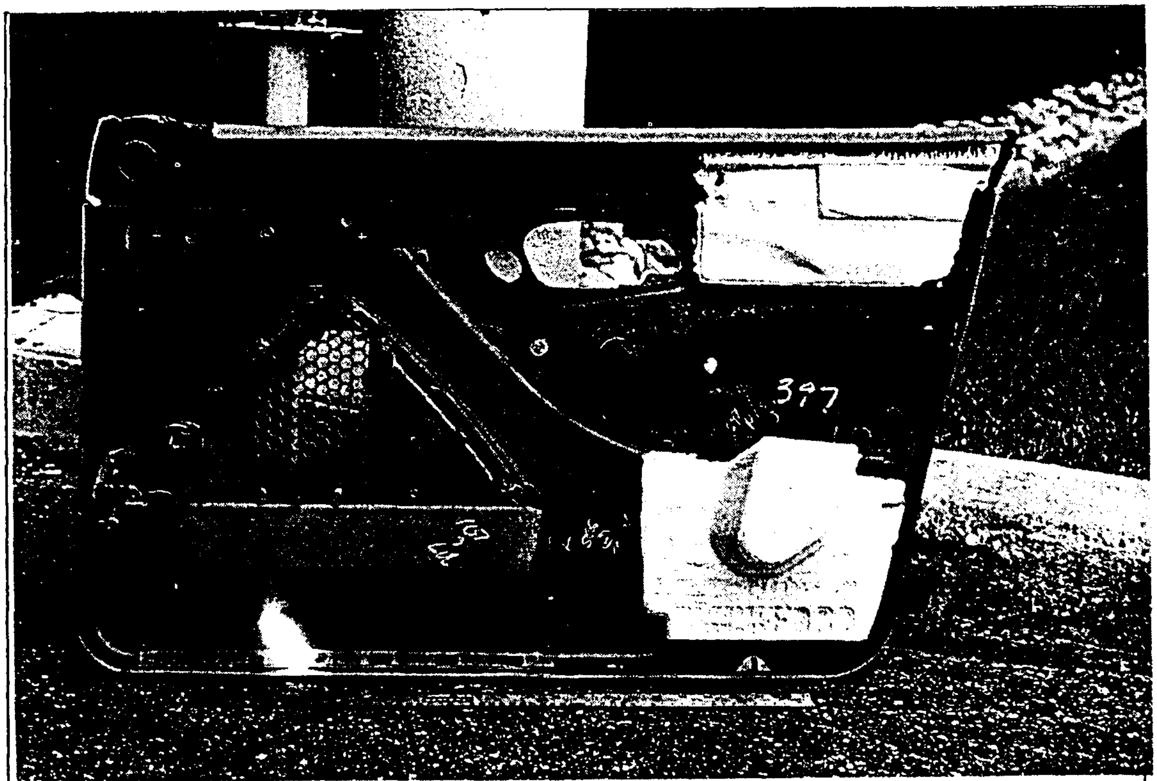

Figure 5. - 1995 Ford Contour Front Left Door Assembly

Table 4.
Comparison of Driver ATD Responses
Volvo Test Series

Maximum Thoracic Deflection (BioSID)

Bullet Vehicle	Peak Values - mm (SAE 180) SIPS Only	SIPS with Sipsbag
US MDB/54 km/h/Crabbed	29.3	18.6
Taurus/50km/h/Not Crabbed	24.8	21.9

Maximum Abdominal Deflection (BioSID)

Bullet Vehicle	Peak Values - mm (SAE 180) SIPS Only	SIPS with Sipsbag
US MDB/54 km/h/Crabbed	43.7	25.9
Taurus/50km/h/Not Crabbed	49.6	30.6

Maximum Pubic Force (BioSID)

Bullet Vehicle	Peak Values - N (SAE 600) SIPS Only	SIPS with Sipsbag
US MDB/54 km/h/Crabbed	2,292	1,958
Taurus/50km/h/Not Crabbed	3,896	2,572

TTI (BioSID)

Bullet Vehicle	Peak Values - g (FIR) SIPS Only	SIPS with Sipsbag
US MDB/54 km/h/Crabbed	73	65
Taurus/50km/h/Not Crabbed	72	58

Volvo Test Series

Tests of the Volvo were included as part of the Phase II programme to examine levels of side impact protection achievable through structural reinforcement and through the use of a side-mounted air bag system. A secondary objective was to evaluate the ability of the US moving deformable barrier (MDB) to support meaningful assessments of vehicle performance in the case of vehicles with highly modified side structures and/or fitted with side-mounted air bag systems

A total of four Volvo 850 vehicles were tested in the programme. Two of the vehicles were fitted solely with the Side Impact Protection System (SIPS), an integrated package of structural reinforcements designed to limit passenger compartment intrusion in a side impact. The other two vehicles were fitted with both the SIPS and the SIPS air bag system referred to as the "Sipsbag". Each of the two vehicle configurations was tested once following the US testing protocols and once in a vehicle-to-vehicle impact. In the latter cases, a Ford Taurus was used as the bullet vehicle in a pure perpendicular impact with the bullet vehicle travelling at 50 km/h.

From the findings presented in Table 4, it can be seen that all of the thoracic rib deflections in the Volvo test series were very low, under 30 mm. The lowest deflections were obtained with the Sipsbag in the test with the US MDB where the maximum thoracic rib deflection was below 19 mm, the lowest value recorded by a production vehicle in Transport Canada tests. Moreover, from the additional data provided in Appendix A, it can also be observed that in both tests with the Sipsbag, the loads applied to the thorax were very well distributed. In each of these two tests the maximum difference between the deflections across the three thoracic rib assemblies was no greater than 0.6 mm.

The deflections measured by the abdominal rib assemblies were higher than those measured by the thoracic rib assemblies in all 4 tests. The abdominal rib deflections in the vehicle-to vehicle tests were higher than those measured in the MDB tests in the test pairings with and without the Sipsbag. This is consistent with the vehicle damage pattern produced in the vehicle-to-vehicle tests. The latter tests produce more localized loading of the Volvo side structure above the height of the H-point of the dummy. The levels of intrusion in the MDB tests were less pronounced and the intrusion profiles were much flatter.

Both tests with the Sipsbag produced substantially lower peak abdominal deflections. In addition to providing force attenuation at the level of the abdomen, it is likely that the Sipsbag also functions as a shoulder pad which limits the magnitude of the loads transmitted to the abdomen as the door intrudes inward. This is also suggested by the peak pubic loads which can be seen to be closely correlated with the peak abdominal deflection values. Again, controlling for the Sipsbag, it can be seen that the vehicle-to-vehicle tests produced higher peak pubic load values than the MDB tests. Similarly, controlling for the type of bullet vehicle, it can be seen that the tests with the Sipsbag produced lower peak pubic force levels than tests without the Sipsbag.

The TTI values observed in the Sipsbag tests can also seen to be lower than those observed in the tests without the Sipsbag. However, the lowest TTI value was recorded in the vehicle-to-vehicle test with the Sipsbag, an outcome inconsistent with the chest and abdominal deflection data. As noted earlier, loading environments which involve localized deformation of the abdominal region frequently produce low TTI values. This again highlights the need for the capability to monitor the response of the abdomen in any dummy employed to regulate side impact protection.

DISCUSSION

The findings of the present study are encouraging in that they indicate that substantial improvements in side impact protection can be achieved, at minimal added cost and with little encroachment of interior space, through the use of innovative interior padding schemes. While the utility of the US SID and TTI remain the subject of debate, it is nevertheless encouraging that padding schemes can be devised which, in addition satisfying all deflection and force criteria, can be expected to comply with the kinematic requirements imposed by US regulations. Earlier tests conducted by Transport Canada demonstrated that vehicle models designed without the aid of advanced dummies such as the BioSID frequently showed exceptionally low TTI values when tested with the US SID but exceptionally high abdominal deflection, VC, and force values when tested with either the EuroSID or BioSID under identical test conditions. The need for a regulatory dummy capable of supporting deflection and force criteria will become all the more important in the near future as competitive pressures are likely to force vehicle manufacturers to accelerate the development and fitment of side air bag systems.

All three vehicle models evaluated in the present study showed exceptionally good performance in the highly idealized loading environment produced in a regulatory-type test. Each design tested outperformed the others in one or more aspects. The modified Honda Accord produced the lowest abdominal deflection and VC values, while the Ford Contour produced the lowest pubic force and pelvic acceleration values. The Volvo Sipsbag produced the lowest chest deflection and VC values. The uniformity of deformation pattern across the thorax achieved with the Sipsbag was particularly impressive. As most of the peak response values were far removed from injury threshold values, the above differences between vehicles, at best, reflect margins of safety which may not translate necessarily to reduced levels of injury risk. An occupant who experiences 15 mm of abdominal penetration is at no greater risk of injury than an occupant who experiences 10 mm of abdominal penetration. At higher collision severities, the deformation pattern of the side structure may render any margin of safety indicated in a regulatory test totally irrelevant. On the other hand, if the assessments of performance provided by BioSID and EuroSID 1 are accurate, then it is clear from the test results that each of the above vehicles can be expected to afford a very high level of occupant safety in a side impact at collision severities at or below that represented in these tests.

In order to gain addition insight on the relevance of performance measures obtained in regulatory-type tests, Transport Canada recently initiated a directed study to examine the injury experience of occupants in side impacts at collision severities just above, at, or below that represented by current regulatory practices.

Accident reconstructions with different dummies will also be carried out The new data and the findings obtained already from previous crash test programs will be used to arrive at a final decision on which combination of dummy and injury criteria will be employed in Canada to regulate side impact protection.

ACKNOWLEDGMENTS

The authors would like to acknowledge the significant contribution made by staff of the Motor Vehicle Test Centre. Special thanks also goes to Jim Bain of Transport Canada for his assistance in securing the vehicles and test hardware used in this programme.

The opinions expressed and conclusions reached in this paper are solely the responsibility of the authors and do not necessarily represent the official policy of Transport Canada.

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WORKING TOWARDS A HARMONISED DYNAMIC SIDE IMPACT STANDARD - AN AUSTRALIAN PERSPECTIVE

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Paper Number 96-S6-O-05

ABSTRACT

This paper reviews the differences between the US and European regulations and outlines the Australian Federal Office of Road Safety's research program to examine the likely benefits of allowing a third alternative which combines the better features of the two regulations and promotes the use of the BioSID dummy.

INTRODUCTION

After frontal impact crashes, side impacts are the greatest killers of vehicle occupants on Australian roads, accounting for over 25% of fatalities.

Australian Design Rule (ADR) [1] 29/00 - Side Door Strength was introduced in 1977 to provide side impact crash protection. Australia was the only country outside North America to introduce this design requirement.

In May 1995, the Federal Office of Road Safety released for comment a draft ADR for dynamic side impact protection which allows compliance to be demonstrated to either the US Federal Motor Vehicle Safety Standard 214 or the Economic Community for Europe Regulation 95. These two regulations were developed on either side of the Atlantic during the 1980s and early 1990s.

Although their intent is the same (to improve side impact protection), their detailed requirements are quite different. Both use a mobile trolley with a deformable face to impact the car being tested. However, the mass of the trolley, specification of the deformable face, test speed, the test dummy and injury criteria are different. While Australian crashed vehicle studies have shown that head and neck injuries are prevalent locally, head injury is only addressed in the European regulation.

The current situation has forced manufacturers to "fine tune" their designs to ensure compliance with the US or European regulations, depending on the market into which the vehicle is sold. Manufacturers around the world have indicated general support for a single harmonised standard to which the car is designed.

CURRENT OVERSEAS REGULATIONS

As noted previously, there are <u>two</u> fundamentally different test procedures and test dummies implemented or planned for dynamic side impact regulation in the US and Europe. There is no work currently being undertaken to harmonise the two regulations.

US Standard FMVSS 214

In September 1993, the US government mandated their revised FMVSS 214 regulation incorporating a dynamic side impact test using their Side Impact Dummy (SID). It allows for phased introduction for all manufacturers with a 10% requirement for the first year (1994 models), a 25% requirement for 1995 models, a 40% requirement for 1996 models and a 100% requirement for 1997 models and beyond. The major components of the US dynamic test specified in regulation FMVSS 214 comprise:

- a moving trolley of 3010 lbm (1365 kg),
- a homogeneous deformable barrier face,
- a crabbed barrier impact angle of 27 deg,
- a barrier impact speed of 33.5 mph (54 km/h), and
- SID dummies in the front and rear near-side seats.

<u>Trolley Configuration</u> - FMVSS 214 calls for the impacting trolley to be "crabbed" at 27 degrees and to strike the test vehicle at a travel speed of 33.5 mph (about 54 km/h). This is illustrated in Figure 1. The crabbed configuration was important to simulate real world intersection crashes where both vehicles are moving. This was subsequently confirmed in comparative crash tests undertaken by Transport Canada[2] using North American vehicles.

<u>US Deformable Barrier Face</u> - The US barrier construction is essentially homogeneous with a protruding bumper layout as shown in Figure 2. The main section is constructed from 45 psi (±2.5 psi) honeycomb material with the bumper section in 245 psi (±15 psi) aluminium honeycomb material. Because of its size and test arrangement, the US feel that it is less likely to over-emphasise side door strength(3). NHTSA argued that the force and deflection characteristics of the barrier are important when simulating real world crashes. They noted that their barrier is considerably stiffer than the European barrier and that the latter has experienced problems in earlier testing obtaining reproducible results.

The bottom edge of the US barrier is 280 mm from the ground. The bumper element is 330 mm above the ground and the barrier is 1676 mm wide.

The SID Dummy - The US regulation calls for tests involving the Side Impact Dummy (SID) developed by the NHTSA. SID is a modified Hybrid 2 model developed specifically for side impact testing after extensive cadaver testing in the US and Germany. Its biofidelity requirements led to unequal masses in the dummy, especially its relatively soft arms which was intended to incorporate rib characteristics.

SID has been shown to be less sensitive to door padding stiffness than EuroSID or BioSID, due mainly to its construction and injury criteria. For example, when examining padding selection for a particular structure using the three existing side impact dummies (and their respective injury criteria), both EuroSID and BioSID showed that 10 psi material gives optimal performance. While 10 psi would also be optimal for SID, so too would any material from 10 to 40 psi(3).

US Injury Criteria - In developing SID, measurement of deflection forces was difficult because of rotation, therefore acceleration of the thorax and lower spine became the major injury criteria. This has since become a criticism of SID, both outside and inside the US. Delta-V distributions from NASS showed that the 50th percentile was somewhere between 15 and 20 mph which was subsequently adopted as the design speed.

The injury criteria are limited to peak lateral pelvis acceleration and the Thoracic Trauma Index (TTI(d)) where:

$$TTI(d) = \tfrac{1}{2} (G_R + G_{LS})$$

G_R = greater of either upper or lower rib accelerations
G_{LS} = lower spine (T12) peak acceleration

The SID dummy criteria was based on hard thorax injuries including liver and kidney injuries but not soft tissue injury in the abdomen. There is no instrumentation available for measuring these injuries other than those covered by rib acceleration. Cavanaugh et al (1993) proposed an additional injury criteria of Average Spine Acceleration (ACA) was claimed to overcome some of the insensitivity of SID. So far, this has not been adopted.

SID has no provision for specifying any head injury criteria. US accident data shows that the greatest source of severe injury in side impacts is to the head, not the thorax. Thus, it is argued, FMVSS 214 does not really address the major source of injury from side impacts. The US have issued a revision to FMVSS 201 which is effectively an upper interior padding standard for side rails and A- and B-pillars aimed at addressing at least part of these head injuries from side impacts.

Impact Point - FMVSS 214 requires the front edge of the impacting barrier to strike the test vehicle at a point dependent on the wheelbase (W) of the vehicle:

- 37 inches (940 mm) forward of the centre of the vehicle's wheelbase, if W < 114 inches (2896 mm),
- 20 inches (508 mm) rearward of the front axle centreline if W < 114 inches.

The majority of cars available in Australia fall into the first category.

ECE Regulation 95

A different dynamic side impact test procedure and injury criteria have been incorporated into a new United Nations ECE regulation which been introduced for new models manufactured after 1st October 1995. The test procedure was developed by the European Experimental Vehicle Committee (EEVC) and the major components of the dynamic test specified in ECE Regulation 95 comprise:

- a moving trolley of 950 kg (2090 lbm),
- a non-homogeneous deformable barrier face,
- a perpendicular barrier impact,
- a barrier impact speed of 50 km/h (30 mph), and
- EuroSID dummy in the front near side seat only.

Trolley Configuration - A trolley mass is 950 kg which was about the average mass of European vehicles at the time it was developed. There was very little effect observed in testing different masses up to 1100 or 1300 kg because most of the peak loads occur between 35 and 50 msecs and the trolley mass has little influence at that time. The mass of the trolley certainly influences the amount of intrusion but has less effect on dummy performance compared to peak loading.

European Deformable Barrier Face - The European barrier design comprises of six blocks (3 on the top and 3 on the bottom which slightly protrude) which they claimed effectively represent the stiffness values of impacting passenger car front structures, ie front longitudinals, engine etc. These values were derived from French testing of representative European passenger car crashes against a rigid barrier wall. Subsequent testing of Japanese cars in Japan showed that these cars also correlated well with these European force characteristics. The barrier face is 1500 mm wide (see Figure 3).

The height of the barrier has been somewhat controversial. Originally, it was set at 300 mm from the ground surface to the lower edge and practically all development work involved in ECE Regulation 95 has been based on this barrier height. This was slightly above the bottom edge of the US barrier (280 mm) but below the US barrier's bumper height of 330 mm. Representations by a few European member countries led to the barrier height being lowered to 260 mm when Regulation 95 was first issued. However, the EEC directive for dynamic side impact protection has recently been finalised with a barrier ground clearance of 300 mm and it is expected that ECE R95 will revert to this figure in the near future.

Impact Speed and Direction - A perpendicular impact configuration was chosen because some European manufacturers believed this configuration offered best protection to occupants of their vehicles in real world accidents. A perpendicular impact was also the simplest testing option and did not appear to compromise safe vehicle design.

Early tests by the AAMA compared crabbed with perpendicular impact configurations did not show a lot of difference in performance. This was because of the mass of the dummy and the difference in striking direction did not seem to have much effect during the first 35 msecs when the injury effects of side impact collisions are at their maximum. This was also confirmed by Canadians when they crashed vehicles in both crash configurations. It was pointed out, however, that this is somewhat dependent on the type of vehicle, the dummies on-board and the effects on the rear seat passengers. One manufacturer noted the need to take action to improve rear dummy performance when the test configuration was crabbed because the barrier has a tendency to slide along the car towards the rear.

An impact speed of 50 km/h was chosen for the standard based on the distribution of impact speeds observed in real world accidents in Europe.

Canadian tests compared both barriers in crashes to North American vehicles and felt that the US barrier was slightly more representative of US vehicle crashes, particularly those involving MPV's. European tests claim that the European barrier reproduced quite well the worst case outcomes for a European vehicle fleet.

EuroSID Dummy - The Europeans felt that there was a need for a more sensitive measuring instrument and injury criteria in side impacts than that offered by SID. As a result, they set about developing EuroSID, a joint exercise involving several European countries and different test. While EuroSID has arms, the specification calls for them to be out-of-the-way during impact to minimise their protective role for the chest.

Dummy Test Criteria - European studies had shown that the most severe injuries in side impacts were to the head, thorax, abdomen and pelvis, so EuroSID was required to detect injuries in these areas.

Head Injury Criteria (HIC) was considered adequate for measuring head injury. For the chest, the Europeans felt that TTI was not appropriate for measuring these injuries and subsequently adopted chest deflection and Viscous Criteria (V*C). Appropriate values of this parameter were determined for EuroSID (European tests showed that a V*C of 1 = 30% to 40% probability of injury for AIS3 or above). Concern has been expressed by some about the repeatability of the Viscous Criteria with the EuroSID dummy so it has been agreed to just record the readings for the first 2 years of the regulation without it being considered as a pass/fail criterion.

Regulation 95 also has abdominal and pelvic injury criteria which limit the peak abdominal and pubic symphysis force as measured by EuroSID.

Impact Point - The EEVC did recommend dummies in both the front and the rear seating positions on the struck side only. However, it seems that most of the development work has been done with only a front seat dummy on-board (the back seat has tended to be heavily loaded with instrumentation and cameras). ECE has subsequently dropped the requirement for a rear dummy in the proposed regulation. Given that the impact of the barrier is centred on the "R-point" of the vehicle, this seems to be quite sensible as it presents a rather strange crash profile for the rear dummy. It should be remembered that the European barrier is 176 mm narrower than the US one.

EXAMINING THE FEASIBILITY OF A HARMONISED STANDARD

Australian field data also indicated that side impact crashes caused head, thoracic, abdominal and pelvic injuries. Therefore any harmonised standard from Australia's view needed to address these injuries.

The first stage of the research program was to conduct crash tests to the following requirements using a vehicle model understood to comply with FMVSS 214:

- US FMVSS 214
- ECE Regulation 95
- A hybrid regulation described below.

The two tests based on current regulations will be conducted in full accordance with test procedures set out for FMVSS 214 and ECE Regulation 95. Unfortunately, none of the tests were completed in time for the deadline of this paper. It is hoped to present some preliminary data during the ESV presentation.

HYBRID SIDE IMPACT TEST

The hybrid dynamic side impact procedure included the following features:

- BioSID dummies in the front and rear outboard seating positions on the impacted side.
- FMVSS 214 crabbed trolley with ECE R 95 deformable barrier element.
- FMVSS 214 impact geometry.
- ECE Regulation 95 injury criteria to the degree which BioSID is capable of recording.

This test configuration was chosen for the following reasons:

- FMVSS 214 crabbed barrier better reproduces a typical intersection side impact crash.
- FMVSS 214 test configuration requires countermeasures for both front and rear seat occupants.
- BioSID is generally considered to be the more biofidelic dummy.
- ECE R 95 barrier face better represents a vehicle front structure.
- ECE R 95 injury criteria more fully covers the range of injuries seen in side impact crashes.

US experience confirms that the benefits of having a rear dummy are really quite small since occupancy rates, like Australia, are quite low. Cost benefit analysis would be hard pressed to justify the need for a rear seat dummy. It should be noted that performance standards will not necessarily guarantee rear seat protection without a rear seat dummy and a separate impact test involving a more rearward impact location.

For this project, a rear BioSID dummy will also be used and the benefits determined. Because the US barrier is wider, there is expected to be benefits for smaller cars where the crush profile will encompass the rear seating position.

TEST VEHICLES

Ford EF2 Falcon Gli sedans (wheelbase 2791 mm) were used for all the tests. This vehicle was chosen because it is a high volume local produced vehicle and Ford had claimed that it met the requirements of FMVSS 214.

The vehicles were purchased through the Department of Administrative Services (DAS) government fleet contract. All test vehicles were fitted with driver's side airbags, automatic transmission, front bucket seats, and air-conditioning which was the standard DAS fleet specification.

Seats and trim were removed from the non-impacted side as required to install data acquisition equipment etc. The vehicles were ballasted as necessary to the requirements of the particular test procedure.

METHODOLOGY TO EXAMINE BENEFITS

A study [4] was undertaken by the Monash University Accident Research Centre on behalf of the Federal Office of Road Safety to estimate the benefits likely to accrue from adoption of a side impact ADR which will allow demonstration of compliance using either the US or ECE regulation.

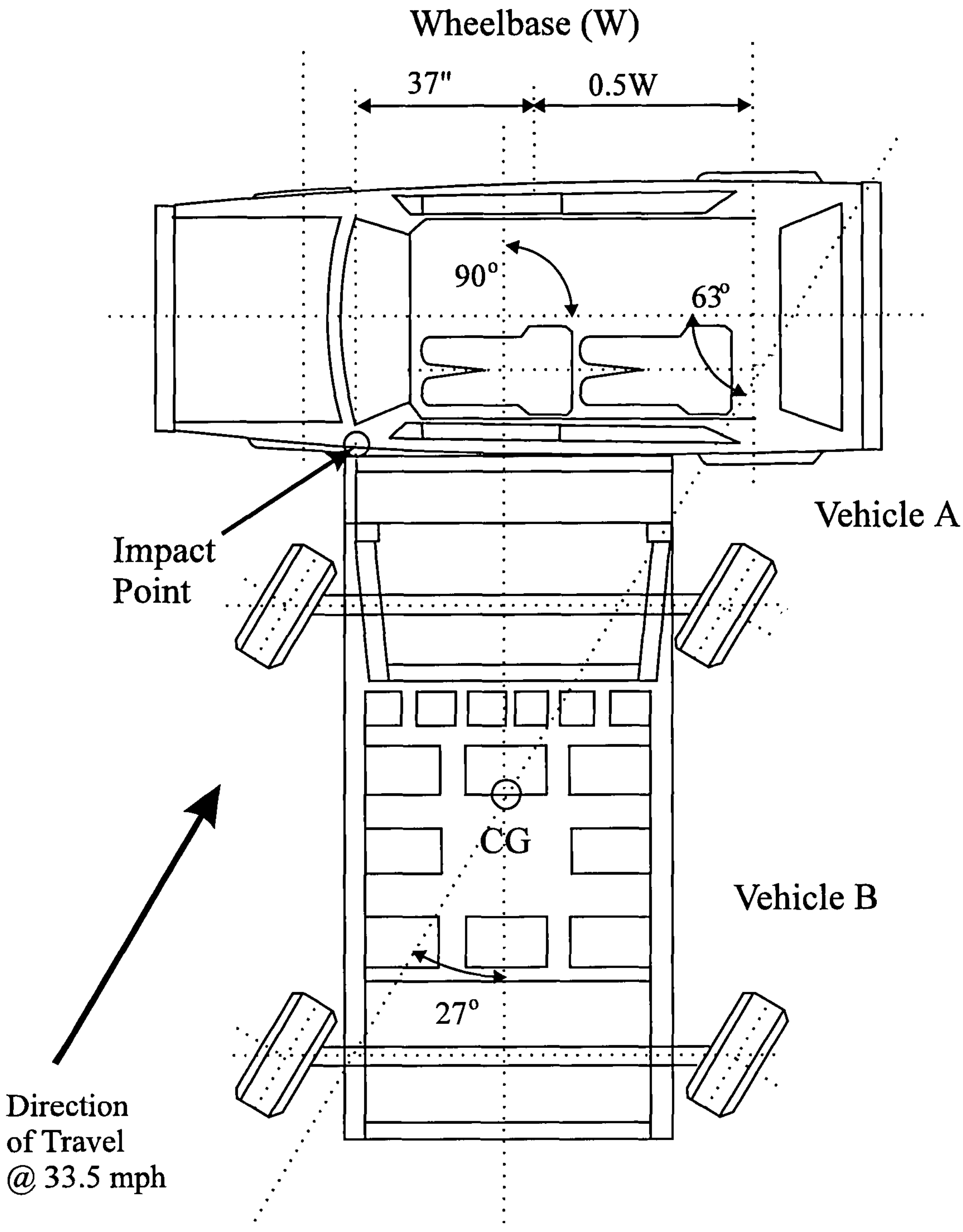

Figure 1 FMVSS 214 crabbed trolley test configuration (NHTSA 1990).

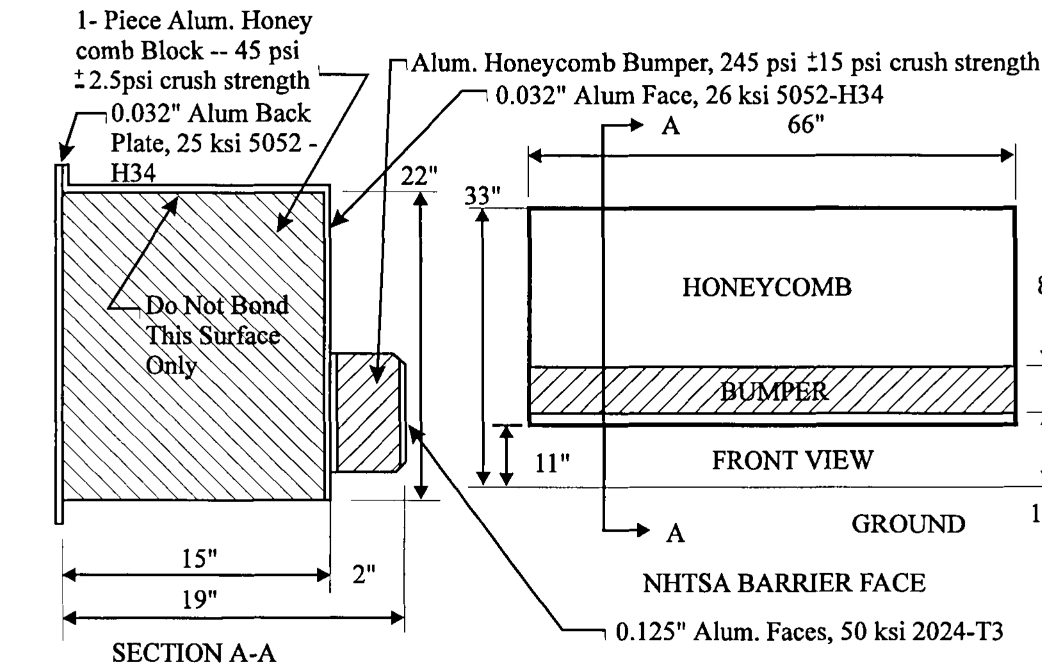

Figure 2 FMVSS 214 side impact deformable barrier face(NHTSA 1990).

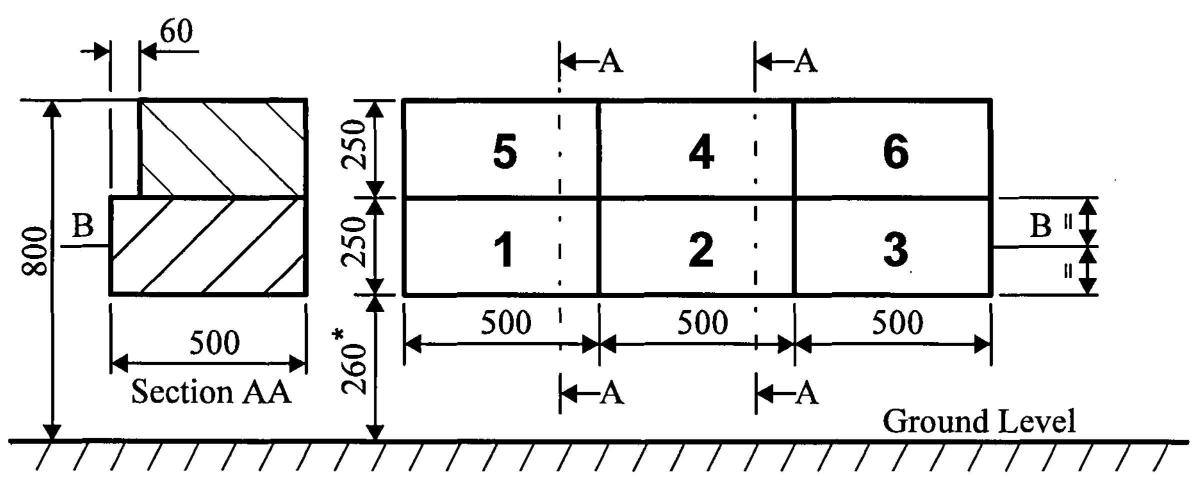

Figure 3 ECE Regulation 95 deformable barrier face dimensions

The aim of the project was to specify what the unit benefits would be if cars met these standards (ie; what the manufacturing costs would need to be to break-even). The Australian Federal Chamber of Automotive Industries provided more definitive cost information which enabled a benefit-cost analysis to be undertaken.

The study used the Harm reduction method to compute the potential benefits of the proposed dynamic side impact requirement. Harm refers to the cost of trauma and is the product of the frequency and cost of injury to the community. MUARC were the first agency in Australia to adopt this method developed by Malliaris in the US for use in this country in the previous FORS report CR 100[5] which described the potential benefits of frontal crash measures.

In its original form (Malliaris et al 1982; 1985; 1987) Harm was used to specify the total injury savings by the introduction of a particular safety measure. However, in conjunction with Kennerley Digges of University of Washington, MUARC subsequently expanded the method to permit a more detailed and systematic assessment of injury reduction by body region and seating position which could then be summed to total Harm reduction and unit Harm benefits.

This approach enabled test and crash data findings published in the road safety literature to be incorporated in the calculations, thereby reducing the amount of guess-work normally required in calculations such as these. Where no published figures were available, however, the study team were forced to use the consensus view of a panel of experts[6] in arriving at these body region and restraint condition savings. The amount of published data is normally a function of the attention a particular measure has received by the research community as well as its newness. There has not been much published data on side impact improvements using either standard and so heavy reliance was made on whatever test figures were made available and expert panel assessments for computing the likely benefits of a dynamic side impact standard.

From the Harm benefits per vehicle and the indicative costs provided by local manufacturers, the benefit cost ratios for the two alternative standards were calculated as:

- ECE Regulation 95 1.59
- FMVSS 214 (dynamic) 1.47

FUTURE WORK

It is proposed that MUARC use the same methodology and the data from the three crash tests (FMVSS 214, ECE R95 and the hybrid procedure) to conduct a benefits analysis for the hybrid test procedure. If this produces benefits equal or greater than those for the current US and European regulations, then this would support the hybrid test proposal being considered in international forums as an alternative to those standards.

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DEVELOPMENT OF MOVING DEFORMABLE BARRIERS IN JAPAN

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ABSTRACT

A Moving Deformable Barrier (MDB) used to simulate a colliding vehicle is an important element in full-scale side impact test procedures. There are several differences in weights, dimensions, force deflection characteristics, etc. between the US MDB and European MDB, by reflecting the difference in vehicle population in their markets. During the past studies being made for the introduction of side impact testing procedures into Japan, it is expected to develop a practical MDB corresponding to the situations of the Japanese automobile.

This paper reports the results of the studies of Japanese automobile characteristics, and also describes the design and performance of a MDB developed in Japan.

The dimensions of the vehicle front structure and vehicle weight were studied for Japanese passenger vehicles being registered in 1993. The force deflection characteristics of the vehicles were obtained from frontal impact tests against an instrumented fixed rigid barrier for the typical Japanese passenger vehicle. The values of these parameters were weighted by the number of vehicles sold during that year. The results of these studies were compared with the specifications of the US and European MDB.

Based on the investigation results of Japanese automobile, the specifications of the MDB to be used in Japanese side impact test procedures were determined. To meet these specifications, studies were made for materials and construction of the MDB, paying attention to form a practical test procedures. Frontal barrier impact tests and full-scale side impact tests were carried out to evaluate the characteristics of the MDB.

INTRODUCTION

The reduction of injuries of occupants during side collision is a major problem worldwide. Extensive research into this problem is now being carried out, mainly in the US, Europe, and Japan. As a result, vigorous controversy has come about during various international conferences promoting international harmonization, such as WP29/GRSP, ISO, ESV, and the like. However, international harmonization has not yet been attained, and in the US a new test method FMVSS214 in which an actual vehicle dynamic test method has been added to the conventional static side door strength test method came into effect in October 1990. On the other hand, in Europe, WP29 has included the ECE standard ECE/R.95 which incorporates an actual vehicle test method, and this is expected to be put into effect very soon. The EC commission is also studying an EEC directive with almost the same content as the ECE standard, aiming at its practical application from October 1998. These two methods differ with respect to the MDB, the dummy, and the injury criteria.

In Japan, research into the reduction of injuries of vehicle occupants during side collisions reported in a transportation technology inquiry commission report was adopted as an item for research in the intermediate term. In Japan it is expected that a proposal for a suitable test method will be made within a few years.

This report, based on such existing situations, summarizes the development of an MDB as one link in the basic research for studying effective test methods relating to Japanese situations, with the ultimate goal of reducing injuries of vehicle occupants during side collisions.

INVESTIGATION OF ITEMS FOR STUDY OF MDB

In order to study the items involved in developing an MDB, investigations were made into the curb weight, dimensions of frontal sections, and frontal stiffness of bonnet type automobiles (117 vehicle types, 4,370,908 vehicles) registered in Japan in 1993. Became of the actual number of vehicle types in existence, a large number of vehicle types had to be investigated, classified other than by vehicle model, according to body shape (two-door, four-door, etc.). The data on the 1993 model automobiles was used because there is no large difference between the ratio of the number of possessed vehicles currently used on the market and the number of different types of vehicles sold. The various types of data obtained in this investigation were weighted according to the number of vehicles sold, and accumulative fiftieth percentile values were obtained for curb weight, dimensions of frontal sections, and frontal stiffness.

Investigation of Curb Weights

The accumulate frequency of the curb weight of Japanese vehicles is shown in Figure 1. From this graph, the curb weight of Japanese vehicles is 1080 kg in the accumulative fiftieth percentile. The curb weight also differs according to the grade of identical vehicle types. In this investigation the average values were obtained for the all grades of identical vehicles.

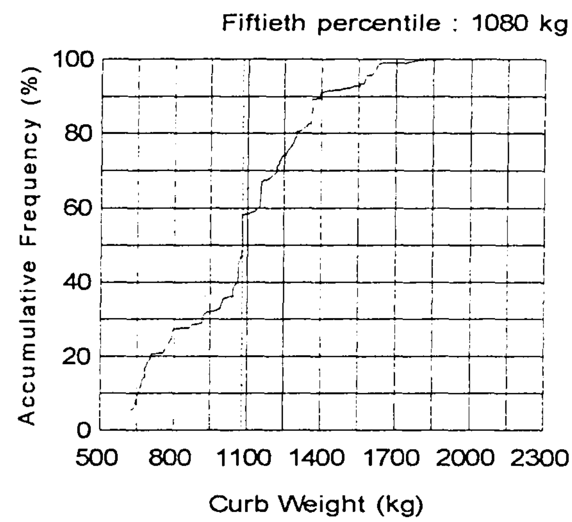


Figure 1. The Accumulative frequency of the curb weights of Japanese vehicles.

The results of this investigation show that the curb weights of Japanese vehicles are midway between the weights of the specified MDBs for the various European and American test methods. Because the time periods of the European and US investigations differ from the periods for the Japanese investigations, the results of the investigation based on the Japanese vehicles of the same period (1978) are given in Figure 2. From this graph, the curb weight is 865 kg in the accumulative fiftieth percentile, and the values for the investigation period corresponding to the periods for the European and US investigations are close to the weight of the MDB proposed by the Europeans.

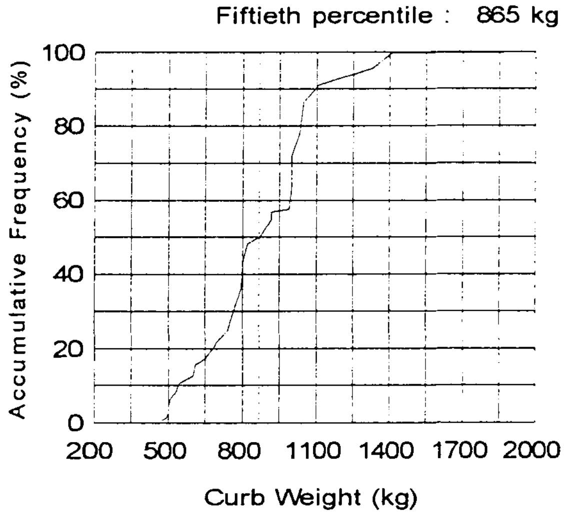


Figure 2 The accumulative frequency of curb weights of Japanese vehicles (1978 data).

The curb weights have had a tendency to increase because the vehicles have tended in recent years to become large and because of legislated vehicle construction changes as safety measures. The curb weights of the Japanese vehicles are clearly values close to the European investigative results if taken retroactively to the respective European and US investigation periods.

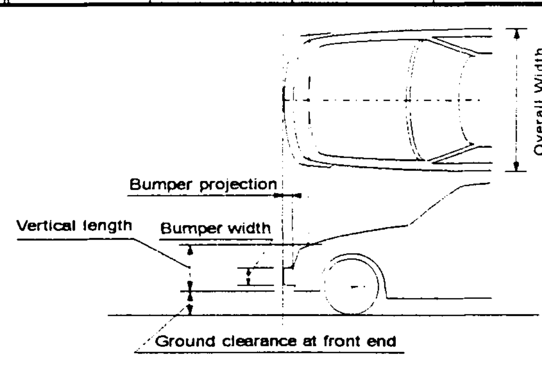
Investigation of Dimensions of Frontal Sections

The parts covered in the studies on the dimensions of the MDB and the dimensions of the accumulative fiftieth percentile obtained as a result are given in Table 1. This table shows that the dimensions of the parts of the Japanese vehicles are 1685 mm overall width, 426 mm vertical length (vehicle thickness) of the bonnet front end position, 61 mm bumper projection, 290 mm bumper width, and 240 mm ground clearance at the front end of the vehicle. This data is also weighted according to the number of vehicles sold, and then analyzed. The width of the vehicle is taken at the part of the frontal surface which does not contribute to the stiffness during a collision as a result of the body closure and the inner structure. So the width of vehicle is 1570 mm as shown in Figure 3 when the body width between the bumper corners. Also, the vertical length is 500 mm as shown in Figure 3 when at a position 300 mm from the front end, because the frontal shape changes along the vertical length after a collision.

Table 1.

Accumulative fiftieth percentile of vehicle front section dimensions

Part	Overall width	Vertical length	Bumper projection	Bumper width	Ground clearance at front end
Dimension (mm)	1685	426	61	290	240



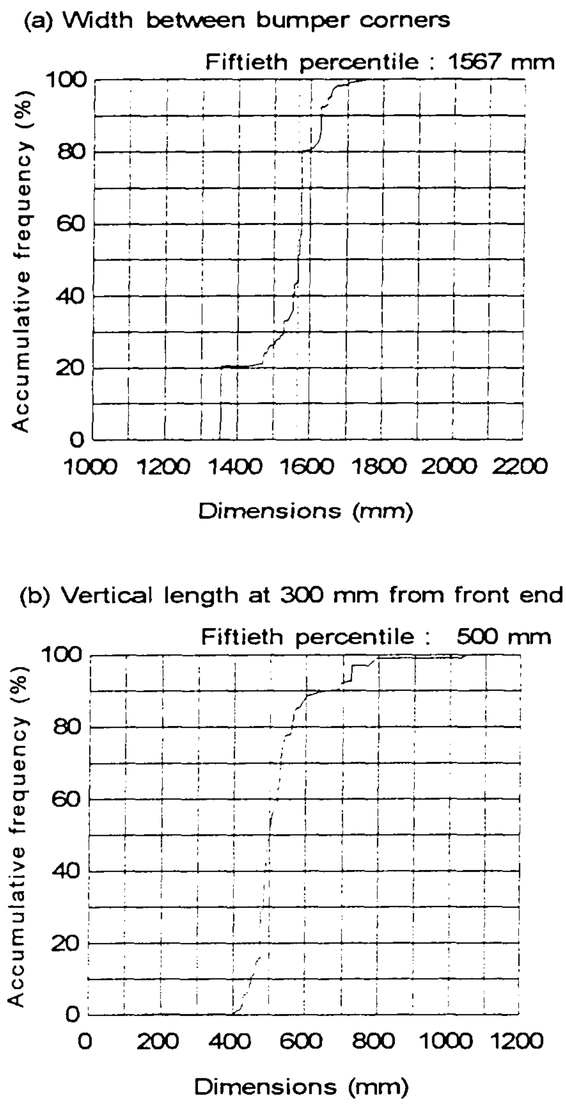


Figure 3. Frontal dimensions relative to strength of materials.

The results of the Japanese investigation are for a different period from those for investigations in Europe and the US, because in Japan there is no major change in dimensions of the vehicles from legislation, and, even for the same periods as the European and US investigations, it is presumed that there are no major differences from the results of the present investigation. In consideration of the foregoing, the various dimensions of the front parts of the Japanese vehicles are close to the proposed European barrier specifications.

Frontal Stiffness

The force occurring for every 50 mm of deflection was investigated, based on the barrier test results of 104 types of 1993 models (4,127,683 vehicles sold). In these results, the number of units sold was weighted, and Figure 4 shows the compiled force-deflection characteristics for the accumulative fiftieth percentile of the force value for the various amounts of deflection. This graph shows the correspondence between the required characteristic zone of the barrier in the European system and the dynamic characteristics of the barrier in the US system. The frontal stiffness of the Japanese vehicles is close to the required characteristics of the proposed barrier using the European system, and in particular the amount of deflection is in a range up to 300 mm which is within the required characteristic zone.

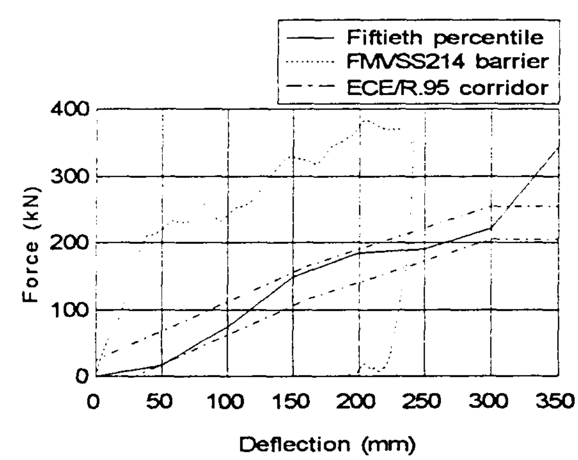


Figure 4. Frontal stiffness of Japanese vehicles.

Next, among the above-mentioned vehicle types, the force divided to six segments for the most common 23 types of vehicles sold (2,764,891 units) are shown in Figure 5. In these graphs, the number of units sold was weighted, and the compiled force-deflection characteristics are illustrated for the cumulative fiftieth percentile of the force values for the various amounts of

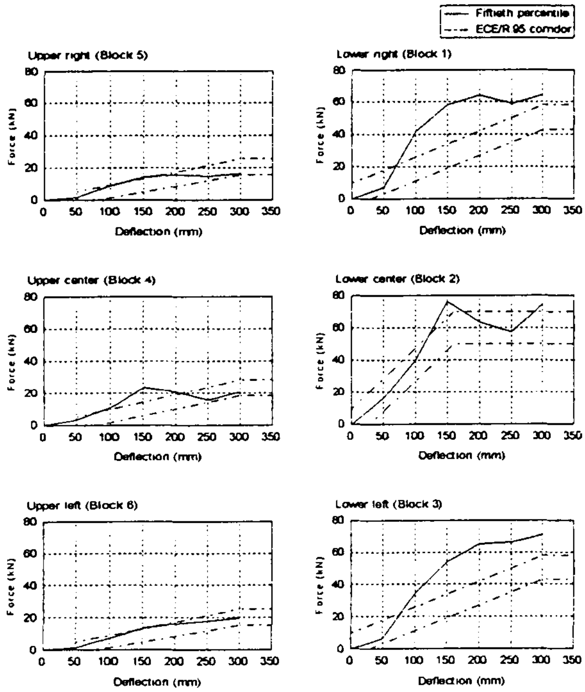


Figure 5. Frontal stiffness of six segments of Japanese vehicles.

deflection. In addition, in these graphs, the correspondence of the required characteristics zone is shown for the proposed barrier using the European system. The three upper segments and the lower central segments illustrate characteristics which are close to the required range in the European proposal. On the other hand, in the lower section the two left and right segments show higher characteristics than the European proposal. All the segments which satisfy the range for the European proposal are in a range up to a deflection of about 75 mm. Where the lower characteristics of the Japanese vehicles differ from the European-proposed barrier, the effect is expected to be greater from the difference in the timing of the investigations. The engine arrangement is mainly longitudinal for the vehicles in the period covered by the European investigation. On the other hand, a total of 80% of the vehicles in the period of the Japanese investigation have a transverse arrangement. As a result of the difference in the engine mounting method, the stiffness of the central part is high in the longitudinal arrangement, and, because there is almost uniform stiffness with the transverse arrangement, it is presumed that a large difference is created by the divided characteristics. There is no data for Japanese cars in 1980 which is the period of the European investigation, but characteristics for the Japanese cars investigated using a representative 14 types of vehicles which were the main passenger vehicles with longitudinal engines in 1985 are given in Figure 6. As shown in the figure, the frontal stiffness of the Japanese vehicles at that time in 1985 mostly satisfies the required performance range for the European barrier.

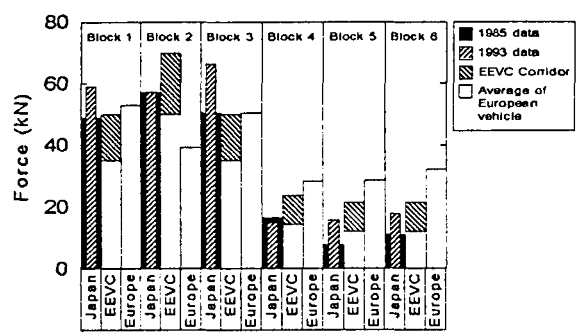


Figure 6. Comparison of Japanese and European frontal stiffness of vehicles.

Summary of Investigative Results

The results of the investigation of various types of Japanese vehicles using 1993 models compared with the item values for the MDB required by the regulations of Europe and the US are given in Table 2. The results from the US MDB differ from the European and Japanese investigation results because results from small trucks are contained in the US results. This table shows that the various characteristics of the MDB compiled from average values of the Japanese vehicles are close to the required items for the MDB proposed in the European system. However, a difference is observed in the curb weight and the frontal stiffness of the vehicles because of the difference in the investigation periods. We believe that, this difference will cause a problem inasmuch as, in the future when the MDB will be restudied to harmonize internationally.

Table 2.
Comparison of items relating to European and the US MDBs and Japanese investigative results

		ECE/R. 95	FMVSS214	Japan
Weight (kg)		950	1369	1080 (1993 data) 865 (1978 data)
Width (mm)		1500	1676	1567
Length (mm)		500	559	500
Depth (mm)		500	482	–
Ground clearance (mm)		260	280	240
Stiffness (kN)	Dfl. 100mm	60~110	245	75
	Dfl. 200mm	140~190	380	175
	Dfl. 300mm	210~260	–	210

DEVELOPMENT OF MDB FOR EUROPEAN SYSTEM

When the previously mentioned results of the investigation of Japanese vehicles were taken into consideration. the European-proposed MDB was judged to be suitable for the Japanese market. Accordingly, when legislation is enacted for a side impact test method in Japan, the development of an MDB satisfying the European-proposed barrier characteristics was commenced with the objective of utilizing a approval test and a test for vehicle development. Various MDBs compatible with the ECE/R.95 standard were developed, using the European side impact test system. These include a hard foamed urethane unit manufactured by the Fritzmeier Co. of Germany, a triangular pyramid-shaped aluminum honeycomb unit manufactured by UTAC of France, triangular pyramid-shaped and multi-layered aluminum honeycomb units manufactured by the Cellbond Co. of the UK, a multi-layered aluminum honeycomb unit manufactured by the Plascore Co. of the US, and the like.

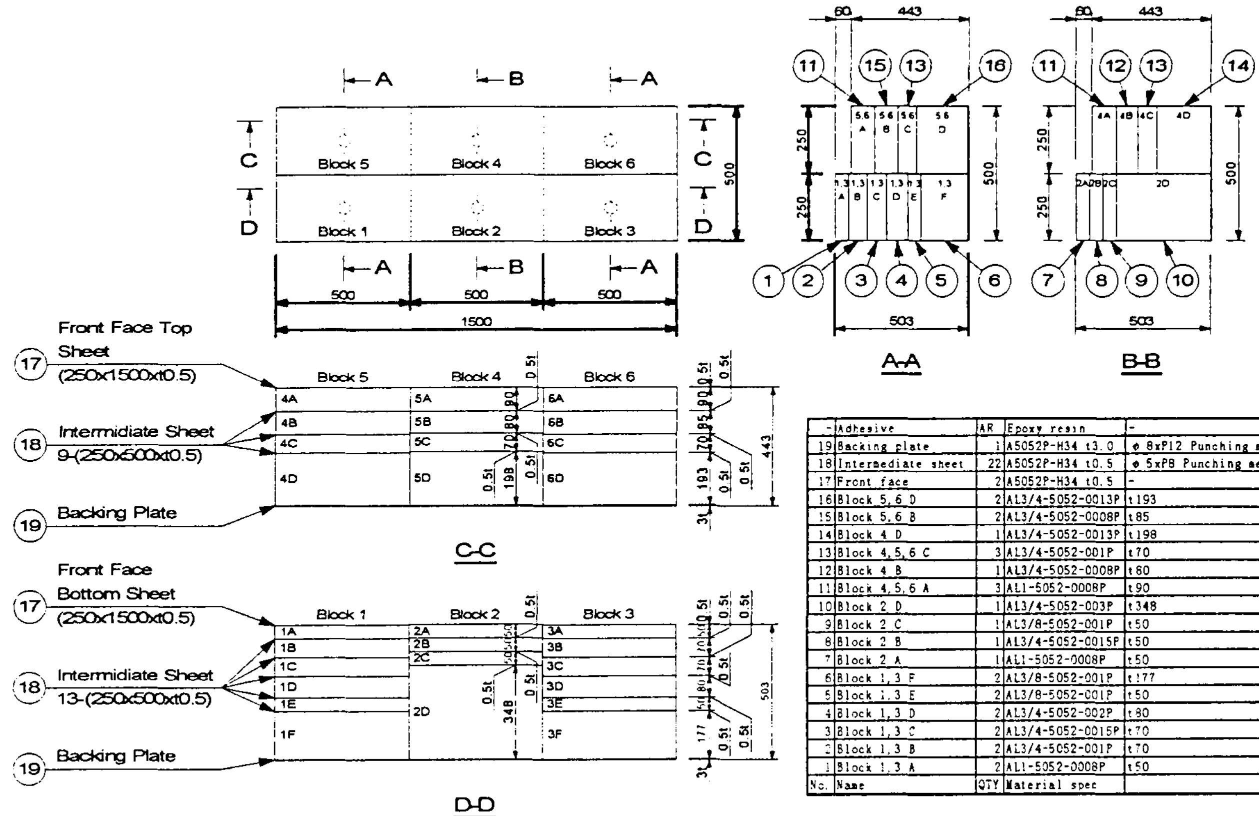
These MDBs were also initially imported into Japan to be used for basic research, but up to the present time, after deliveries, cost, and performance were taken into consideration, the research and development was carried out in parallel with the development of Japanese MDBs. In Japan, hard foamed ethylene units and triangular pyramid shaped aluminum honeycomb units were manufactured in the past for use in various tests with actual vehicles. However, the performance of these units did not completely satisfy ECE/R.95. On the basis of the results of this past research a new MDB was developed to primarily include the following characteristics.

(1) Superior in reproducibility.
(2) No change in performance in environmental conditions of temperature, humidity, and the like.
(3) Complete understanding of deflection conditions after collision.
(4) Low cost, stable supply.
(5) Light weight, easily handled.

To provide a unit to satisfy these conditions, aluminum honeycomb was selected as the material and the multi-layered structure was designed because of the difference in impact point of the MDB during a collision with an actual vehicle.

MDB Structure

Figure 7 shows the structure and configuration of the MDB developed in Japan. This figure also provides the specifications of the aluminum honeycomb used and the specifications of the aluminum plate. In addition, Table 3 gives the crush strength of the honeycomb blocks used in the MDB. Figure 8 shows the significance of the terms used in these figures and tables for the honeycomb designation.



No.	Name	QTY	Material spec	
-	Adhesive	AR	Epoxy resin	-
19	Backing plate	1	A5052P-H34 t3.0	ϕ 8xP12 Punching metal
18	Intermediate sheet	22	A5052P-H34 t0.5	ϕ 5xP8 Punching metal
17	Front face	2	A5052P-H34 t0.5	-
16	Block 5, 6 D	2	AL3/4-5052-0013P	t193
15	Block 5, 6 B	2	AL3/4-5052-0008P	t85
14	Block 4 D	1	AL3/4-5052-0013P	t198
13	Block 4, 5, 6 C	3	AL3/4-5052-001P	t70
12	Block 4 B	1	AL3/4-5052-0008P	t80
11	Block 4, 5, 6 A	3	AL1-5052-0008P	t90
10	Block 2 D	1	AL3/4-5052-003P	t348
9	Block 2 C	1	AL3/8-5052-001P	t50
8	Block 2 B	1	AL3/4-5052-0015P	t50
7	Block 2 A	1	AL1-5052-0008P	t50
6	Block 1, 3 F	2	AL3/8-5052-001P	t177
5	Block 1, 3 E	2	AL3/8-5052-001P	t50
4	Block 1, 3 D	2	AL3/4-5052-002P	t80
3	Block 1, 3 C	2	AL3/4-5052-0015P	t70
2	Block 1, 3 B	2	AL3/4-5052-001P	t70
1	Block 1, 3 A	2	AL1-5052-0008P	t50

Figure 7. The structure and the configuration of the MDB developed in Japan.

Table 3.
The crash strength of the honeycomb blocks used in the MDB

Block No.	Designation	Crash Strength	
		MPa	PSI
1A , 3A	AL1-5052-0008P	0.032	4.7
1B , 3B	AL3/4-5052-001P	0.086	12.5
1C , 3C	AL3/4-5052-0015P	0.148	21.5
1D , 3D	AL3/4-5052-002P	0.231	33.5
1E , 3E	AL3/8-5052-001P	0.310	45.0
1F , 3F	AL3/8-5052-001P	0.345	50.0
2A	AL1-5052-0008P	0.032	4.7
2B	AL3/4-5052-0015P	0.148	21.5
2C	AL3/8-5052-001P	0.310	45.0
2D	AL3/4-5052-003P	0.372	54.0
4A , 5A , 6A	AL1-5052-0008P	0.032	4.7
4B , 5B , 6B	AL3/4-5052-0008P	0.061	8.8
4C , 5C , 6C	AL3/4-5052-001P	0.086	12.5
4D , 5D , 6D	AL3/4-5052-0013P	0.114	16.5

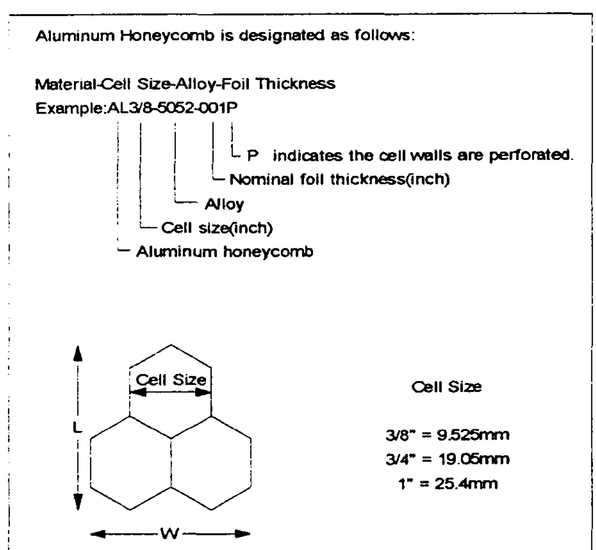


Figure 8. The significance of the honeycomb designation.

The dimensions of the barrier satisfy the standards stipulated in ECE/R.95. This barrier is fabricated as six segments. The aluminum honeycomb material used in these segments is AL-5052, with a segment length of 250 mm and width of 500 mm, which is commonly used in Japan. In order to satisfy the performance requirements of ECE/R.95, the configuration incorporates 4 to 6 layers of honeycomb blocks of different core sizes and foil thicknesses. Each block layer is pre-crashed by 5 mm prior to bonding so that the characteristic initial peak of the aluminum honeycomb is eliminated. In addition, the honeycomb used in this barrier is provided with a perforated core to remove all air. The pre-crashed surface of the honeycomb blocks is the side which receives the impact, and these blocks are joined together through an aluminum plate 250 mm long, 500 mm wide and 0.5 mm thick in which 5 mm diameter holes have been drilled, using an epoxy resin adhesive. The back plate is an aluminum plate 500 mm long, 1500 mm wide and 3 mm thick in which 8 mm diameter holes have been drilled. The six honeycomb segments are attached to this back plate using an epoxy resin adhesive. The perforated plate is used to eliminate performance differences caused by air compression during the collision. Figure 9 shows the structure of the assembled MDB developed in Japan.

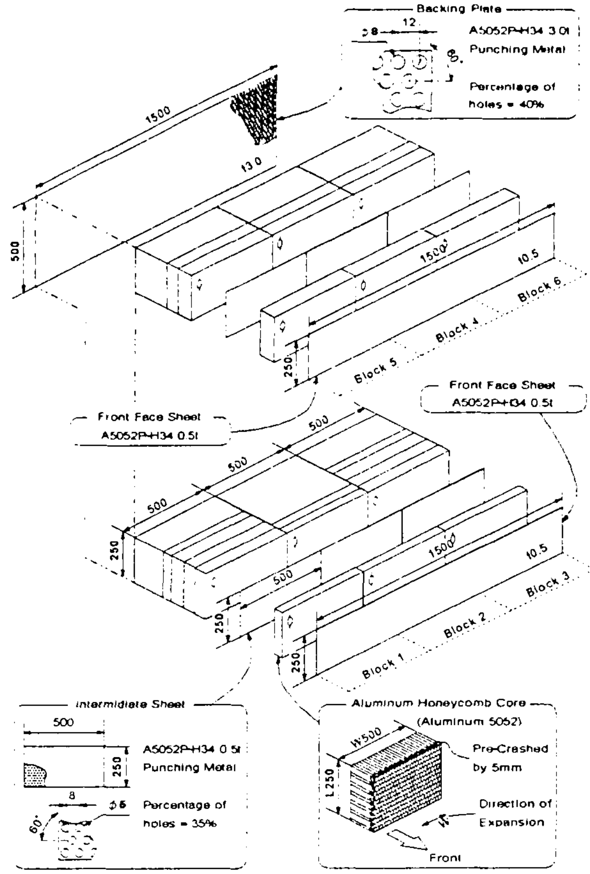


Figure 9. The structure of assembled MDB developed in Japan.

Calibration Test

Test Method - A barrier collision test from ECE/R.95 as shown in Figure 10 has been proposed as a MDB calibration test.

In this test method, an MDB weighing 950 kg is colliding at an impact velocity of 35 km/h to a rigid barrier installed with a load cell divided into six segments. A method of evaluating the calibration results is stipulated by the force-deflection characteristics of the total barrier and the each segments. The amount of the dissipated energy and the amount of permanent deflection of the middle intersecting surface of the lower segments (Level B) is shown in Table 4.

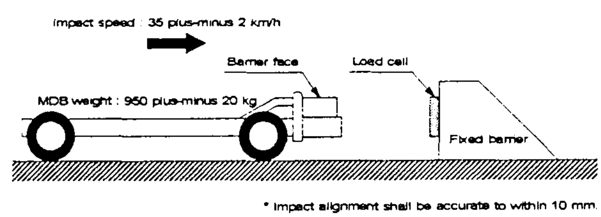


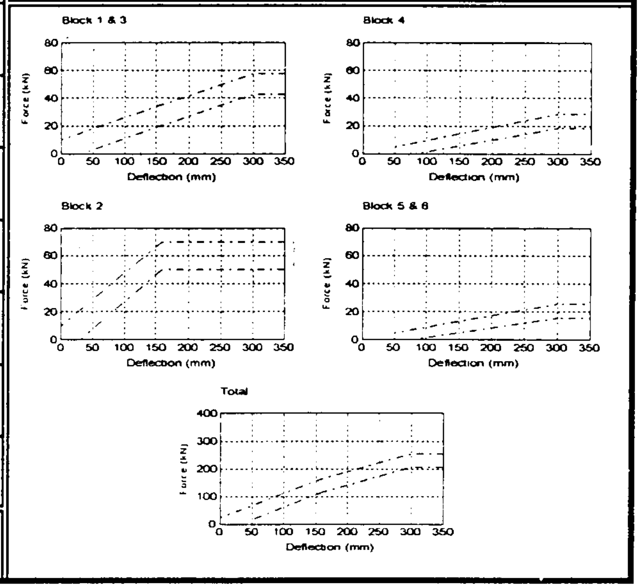
Figure 10. Calibration test method of European proposed MDB.

In this test method, the amount of offset between each segment and the impact position of the corresponding force measurement surface is stipulated to be within plus-minus 10 mm in order to obtain the correct characteristics of the segments.

Table 4.

The requirement characteristics of European proposed MDB

Dissipated energy		Force-deflection curves
Block 1 (kJ)	10±2	
Block 2 (kJ)	14±2	
Block 3 (kJ)	10±2	
Block 4 (kJ)	4±1	
Block 5 (kJ)	3.5±1	
Block 6 (kJ)	3.5±1	
Total (kJ)	45±5	
Deflection of Level B (mm)	330±20	



A performance test has been implemented in Japan based upon the test method specified by the R.95, using the test device shown in Figure 11. In the test, the acceleration at the center of gravity of the moving barrier and the loads on the six segments on the fixed barrier are measured, and load values are obtained by processing the data through a filter conforming to SAE J211 CFC180. The deflection, energy, etc. are obtained by an integral calculus calculation. From these results, the force-deflection and energy-deflection characteristics and the like are obtained.

The barrier mounted to moving barrier with grilled type back plate to remove air during the collision.

<u>Test Results (Performance Test)</u> - The MDB conditions before and after the test are shown in Figure 12. In this test the collision velocity is 35.0 km/h, while the offset between the MDB and the load measurement surface is 5 mm laterally and 5 mm vertically.

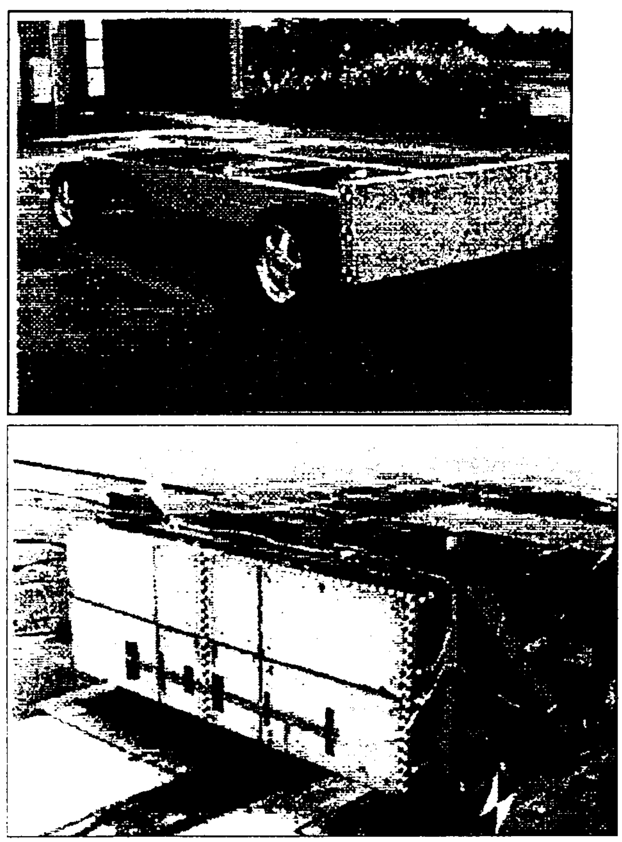


Figure 11. The calibration test device of JARI.

Before the test

After the test

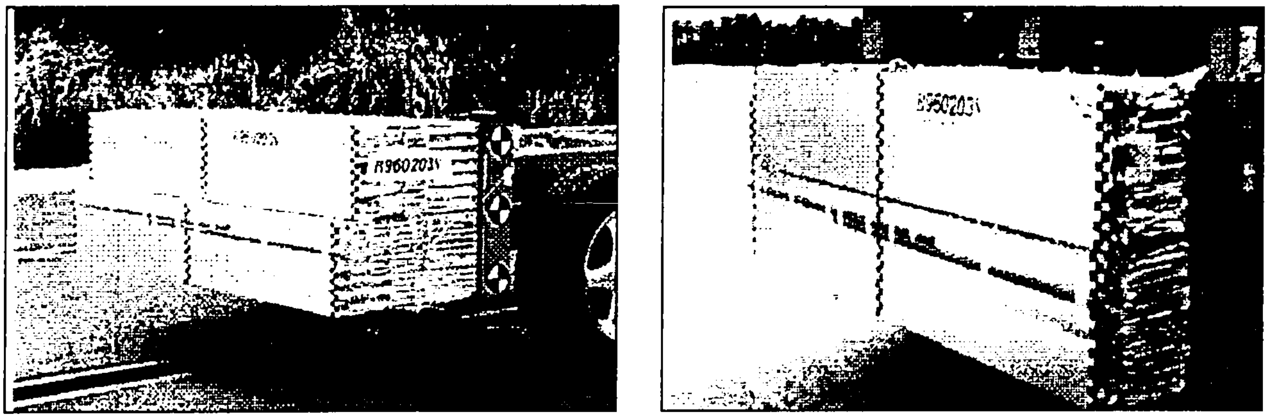


Figure 12. The MDB conditions before and after the calibration test.

Table 5.

The dissipated energy and deflection of Level B after the calibration test of the MDB developed in Japan

	Test 1
Block 1 (kJ)	10. 3
Block 2 (kJ)	15. 0
Block 3 (kJ)	10. 2
Block 4 (kJ)	4. 1
Block 5 (kJ)	3. 7
Block 6 (kJ)	3. 7
Total (kJ)	47. 0
Deflection of Lebel B (mm)	335

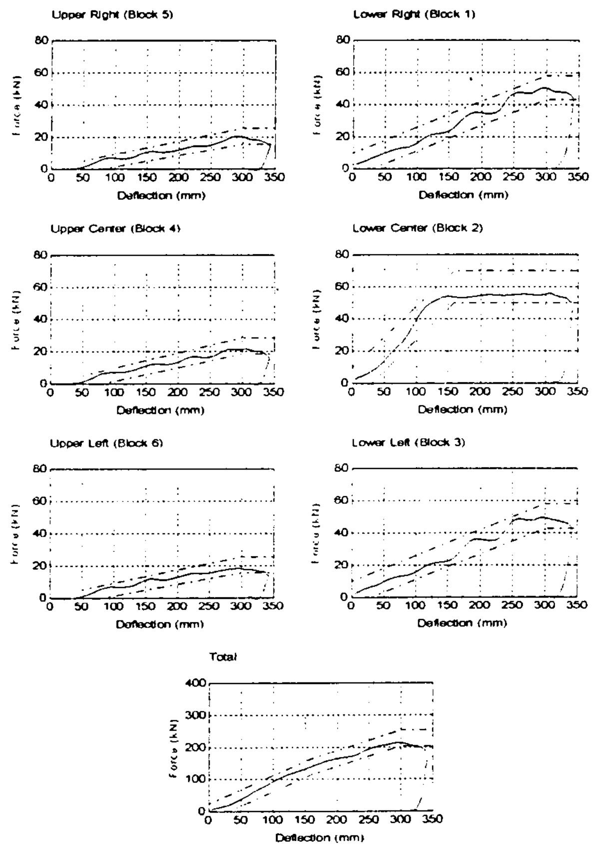


Figure 13. The force-deflection characteristics of the MDB developed in Japan.

The permanent deflection at Level B after the test and the amount of dissipated energy at the total barrier and at each segments are given in Table 5. From this table it is seen that the basic conditions of 330 plus-minus 20 mm are satisfied at a permanent deflection of 335 mm. In addition, both the amount of dissipated energy is satisfied the standard range for each of the six segments and the total barrier.

The force-deflection characteristics for all six segments are shown in Figure 13. The force values for the deflection at the instant of the collision fall in the standard range. However, for the critical deflections, where the forces go out from the standard range, for the total and for segment 4, 5, and 6 are each less than 320 mm. However, because the permanent deflection in the standard is 330 plus-minus 20 mm and there is almost no deflection above 320 mm when the actual vehicle test is used, this MDB can be evaluated as satisfying the standard for ECE/R.95.

Test Results (Reproducibility Test) - A test was implemented for investigating reproducibility under these conditions. Figure 14 shows force-deflection characteristics for two repeated tests, while Table 6 gives the permanent deflection and the dissipated energy. From these results it is seen that the MDB developed in Japan is greatly superior in reproducibility.

Table 6.
The reproducibility of the dissipated energy and deflection of Level B of the MDB developed in Japan

	Test 1	Test 2
Block 1 (kJ)	10.3	10.5
Block 2 (kJ)	15.0	15.3
Block 3 (kJ)	10.2	10.4
Block 4 (kJ)	4.1	4.1
Block 5 (kJ)	3.7	3.7
Block 6 (kJ)	3.7	3.8
Total (kJ)	47.0	47.7
Deflection of Level B (mm)	335	335

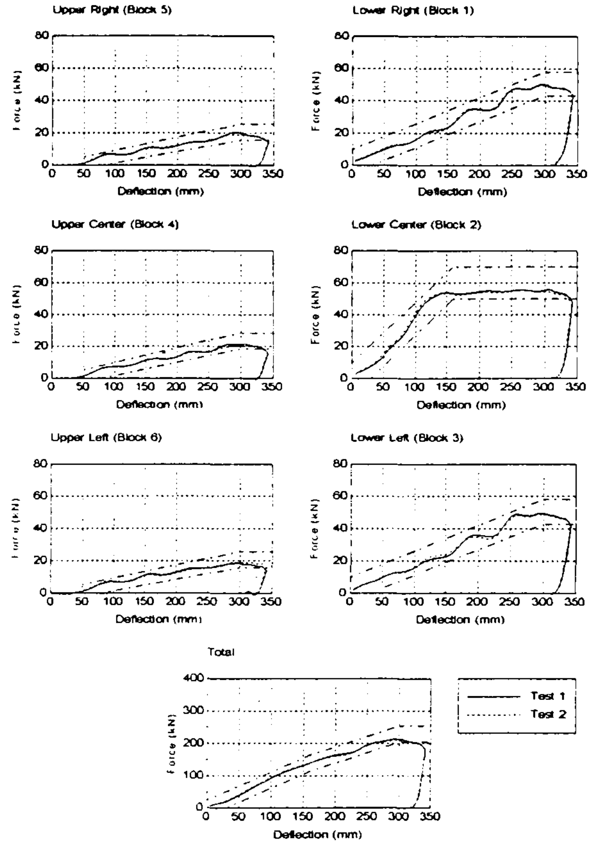


Figure 14. The reproducibility of the force-deflection characteristics of the MDB developed in Japan.

Summary of MDB Development

An MDB was developed which satisfied the conditions stated under ECE/R.95. This MDB is light and provides superior handling and performance with excellent reproducibility characteristics. In the future, it expected to use in Japan for type approval test for regulation of side impact test and for developing vehicles.

CONCLUSION

A MDB that best fits to the market situations in Japan and satisfies the performance criteria proposed by the Europeans has been developed with the result of investigations of the dimensions and characteristics of Japanese vehicles. As a result, it became possible to construct a MDB by using aluminum honeycomb, that have light weight, superior handling, excellent reproducibility, and stable performance against temperature, humidity, etc.

In the future, it will be expected to use in Japan for type approval testing for side-impact safety and testing for developing vehicles.

Finally, we would like to thank all the members of the Showa Aircraft Industry Co., Ltd. who cooperated in the development of the MDB of this research.

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AN AUSTRALIAN PERSPECTIVE ON SIDE IMPACT PROTECTION

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ABSTRACT

This paper discusses the side impact crashes that occur in Australia, and the reasons for the differences between crashes in Australia, USA and Europe. It provides some information about the Australian road system and the Australian vehicle fleet which lead to these differences. It proposes an approach to the development of occupant protection in side impact crashes to achieve the maximum benefit for the Australian community.

INTRODUCTION

Effective safety development requires a quantitative understanding of the frequency and severity of injury resulting from road crashes in Australia. It is necessary to know the type, frequency and severity of the crashes as well as the severity of the resulting injuries. This knowledge then focuses the research and development, ensuring that priority is given to achieving the maximum injury risk reduction.

Motor vehicle frontal crash conditions are sufficiently different in Australia to Europe and the USA to warrant a different approach to restraint system development. The high rate of seat belt use, the high frequency of frontal collisions due to the large proportion of undivided highway, and the strong bias of crash frequency to lower speeds, encouraged the development of a restraint system which didn't simply meet government regulation, or provided good numbers in NCAP tests, but one which was optimised to achieve minimal societal harm (1).

There are a number of characteristics of side impact crashes in Australia that similarly warrant a specific approach to side impact protection.

The Australian Passenger Car Fleet

The Australian fleet differs from passenger car fleets in Europe and the USA in both size and age. The typical Australian family car is approximately 5m long, weighs 1500 kg and is rear wheel drive. The fleet average age is over 12 years, and although there is a growing popularity of small, front wheel drive cars, the older fleet is predominantly full size vehicles. One segment of the market that has expanded rapidly in recent years is 4 wheel drive vehicles, which are used predominantly in suburban areas. These vehicles have a large size and mass, and of significance in side impact crashes, have a stiff front structure. This characteristic is exacerbated when vehicle are fitted with aggressive bull bars. This accessory was initially designed to provide vehicles with protection against animal collisions in outback areas. It is now becoming a fashionable accessory on vehicles which are used predominantly in suburban areas. As well as representing an extreme risk to pedestrians, they also make the vehicle more aggressive in a side impact collision.

The typical vehicle in Europe is almost a metre shorter, is front wheel drive and has a mass of 1000 kg. The average vehicle size in the USA is also reducing, and as the fleet average age is less than in Australia, the "bullet" car is more likely to be of a similar size and mass.

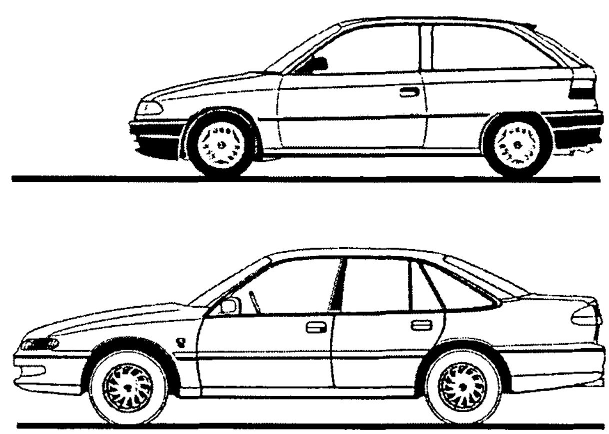

Figure 1: Comparison of Australian & European Vehicle Length

One further difference between vehicles is the proportion of 2 door sedans in Europe. These vehicles have a relatively long front door, but potentially increased structure in the rear of the vehicle. The front door of the Australian family car is also long, typically 100 mm longer than the US design, with the B-pillar located further rearwards from the front passenger. These differences may influence structural support and hence velocity of occupant impact, and may influence frequency of head impact with centre pillar. A comparison of the characteristics of these vehicles is given in Table 1.

Table 1
Comparison of Vehicle Size, Australia, Europe and USA

Vehicle Model	Length	Width	Wheelbase	Mass
Commodore	4850	1800	2750	1420
Kingswood (1975)	4850	1900	2800	1370
Toyota Landcruiser	4800	1950	2850	2100
Europe (Opel Astra)	4050	1650	2500	1000
USA (Buick Regal)	4900	1850	2750	1500

The majority of the 8 million passenger vehicles in the Australia fleet are medium to large car size, with approximately 70% having a kerb mass greater than 1000 kg. Mass distribution of the fleet is given in Table 2 (2).

Table 2
Mass of the Australian Fleet

Mass	Proportion
< 900 kg	15%
900 - 1100 kg	27%
1100 - 1300 kg	30%
1300 - 1500 kg	20%
> 1500 kg	8%

The Australian Road System

The road system in Australia has some unique characteristics. Australia is a large country, with an area greater than that of the USA, but with a population of 19 million people. This population is concentrated in the main cities - Melbourne and Sydney have populations greater than 3 million each, and Adelaide, Brisbane, Perth, Newcastle and Wollongong have populations of approximately 1 million each. Although the road systems in these cities are generally well developed, roads outside these urban areas are less developed than equivalent roads in Europe and the USA. The majority of main country roads are undivided, and have many uncontrolled intersections. There is thus the risk of high speed side impact crashes.

Side Impact Crash Frequency

There is a significant difference in the frequency of side impact crashes in Australia compared with the USA, as shown in Figure 2 (3,4,).

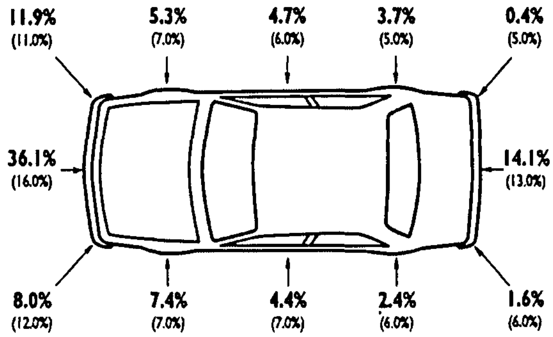

Figure 2: Australian & American Impact Direction

Side Impact Crash Severity

Although the proportion of side impact crashes is less, they tend to be higher speed crashes (5,9) and the injury outcomes are more serious.

Table 2
Injury Type and Severity

	FRONT	SIDE
ALL INJURY TYPES: Victoria	56%	28%
USA	39%	38%
SERIOUS & FATAL: Victoria	55%	32%
FATAL: Victoria	51%	46%

The proportion of serious head and neck injuries resulting from side impact crashes in both Australia and USA is shown in Figure 3 (6).

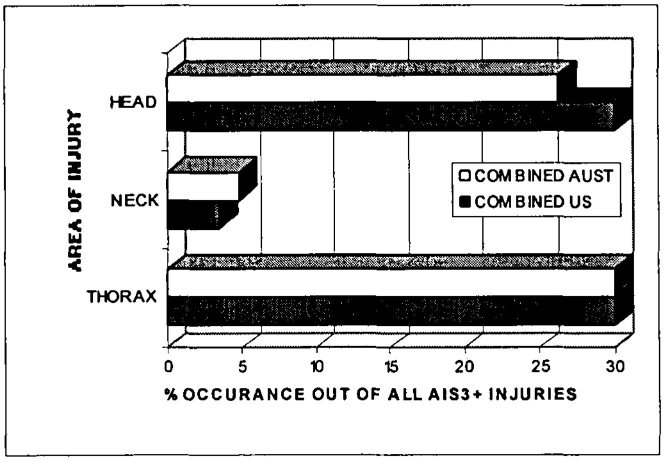

Figure 3. Injuries From Side Impact Crashes

In Australia, as in USA, older drivers and young drivers are over-involved in fatal crashes per distance travelled, as shown in Figure 4 (7). While this indicates inexperience or risk taking amongst young drivers, the increased risk to older drivers is in part a result of the increasing fragility with age. It is important that side impact protection strategies account for the varying size and fragility of car occupants.

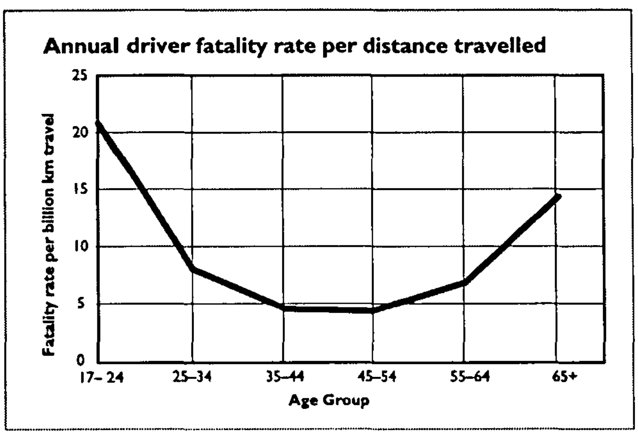

Figure 4: Driver Age in Fatal Crashes

Side Impact Injury Risk Assessment

Government legislation planned for Australia will provide for the use of either ECE or FMVSS 214 test procedures to demonstrate compliance. Given the Australian fleet size, use of the ECE test barrier of 950 kg mass would be inappropriate. The FMVSS test barrier mass of 1370 kg (3000 lb) is more representative of the Australian vehicle fleet.

Use of the SID dummy as specified in FMVSS 214 is seen as unsuitable in light of the need to ensure the development of maximum protection against head and neck injury. Side impact crash protection development is done by Holden using BioSID dummies and a range of car to car and FMVSS moving deformable barrier tests. Side impact sled testing requires the development of new techniques to duplicate the side impact crash pulse, which is quite unlike that of a frontal crash. New computer modelling techniques were also required to simulate the BioSID dummy, as these were not available. Holden has developed lumped mass and finite element models of the BioSID to allow development of side impact protection. A copy of the lumped mass model has been passed to TNO to be made available for public use.

The Australian Challenge

The challenge for Australia is to develop side impact crash protection that utilises the best of European and American safety technology, but at the same time recognises the differences in the injury risk that exist in Australia compared with Europe and the USA. Use of a technique for optimisation for minimum community cost will result in the maximum benefit for the Australian community.

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COMPUTER ANALYSIS FOR SIDE IMPACT OCCUPANT PROTECTION

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Paper Number 96-S6-O-08

ABSTRACT

In order to assist in providing effective protection for the occupants in a side impact collision, we must understand the crash phenomena. This understanding assist the effort to control the deformation mode, intrusion velocity, and the like of each section of the vehicle to the extent possible. For that purpose, computer analysis is very effective in that it allows investigation of data which is difficult to segregate by crash testing.

Among side impact test types, this paper deals with the European regulation ECE95. For simulation we used PAM-CRASH finite element code. First, in case of analyzing the side impact with computer, we explain that it is important to install finite element (FE) dummy model by comparing three simulation results, without dummy model, with rigid body dummy model and with FE dummy model. Next, we carried out two cases of simulations that had differences in side structure strengths, with FE dummy model. We compared the crash tests and simulations of the two cases. In these results, we found that the simulation results were reasonably well correlated to the vehicle crash test results within the parameters which we studied.

INTRODUCTION

Safety in collision is attracting a great deal of media attention. Various types of crash tests have been conducted aiming at automotive vehicle improvements. Along with this trend, the number of vehicles used for crash testing is constantly increasing, and manufacturers seek more efficient methods of vehicle design. We believe that it is one effective way to utilize Computer Aided Engineering (CAE) in the type of analysis described in this presentation. CAE is currently utilized to clarify crash phenomena that are not so easily revealed by crash tests and also to optimize some details of body structure. And, CAE is useful to shorten time required and increase precision of design improvements.

It is generally difficult to make absolute evaluation of crash characteristics by simulation. However, certain effects of body design refinement may well be predicted by following processes. First we conduct a test and a simulation using the same specifications, and compare these results. Next we perform another simulation, and comprehend the relationship of the two simulation results by comparing these.

This paper deals with side impact type of ECE95, a type used in crash testing. Subjected to a side impact, vehicle side structure will deform resulting sometimes the interior trim intruding into the inside of the vehicle and may contacting the occupants with potential injury. Some direct occupant injury factors depend on the door trim profile, the intrusion velocity, deformation properties of the door (center pillar), the occupant's body structure and deformation properties. In other words we need to understand very complex interaction of the above parameters attempting to predict occupant injury. For these reasons, installation of an FE dummy model in the simulation model is a more effective in simulation for the prediction of damage potential and dummy injury level. However, it takes a large amount of computer (CPU) time to perform an analysis with the FE dummy model. In this paper we try to investigate the effectiveness of the simulation without the dummy model and with a rigid dummy model (which save CPU time and are completely stable). Next we carry out two cases (with different side structure strength) of crash tests and simulations with FE dummy.

Model Outline

In this study we prepared a model of a prototype body. The model used for simulation consisted of the body model, moving deformable barrier (MDB) model and dummy model. As a dummy model, we used two kinds of dummy models - rigid body dummy model and FE dummy model. Here, rigid body dummy was the model which every part of the dummy was changed to rigid, and arms were omitted for simplifying dummy model. The whole simulation model was constructed with over 50,000 nodes (Figure 1.). The body models required for representing a crash phenomenon consisted of a body-in-white model, seat models and interior trim models. The MDB was modeled using solid elements which conforms to the force - deflection curve by European regulation ECE95. Figure 2 shows the force - deflection curve of this MDB model.

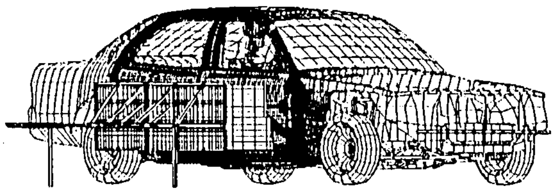

Figure 1. Outward of the Analysis Model.

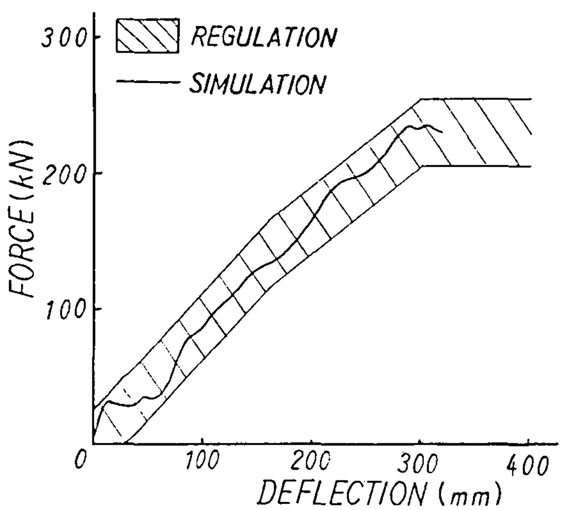

Figure 2. Force-Deflection Curve of ECE95 MDB Model.

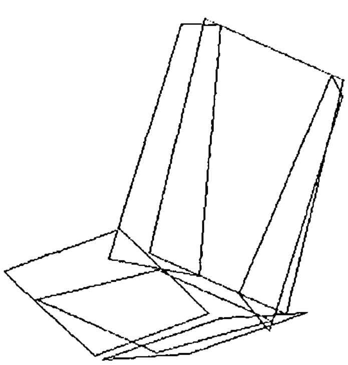

Figure 3. Seat Surface Model.

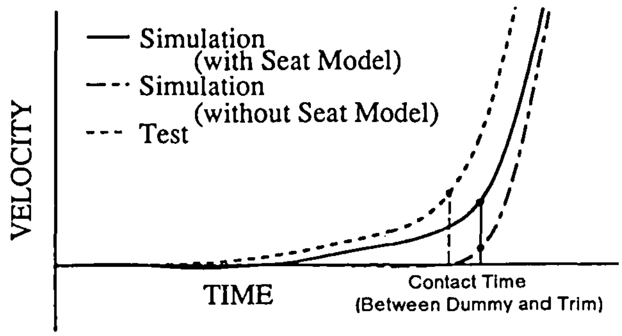

Figure 4. Pelvic Velocity-Time Curve.

Installation of a Finite Element Dummy Model into a Vehicle Model

In terms of simulation, when a part of the analyzed object contacts with another part, we must define contact conditions between them. Especially, the problem of contact between the materials of low Young's modulus and high Young's modulus - between the FE dummy and interior trim in this case - is very difficult. In preparation for the simulation of an overall model, we discussed enough the contact conditions between unassembled units of each type of interior material model and each part of the FE dummy model. By repeating this we obtained proper contact conditions between each part of the dummy and each interior component. Using penalty method in contact we referred to equation 1 when we tried to define proper contact conditions. Thus simulations on an overall model were facilitated.

$$F \propto E \qquad (1.)$$

F : contact force
E : Young's modulus

For more accurate prediction of the damage potential of a side impact on the occupant, the force transmitted from the seat cushion should not be neglected. In order to reproduce actual phenomena and carry out calculations in an accurate manner, we took the following precautions. First we prepared a model of six surfaces equivalent to the seat surfaces (Figure 3.). The contacts between the surfaces and the back and bottom of the FE dummy were defined by the PAM-CRASH sliding interface Type 11. In this contact type, the calculation gives the interaction force between these in response to the deflection value of a part to a plane surface, using a force - deflection curve given by user. Figure 4 shows the pelvic velocity of dummy under the conditions - test and two simulations (with the seat surfaces, without the seat surfaces). This correctly simulated occupant behavior with calculational stability and simplicity, before the contacts of the dummy and interior trim.

Comparison of Simulation Results with the Difference in the Dummy Condition

We tried to carry out three kinds of simulations to compare the simulation results with the difference in the dummy conditions. These simulations were carried out using the same conditions except for the difference in the dummy condition.

(a) Calculation without using a dummy model
(b) Calculation with a rigid body dummy model
(c) Calculation with an FE dummy model

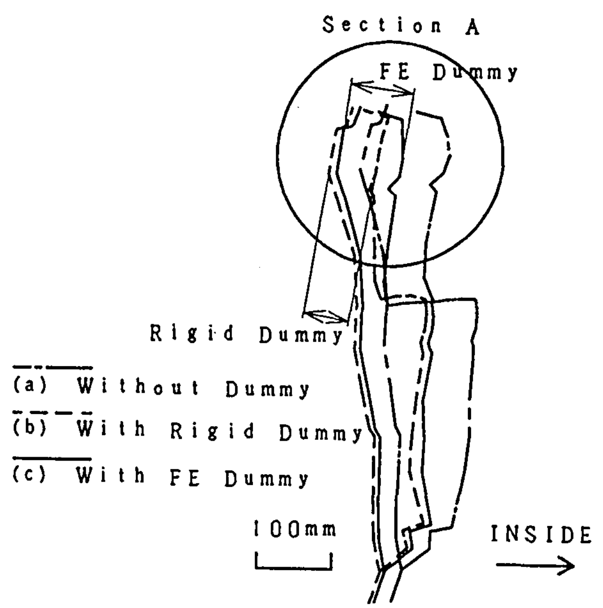

Figure 5. Cross Section of Deformed Door.

Figure 5 shows cross-sections of deformed doors taken at the vehicle longitudinal position at the time when dummy model contacts to the door under the above conditions (a), (b) and (c). Figure 6 shows deformation of the center pillar (center pillar inner panel) taken at the same time of Figure 5 under the above conditions of (a), (b) and (c). Figure 5 and 6 indicate that the intrusion into the vehicle compartment is larger when no dummy is installed (a) than when a dummy is installed (b, c). This is due to the presence of the dummy mass. Substantial differences in the degree of intrusion are present not only in the door section which came into direct contact with the dummy, but also in the center pillar which is part of the vehicle frame system. This indicates the great effect of the dummy mass in side impacts. Figure 7 shows the velocity curve at the center pillar belt line level. The dummy ribs and door trim first come into contact at t_1. With no-dummy (a) the velocity of the center pillar continues to increase as before, while if a dummy is installed (b, c) the velocity decreases (the velocity decrease a little earlier than t_1 due to the dummy hip coming into contact with the door trim beforehand - approximately at t_0). The time when the dummy comes into contact with the interior trim and the velocity of each part at that point can be obtained even from the no-dummy simulation model. The following behavior of the vehicle body, however, greatly depends on the presence of a dummy. Therefore, we cannot obtain detailed information about the body deformation and the behavior of the dummy, after the contact between the body and the dummy.

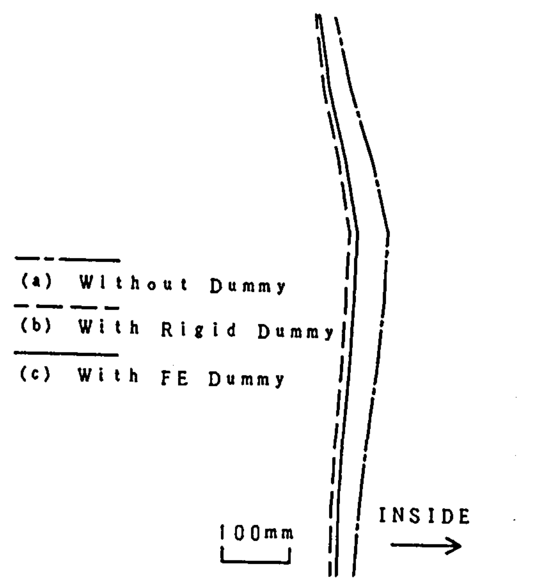

Figure 6. Deformation of the Center Pillar Inner Panel.

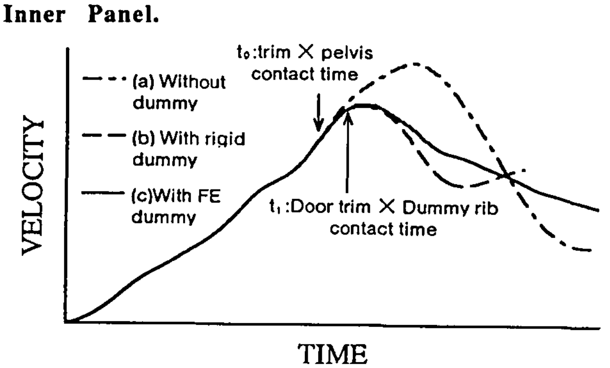

Figure 7. Center Pillar Belt Line level V-T Curve.

Next we compare the simulation results of the rigid body dummy (b) and the FE dummy (c). Although the difference is less than that caused by the presence or absence of the dummy, Figure 5 and 6 both show that the amount of intrusion of (b) is greater than that of (c). Note in particular, the door belt line (Section A) in Figure 5 section A is crashed in the lateral direction of the vehicle more severely in (b) than in (c). This depends on the dummy property being deformable or not. It is known that the deformation properties of the parts of the door greatly influence the injury index of the dummy. If, however, the dummy is rigid, the deformation properties of the door are not precisely converted into the injury index of the dummy as the resistance that the door experiences due to the rigidity of the dummy is greater than during actual collision. Figure 8 shows the velocity curve of the door inner panel taken at the dummy rib crash position. In the case of (b), the velocity decreases rapidly when t_1 at the impact of the dummy ribs and door trim. This is another indication of the unsuitability of the use of a rigid dummy for determining how much influence different door deformation properties give on the dummy injury index. In the case of (c), the velocity decreases at t_2. This is due to the contact between the arm of the FE dummy and the door trim. This decrease in velocity is not present when the rigid body dummy is used because the arm is not included in the rigid body dummy.

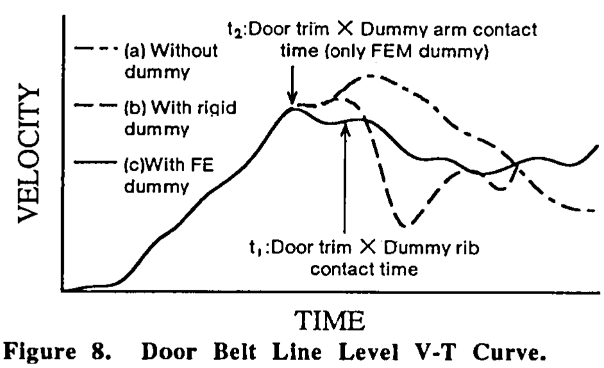

Figure 8. Door Belt Line Level V-T Curve.

In a side impact the cause of the injury is the force between the door panel and the occupant. Differences in the occupant condition lead to significant differences in the behavior of each part of vehicle. It is not sufficient to represent a side crash phenomenon without dummy. In case we use a rigid dummy model the simulation is stabler than using an FE dummy model and the calculation time is also smaller than using the FE dummy model. It is therefore partly useful to install the rigid body dummy model. However, it is certain that using rigid dummy on the simulation may influence the body deformation in a way that is different from the real phenomenon. It is also difficult to predict the occupant injury index. Installation of the FE dummy is therefore important for accurate side impact simulation.

Comparison Between Crash Tests and Simulation Results

We conducted crash tests and simulations including the FE dummy model for two cases based on the same prototype vehicle only differing in side structure strength. In these two cases, we used a prototype body (hereinafter referred to as "Case 1") and a body increased in side structure strength with a reinforced center pillar, locker and roof side rail (hereinafter referred to as "Case 2"). The body model we used to obtain three kinds of results in the previous section was different from case 1 model only in the seat position.

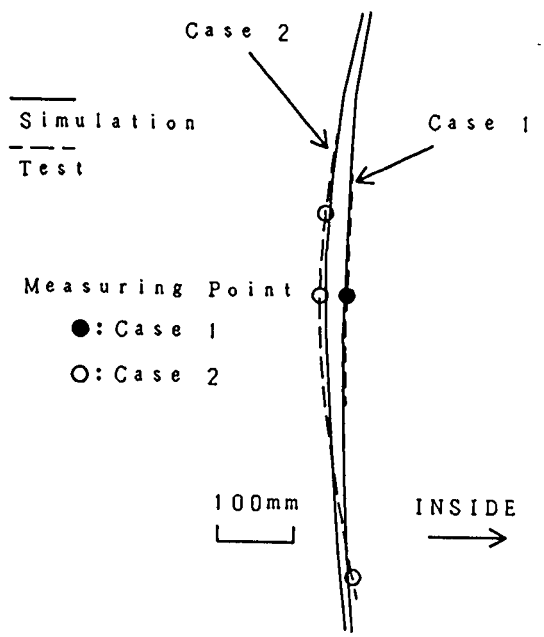

Figure 9. Deformation of the Center Pillar Inner Panels (At the Time of Impacting Dummy).

Comparison of the Deformation and the Velocity of Each Section of The Vehicle Body - Figure 9 shows deformation of the center pillar inner panel taken when the interior trim contacts with the dummy. The deformations between crash tests and simulations are approximately the same in both cases. Next, we tested and calculated V-T curves of three positions: MDB (Vmdb), unstruck side rocker panel (Vveh), and center pillar at the belt line level (Vb) are compared. Figure 10 shows the results of Cases 1 and 2, respectively. In both cases 1 and 2, there is a slight difference in Vmdb in the latter half. Nonetheless, Vb, giving the dummy a direct impact, and Vveh do not deviate on the whole, assuring that the calculation results using this body model properly simulate crash tests.

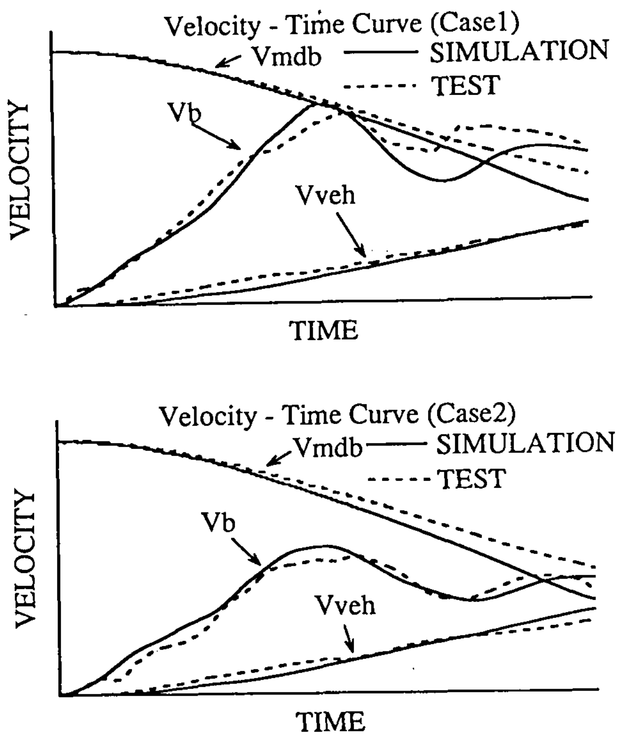

Figure 10. Velocity-Time Curve.

<u>Comparison of Dummy Injury Indexes</u> - Figure 11 shows the results of the middle rib deflection in Cases 1 and 2. The maximum values shown in both cases are approximately the same. However, there is a difference in the velocity of rib deflection (slope of curve in the graph) after contact between the dummy and interior trim. The velocity of calculated rib deflection is greater in Case 1 and smaller in Case 2 comparing with the test results.

Figure 12 shows the results of the middle rib viscous criteria (V×C) in Cases 1 and 2. Resulting from the difference in the velocity of rib deflection, these results have failed to provide an accurate simulation with regard to V×C.

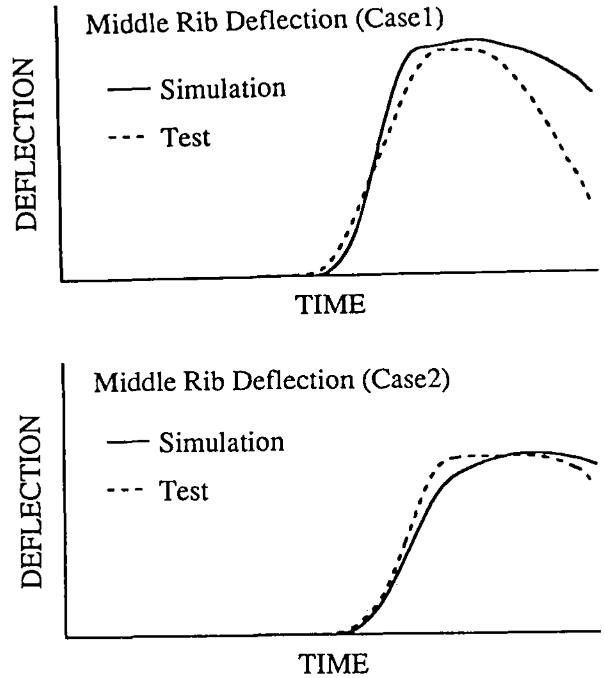

Figure 11. Middle Rib Deflection.

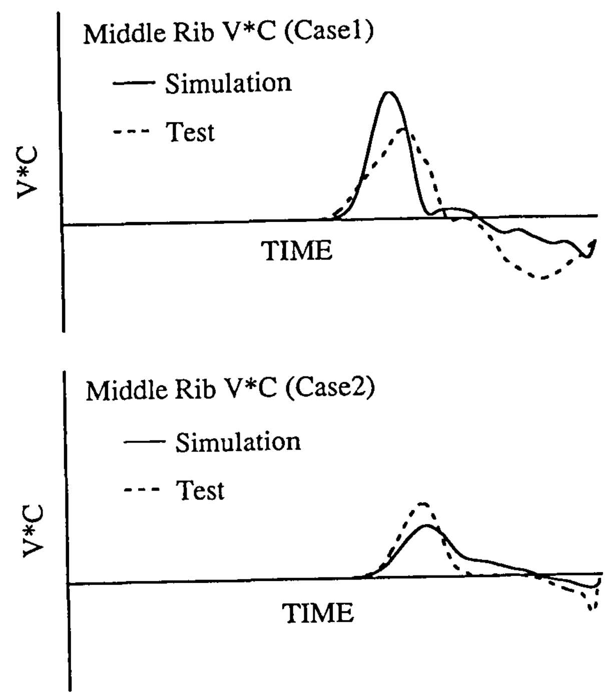

Figure 12. Middle Rib V*C.

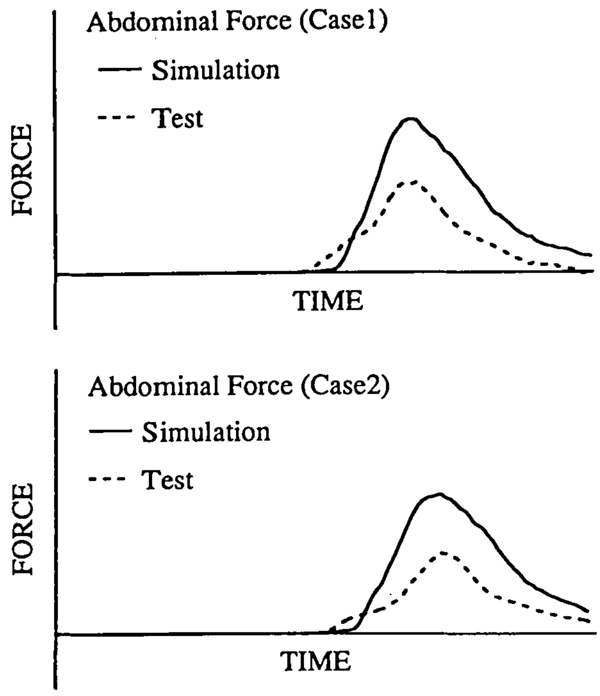

Figure 13. Abdominal Force.

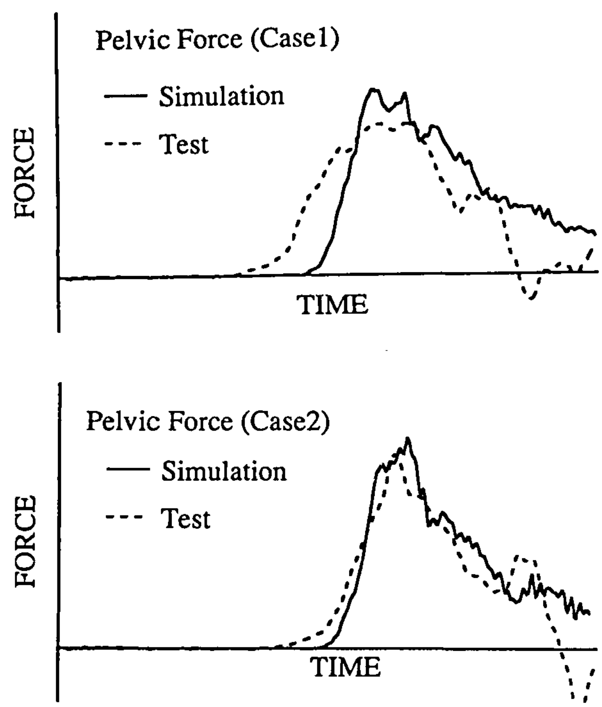

Figure 14. Pelvic Force.

Figure 13 shows the results of abdominal force in Cases 1 and 2. The maximum simulation values shown in both cases are grater than the tests values. But, in both cases, the times of the maximum values are approximately the same.

Figure 14 shows the results of pelvic force in Cases 1 and 2. In both cases the maximum values and the time when those values come to are approximately the same.

Table 1.
Each Injury Index Ratio
(Simulation/Test)

	Case 1	Case 2
Upper Rib Deflection	1.03	1.04
Middle Rib Deflection	1.04	1.02
Lower Rib Deflection	1.11	1.13
Upper Rib VxC	1.23	0.99
Middle Rib VxC	1.40	0.70
Lower Rib VxC	1.31	0.89
Abdominal Force	1.69	1.72
Pelvic Force	1.23	1.09

Table 2.
Each Injury Index Ratio
(Case 2/Case 1)

	Simulation	Test
Upper Rib Deflection	0.78	0.78
Middle Rib Deflection	0.78	0.80
Lower Rib Deflection	0.88	0.86
Upper Rib VxC	0.43	0.54
Middle Rib VxC	0.41	0.81
Lower Rib VxC	0.50	0.74
Abdominal Force	0.91	0.89
Pelvic Force	1.14	1.29

Table 1 shows the ratio of the maximum injury index simulated to the maximum injury index tested (simulation/test). We must pay attention to the correlation of each injury index ratio between Case 1 and

2. Table 2 shows the ratio of the maximum injury index of Case 2 to the maximum injury index of Case 1 (Case 2/Case 1). If the values of Case 2/Case 1 ratios of the simulation equal to those of the test, the calculation results relatively simulate the change of injury index caused by the difference of the body side structure. Except V×C, each value of Table 2 is approximately the same. These results indicate that except for V×C, for which a relation could not be established, other items can be compared with adequate relativity. The models used were sufficiently useful in calculation for predicting the effects of design refinement.

CONCLUSION

1) In computer analysis of side impact phenomena, installation of the FE dummy is important.

2) By installation of the FE dummy model, computer models can provide more effective information and data regarding the deformation and velocity of each section of the vehicle body and dummy injury indexes in ECE95 side impact.

3) In this case, from the viewpoint of the simulation results accuracy, it is difficult to make absolute evaluation of crash characteristics with the simulation model. We will continue to improve the accuracy of the body model and the FE dummy model.

ACKNOWLEDGEMENTS

We would like to thank the Transport Research Laboratory (on behalf of the Department of Transport) and the Ford Motor Company for funding the initial development of the EUROSID finite element model and the Mitsubishi Motor Corporation, Daewoo Motor Company Ltd, Rover Group Ltd and the American Automobile Manufacturers Association who co-operated in its further development.

We would also like to extend our thanks to Ove Arup & Partners for their co-operation in the development and use of the model.

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PAM-CRASH User's Manual, Version 1995.

THE USE OF ADVANCED ANALYTICAL TECHNIQUES IN SIDE IMPACT CRASHWORTHINESS RESEARCH

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ABSTRACT

This paper provides an overview of NHTSA's application of advanced analytical tools in side impact research. Finite Element (FE) approaches are highlighted as emerging simulation tools. Status of the Moving Deformable Barrier (MDB), Side Impact Dummy (SID), and passenger car FE model development efforts at NHTSA are reported. Simulated responses from these models are compared against crash test data. Use of FE models for general vehicle safety performance characterization is illustrated. Advanced lumped mass simulation models for vehicle safety performance and countermeasure characterizations are also discussed. Such models include barrier to car and car to pole impacts using rigid body dynamics and simplified lumped mass techniques in combination with FE techniques. Emphasis is placed on model development and extraction of modeling characteristics from both FE model outputs and crash tests. Future research directions are also presented.

INTRODUCTION

In the earlier days of crash research, most computer models were simple models based on rigid body dynamics. These models proved vital in simulating occupant responses in vehicle crashes. Passenger and Driver Simulations (PADS), Steering Column and Occupant Response Simulations (SCORES), and Crash Reproduction Using Static History (CRUSH) are examples of some computer models used by NHTSA in their frontal and side impact studies (1,2,3). For side impacts, the models were primarily one-dimensional (1-D) simulating the striking vehicle, its collision partner, and the driver's response in 60- and 90-degree impacts. In one of the first models, Prasad (4) used disjointed sets of rigid ellipsoids utilizing the Calspan three-dimensional (3-D) CAL3D simulator (5). Segal (6) and Tomassoni (7) preferred lumped mass representations where the striking vehicle was modeled by a single mass and the struck vehicle was modeled by two masses: the driver door and the remaining vehicle. These models were primarily intended to simulate either the driver's pelvic or thoracic responses. Segal, furthermore, used the simulated door velocities from the lumped mass models to simulate head motions from a "kinematics" CAL3D occupant model. In the mid 1980's, Trella and Kanianthra (8) accounted for the "inner-to-outer" door compliance in their modeling work for occupant injury assessments. This was based on crash test data that indicated broader and lower occupant thoracic acceleration levels when the door was not fully crushed from outside to inside at the time of occupant contact with the door. A multiple mass door and pillar model for the struck vehicle was developed to simulate the interaction between the side structure and driver occupant. Since the mid 1980's, other investigators such as Low and Prasad (9) further applied the rigid mass concept in their side impact studies. They developed a comprehensive 3-D model of the SID using the mathematical dynamic model (MADYMO) developed by Toegepast Naturwetenschappelijk Onderzeok (TNO) Crash-Safety Research Center (10). In ensuing work, these authors and Sundararajan (11) incorporated the 3-D SID model in an ellipsoidal-plane rigid mass vehicle structural model to simulate barrier to car impacts. Additional modeling details of the struck car door structure using MADYMO were presented recently by Sundararajan et al. (12) in their development of a dynamic door component test methodology for side impact simulations. Clearly, the rigid body dynamics approaches will continue to be refined and used for vehicle crash studies.

The 1980's brought forth the development of advanced FE computer programs that work efficiently and reliably for simulating automotive structures in crashes. More recently, low-cost computer workstations and corresponding FE programs have also become available. To date, a number of passenger car FE models have been reported in the literature for full and half offset frontal car to car and car to barrier impacts (13), a 90-degree side impact (14), and car impacts with fixed objects such as a pole (15). More recently, US Electricar (16) reported some limited test comparisons for a pickup and sedan in frontal-barrier impact. The crush patterns of these vehicles modified for a battery pack were also discussed for a frontal 30-degree oblique impact. To complement these activities, numerous dummy FE models in various stages of development were reported, including the

Hybrid III dummy for frontal impacts (17), and the SID (18) and EUROSID (19) for side impacts. There are also a number of FE models of automotive restraint components such as seat belts (20) and air bags (21) which can be fully integrated with the vehicle/dummy FE models (15) for frontal crashworthiness studies.

NHTSA has recently initiated research to apply FE tools in vehicle safety performance studies. One goal is to develop experimentally verified FE models and produce an analytical database of passenger car and light truck FE models. To date, two model year 1991 Ford Taurus FE models have been developed: a frontal model verified in full frontal and half offset impacts and a side model verified in a MDB to vehicle side impact (22, 23). Front-end FE models of the Dodge Intrepid and the Saturn SL were also developed for frontal impacts by West Virginia University (24). Recently, George Washington University has made available to NHTSA, an FE model of a Chevrolet C-1500 pickup. Currently, NHTSA has plans to develop a full vehicle FE model of a Dodge Neon. An FE model of the MDB used in the Federal Motor Vehicle Safety (FMVSS) 214 dynamic side impact test (25) was also developed and verified in full and 30-degree oblique frontal impacts (26). An FE model of the SID has been mainly developed (27) and will be fully verified under both rigid disc and sled impacts prior to its use in vehicle crash modeling studies.

It is worthwhile to contrast the benefits of applying FE methods in automotive safety studies versus the simpler rigid body approaches. FE techniques provide needed detail to allow a more accurate modeling of occupants and vehicles and their various contact interactions, such as with air bags. This lends insight into the development of automotive subsystems for improved safety performance, and the development of injury criteria and countermeasure assessments. However, the rigid body techniques are still needed for parametric studies including countermeasure assessments and test procedure development, and for the development of lumped mass models to be used in vehicle aggressivity and compatibility studies. Such models can be easily parametrized from the FE simulations, thus eliminating the need to run interactively the FE models for fleetwise "full system" safety optimization.

This paper reviews some of the advanced mathematical models developed for simulating side impacts at NHTSA. Side impact crashes can be broadly classified as crashes where a vehicle's side structure in the region of the door is contacted during impact. This impact can be with another vehicle or with objects such as a pole or tree. The FE model development efforts at NHTSA are presented, followed by the developments in lumped mass "concept" models. Where applicable, model verification against test data is presented. The role these models play in side impact research is illustrated through car to car and car to pole crash examples of a passenger car. Future research directions are also discussed.

FINITE ELEMENT MODELING AND SIMULATIONS

This section presents the following FE models developed for side impact crashworthiness studies:

a) MDB
b) SID
c) A passenger car in FMVSS 214 dynamic side impact test configuration

The section also presents the following FE simulation based on the above models:

a) FMVSS 214 impact of passenger car with improved MDB model
b) Normal impact of car into side of car
c) A fully integrated SID-car-MDB FMVSS 214 impact
d) Normal side impact of car into a rigid pole

The models discussed below were developed for simulation using the LS-DYNA3D FE analysis code, Version 936, developed by Livermore Software Technology Corporation (28). The analytical material codes in the LS-DYNA3D program were used exclusively to model the striking and struck vehicles and SID in the various crash situations. A minimum time step of 0.27 microseconds based on the smallest element in the deforming MDB honeycomb front structure was used for all simulations involving the car, since this time step was found to result in reasonable predictions of all energies (kinetic, potential, slide line, stonewall, etc.) within the prescribed simulation times. Vehicle side impacts other than 90 degrees, such as at 75 and 60 degrees, are not discussed in this paper.

MDB Finite Element Model

The MDB was developed for the FVMSS 214 side impact full systems dynamic test that simulates a 90-degree intersection side collision. It is an instrumented test device representing the striking or "bullet" vehicle. The MDB FE model is shown in Figure 1. It has approximately 8,500 elements (approximately 720 shell and 7,800 3-D solid) and 13,500 nodes and it approximates the geometry of the MDB test design. The shell parts include the front-face aluminum bumper cladding, rear-face aluminum bumper cladding, and the

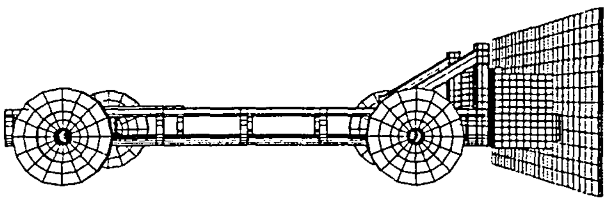

Figure 1. MDB finite element model.

front-face aluminum cladding surrounding the main honeycomb front. The 3-D solid elements are primarily rigid elements that are located rearward of the aluminum backing plate. The other 3-D elements are the honeycomb structures: the softer main front-face honeycomb and the stiffer bumper honeycomb parts. Sheet metal and honeycomb behavior were modeled with an elastoplastic isotropic material model and an empirically based metallic honeycomb model implemented in LS-DYNA3D, respectively. Tire rotation was not modeled and as a first approximation, the entire back of the main front-face honeycomb structure was rigidly attached to the aluminum backing plate in the FE model. This model was initially validated against dynamic tests of an MDB impacting a rigid wall under both normal and oblique (60-degree) impacts as a function of impact speed. See Figures 2 and 3.

Some planned improvements to the FE model include: a) experimental verification of the model at several impact angles and speeds; b) orthotropic characterization of the mechanical properties of honeycomb under axial and shear loadings in moderately and highly compressed conditions; c) development of rear-face interface conditions between honeycomb and aluminum backing plate; d) incorporation of tire rotation and the tire-roadway interface; and e) refinement of the front-face contact.

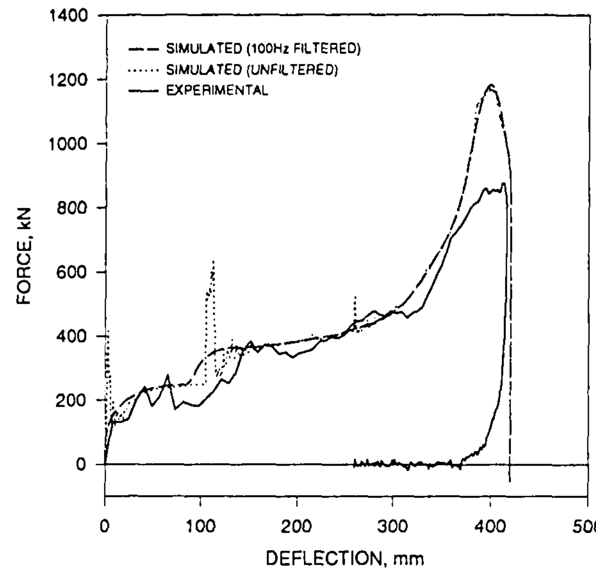

Figure 2. MDB measured and computed force versus deflection in axial direction.

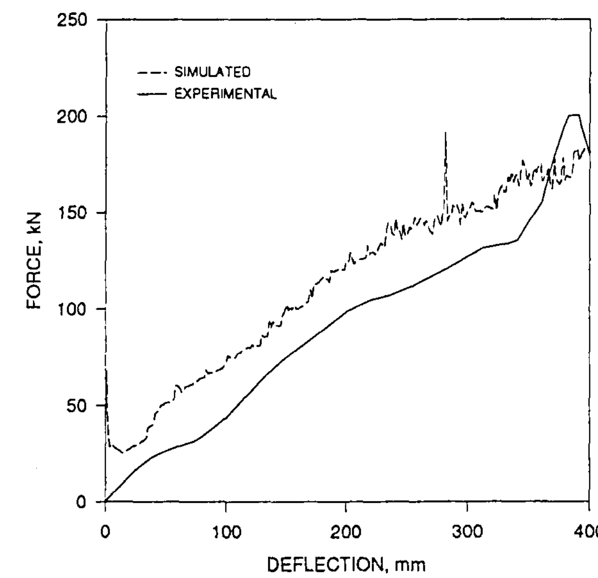

Figure 3. MDB measured and computed force versus deflection 30 degrees from axial direction.

SID Finite Element Model

In 1992, Kirkpatrick, Holms et al. (29) developed for NHTSA a preliminary FE-based 3-D model of the SID for side impact studies. The model elements were primarily 3-D 8-node hexahedron solid elements. However, there were a few shell element 4-node Belytschko-Tsay shell elements located in the thorax area. These were the ribs, jacket, and fabric surrounding the arm foam and shoulder plate, exclusively. The neck, lower spine, bump stop (anti-bottoming out pad) and cushion between the ribs and rib ballasts were modeled with a Blatz-Ko hyperelastic rubber model. The spine box was modeled as rigid to reduce computational time. Kirkpatrick and Holms chose an isotropic-elastic-hydrodynamic material model for the rib wrap and a concrete/geological material model for the arm padding since they reflect a zero Poisson's ratio at high strains (foams generally exhibit this type of characteristic). The thoracic damper was approximated as a linear damper and was modeled by two damping elements positioned in parallel between the dashpot arm and damper assembly. To allow for rotation and translation of the damper assembly during SID impact, a cylindrical joint was introduced between the dashpot arm and body, a revolute joint was used to pivot the dashpot body about the steel spine bracket, and a spherical joint was introduced between the dashpot arm and lead ballast. Since the major development effort concentrated on the thorax, parts further removed from the thorax such as the head and hip/leg assembly were modeled as rigid (non-deformable) elements. The model was verified against a pendulum impact test, using the public domain DYNA3D computer FE model.

Since 1992, based on work of Soni and Ghandi, Battelle through Quantum Consultant Inc. (27) refined the Kirkpatrick SID model by including the abdomen, anti-sag spring and flexural joints in the hip/leg area. A more realistic "low density urethane foam" material model was also introduced for the rib wrap and arm padding. Also, all rubber parts were modeled by the Mooney-Rivilin material model as an alternative to the Blatz-Ko material model in the Kirkpatrick SID. The hip/leg sections consisting of skeletal and flesh parts were also modeled, entirely, by the Mooney-Rivilin material model and not as rigid materials as in the Kirkpatrick SID model. All material interfaces (slide lines) were redefined, particularly in the thoracic area. The Kirkpatrick thoracic damper model was not refined. The extra nodes for rigid body option within LS-DYNA3D was used to tie the rubber neck and lower spine materials with the adjoining solid steel parts. Also, the aluminum hip connector was secured to the upper hip section through the extra nodes for rigid body option. Originally, Soni and Ghandi introduced spherical and revolute joints in the hip, knee, and ankle areas to model the rotational kinematics of the SID's lower extremities. A sliding interface was introduced to restrain over rotation of the adjoining parts, thus eliminating the need to include rotational stiffness at the joint. Since Soni and Ghandi were not able to simulate proper leg rotation, the slide lines were removed and these joints were replaced with spotwelds, thus forming a rigid connection in the joint area. The Battelle SID weighs 78.28 kg (172.58 lbs.) {the mean weight of a Side Impact 50th percentile male Dummy is 76.20 kg (168.0 lbs.)}. Model verification was performed for pendulum impacts in the thorax and pelvic area and sled impacts into rigid and padded walls. See Figure 4. For both thoracic and pelvic impacts, the model responses were compared against test data corridors computed from SID calibration tests at an impact speed of 4.3 m/s (14.1 ft/s), respectively. As noted in Figure 5, the Battelle SID model shows excellent agreement with the test for the thoracic pendulum impacts. However, the model was not able to accurately simulate pendulum impacts in the pelvic area and the two sled impact cases described above. See Figures 6 and 7. Battelle contends that the FE model can be improved by further rework of the mesh in the lower body area and through further improvements in the material codes.

Passenger Car in FMVSS 214 Impact

EASi Engineering developed for NHTSA two FE models of a 1991 Ford Taurus 4-door Sedan; the first for full and half offset frontal impact simulations and the second for side impact safety performance studies (22, 23). The frontal model was developed first. It utilizes a fine mesh for the front-end and a fairly coarse mesh for

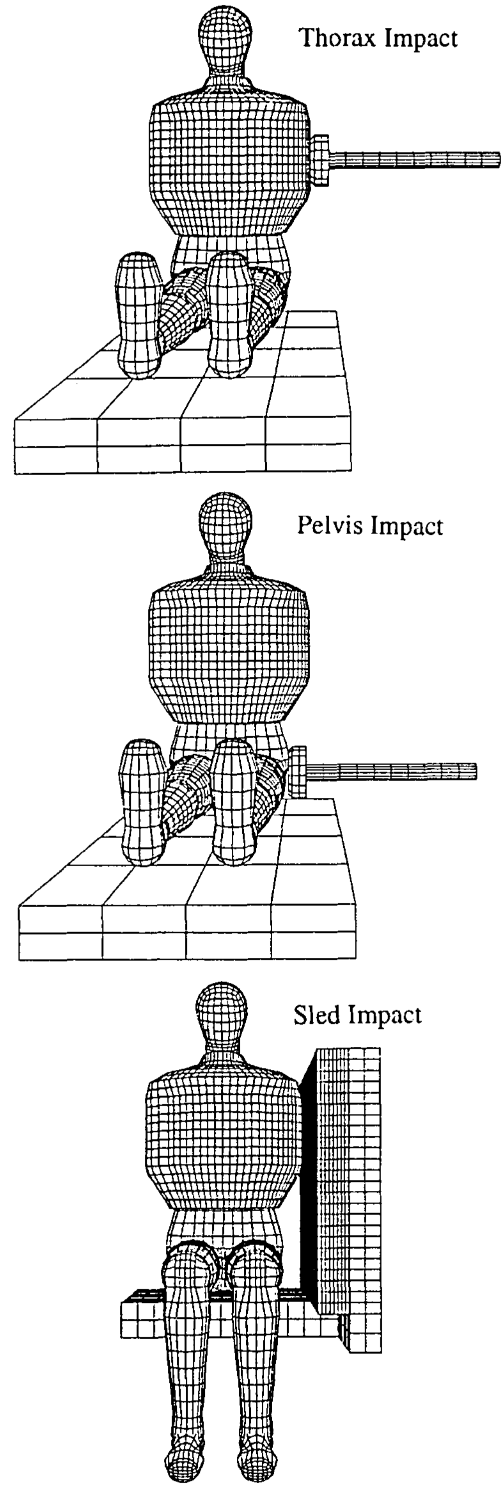

Figure 4. Impact configurations for SID finite element model verifications.

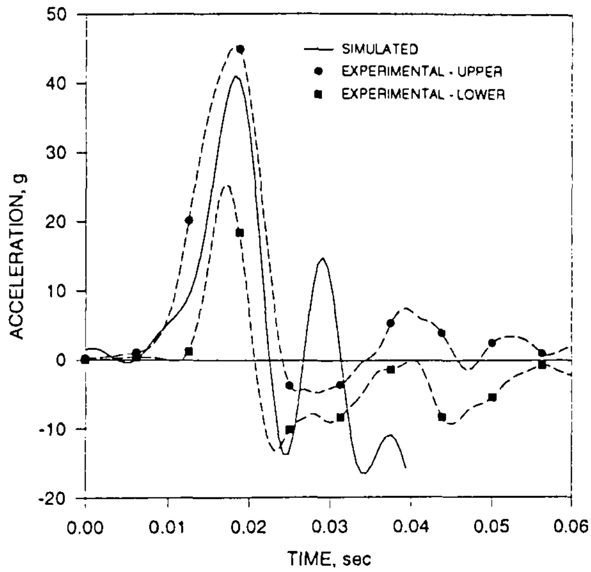

Figure 5. Thorax verification: upper rib lateral acceleration.

the rear end and has about 28,000 elements that are primarily of the shell type. The mild steel sheet metal was modeled using the LS-DYNA3D piecewise linear isotropic plasticity material code with failure. The nodal constraint option was used to model the spotwelds that included both shear and tensile strength failure. All contacts between the various structural parts of the vehicle were modeled as a single surface (type 13) contact. The frontal FE model responses were verified against a 56.3 km/h (35 mph) full barrier test and a 50% offset rigid barrier test. The model is shown in Figure 8. Follow-on efforts led to the development of the side impact model where the side structure on the driver's side was refined. Detailed definitions were included for parts such as the front and rear doors and associated hardware, and for the seat and track. Detailed definitions were also included for the A-, B, and C-pillars and welds, and for the side floor pan. See Figure 9. The side FE model contains about 48,000 elements and 57,000 nodes. To improve simulation time, all front and rear parts of the vehicle were assumed rigid in this model. The responses from the side impact model, which utilized an earlier version of the MDB FE model, were verified in the FMVSS 214 dynamic side impact test configuration, a 90-degree, 56.3 km/h (35 mph) impact, up to 60 ms into the crash. The simulated responses of the side Taurus models were also compared against test data from a break away luminaire support side impact.

MDB to Vehicle Impact Simulation

For the following simulation, the current MDB FE model described in (26) was used. The front of the MDB was positioned perpendicular to the driver side door of the

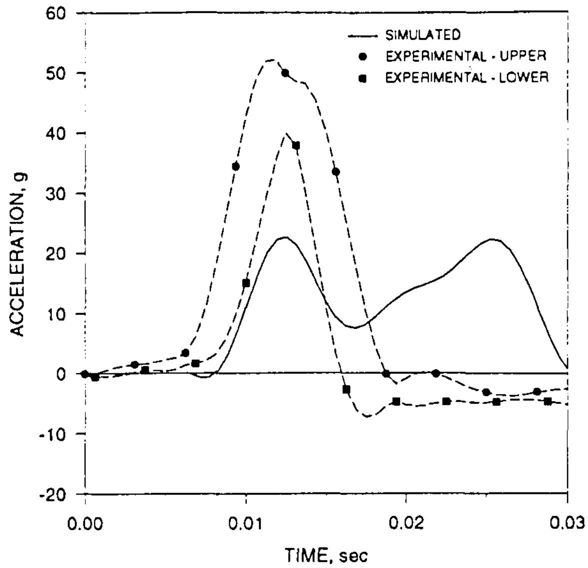

Figure 6. Pelvis verification: pelvis lateral acceleration.

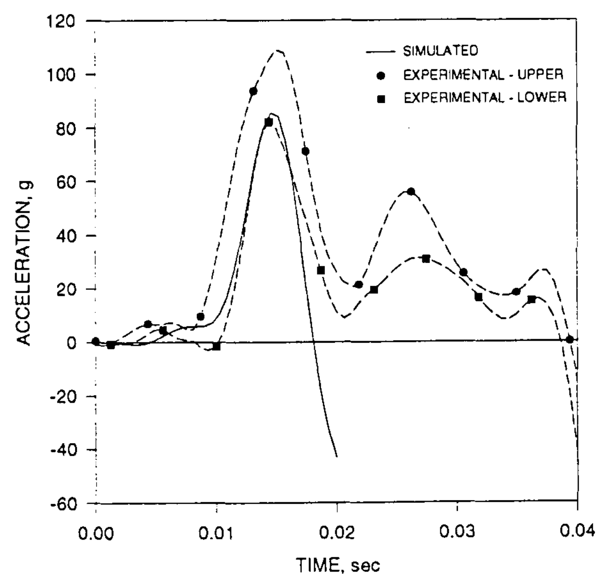

Figure 7. Sled verification: upper spine lateral acceleration.

car according to the procedures outlined in the FMVSS 214 side impact test. The simulation was carried out at a closing speed of 53.9 km/h (33.5 mph). The complete FE model consisted of about 49,000 shell and 8,000 brick elements, and 68,000 nodes. To demonstrate the simulation capabilities of the model, a 90-degree impact situation is shown in Figure 10 at various times into the crash. Certain door velocities' time histories are compared in Figures 11 and 12 against door velocity test data corridors from a series of five FMVSS 214 dynamic side impact tests (31). In this simulation, the tires of the MDB were uncrabbed. However, initial velocities of 48.3 km/h (30 mph) in the axial direction and 24.1 km/h (15

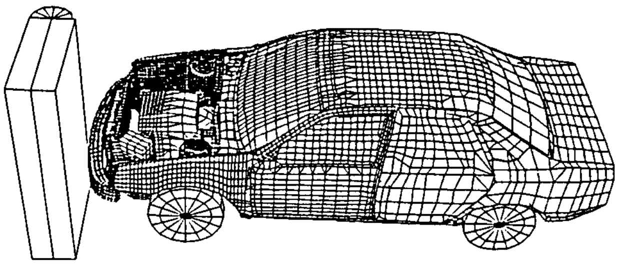

Figure 8. Frontal impact finite element model for ford taurus vehicle.

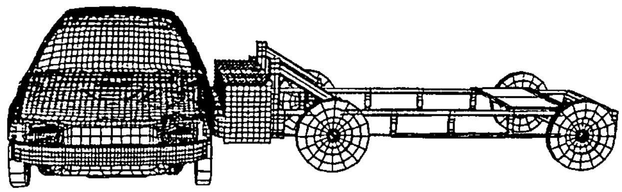

Figure 9. Side impact finite element model for ford taurus vehicle.

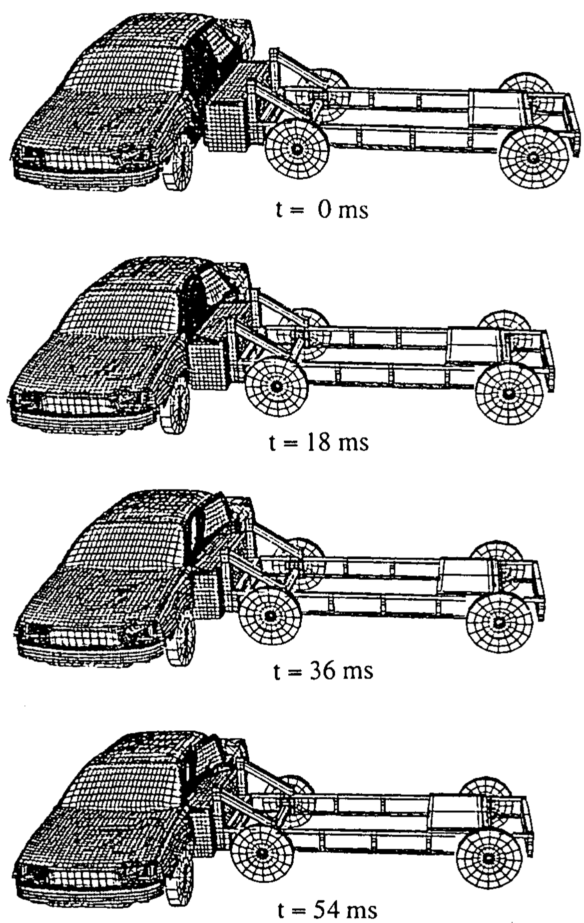

Figure 10. Simulated initial and deformed configurations for ford taurus in FMVSS 214 side impact test.

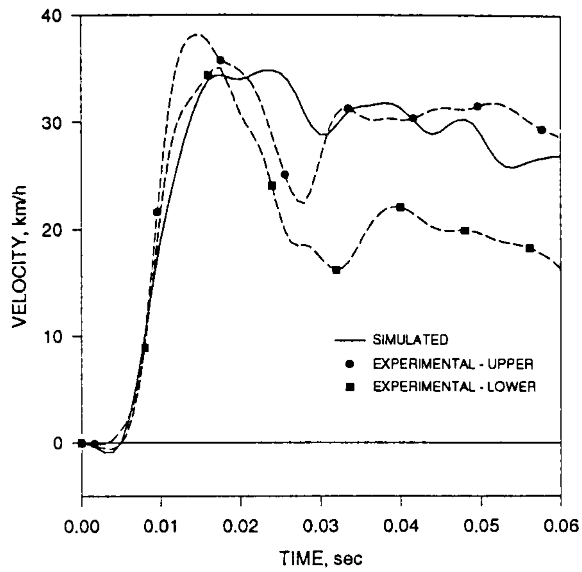

Figure 11. Measured and simulated driver door velocities above armrest.

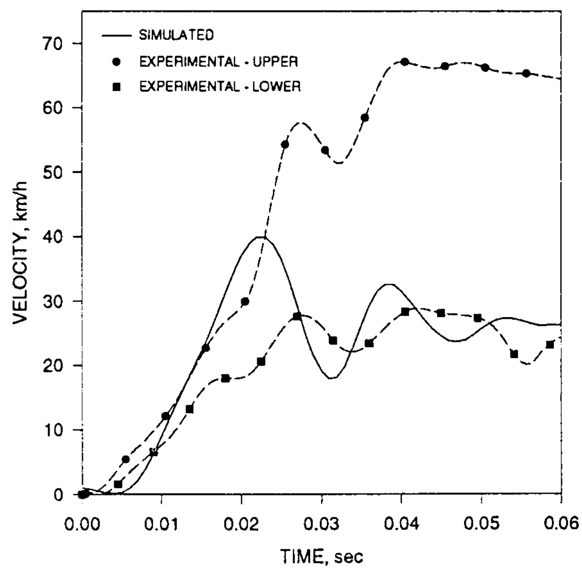

Figure 12. Measured and simulated driver door velocities at beltline.

mph) in the lateral direction were specified which approximated a wheel crab angle of 26 degrees, the angle used in the FMVSS 214 side impact tests. The contact between the various front parts of the MDB and the side structural parts of the car were modeled as a single surface (type 13) contact utilizing the volume option within LS-DYNA3D. The simulated time histories were defined in the accelerometer coordinate system and represent the time histories of a tiny rigid shell element on the vehicle structure to which the accelerometer is attached. The mass of the accelerometer was not compensated in this model. The stone wall option was used to simulate the ground. Tire force deflection characteristics based on a

24.5 kN/m^2 (35psi) tire pressure were used to model the tire interaction with the ground. Tire friction was neglected. Mass scaling was used to limit the integration time step not to be less than 0.27 microseconds.

Vehicle to Vehicle Impact Simulation

The FMVSS 214 full scale dynamic test includes an MDB with a honeycomb front of average stiffness greater than that of typical passenger cars' frontal structures, and more like the stiffness of light trucks. In this simulation, the MDB was removed from the FE model described above and replaced with the Ford Taurus "frontal" FE model (22). Simulations were performed to compare the crash performances of the car to car versus MDB to car impact. The centerline of the striking vehicle was placed in the same position as the centerline of the MDB in the FMVSS 214 side impact test. Thus, the leading left edge of the striking car's front bumper was positioned 876.3 mm (34.5 inches) forward of the struck vehicle's wheelbase centerline. This is 63.5 mm (2.5 inches) rearward of the position used in the FMVSS 214 side impact tests. The two car moving collision was modeled as a single moving vehicle impacting a stationary vehicle with initial velocities of 48.3 km/h (30 mph) and 24.1 km/h (15 mph) in the axial and lateral directions, respectively. As a first approximation, the wheels of the striking car were not crabbed. The ground was modeled as stonewall. The type 13 single surface contact was used to model all contacts among all structural parts. The complete Ford Taurus/Ford Taurus model consists of about 77,000 elements and 84,000 nodes.

The Ford Taurus to Ford Taurus side impact is shown in Figure 13 at various times into the crash. Certain door lateral velocities are compared in Figures 14 and 15 to the door velocities from the MDB to Ford Taurus impact at the same closing speed. Note that these simulations were performed on the IBM-Model 580 Power Server. The actual CPU run time was 10.1 days for a total crash time of 75 ms.

MDB to Vehicle Impact with SID Simulation

To develop a framework for occupant safety performance modeling work, the current FE SID was incorporated in the Taurus FE model. An unbelted SID was positioned, centrally, on the metallic cushionless driver seat in the Taurus model. Preliminary simulations were performed of the MDB impacting the left side of the vehicle at the 53.9 km/h (33.5 mph) closing speed to observe the SID-to-door interaction. This model was developed from the individual models described above through the LS-INGRID (Version n) preprocessor computer program and contains about 68,000 elements

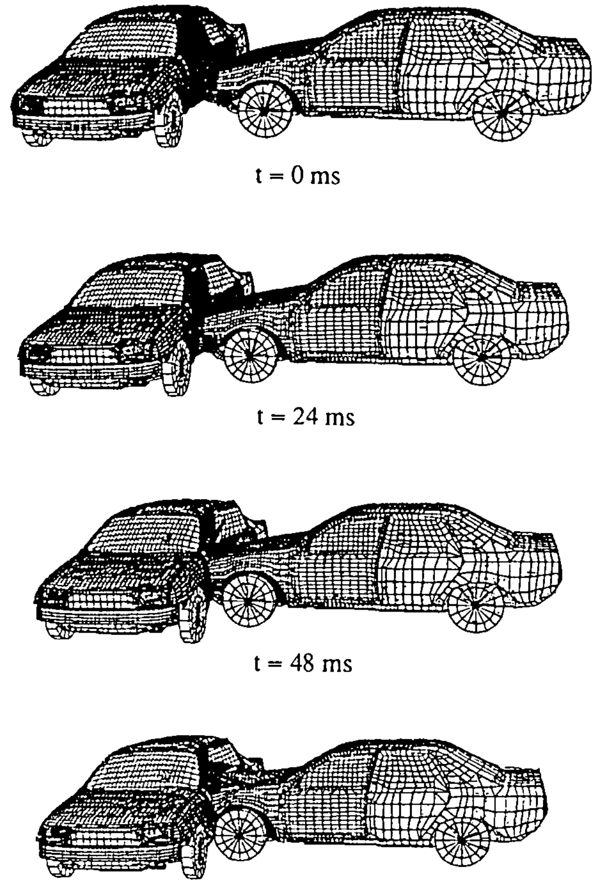

Figure 13. Simulated initial and deformed configurations for taurus to taurus side impact.

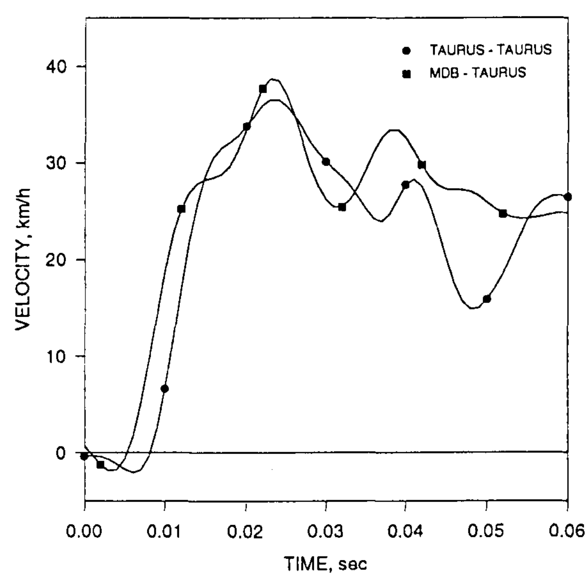

Figure 14. Simulated driver door velocities at mid door location.

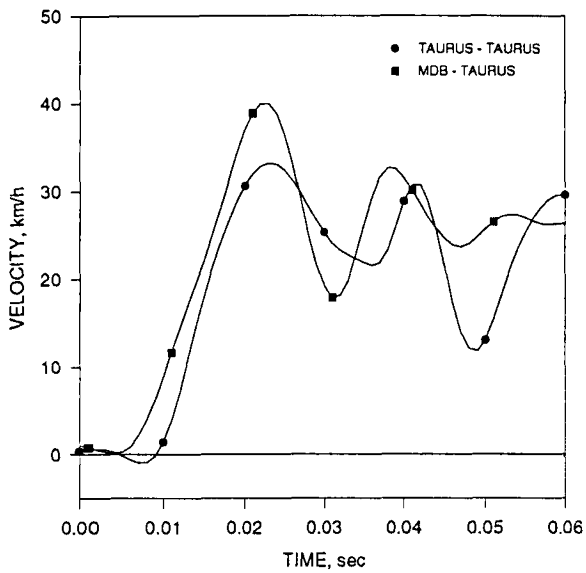

Figure 15. Simulated driver door velocities at upper mid door location.

and 85,000 nodes. As in previous models, all contacts were modeled through the LS-DYNA3D type 13 automatic single surface contact, including SID's contact with the impacted door. A sliding with void contact and no friction was used to model the interface between the SID and driver seat. A minimum time step of 0.1 microseconds was used for these simulations. Figure 16 shows the SID's motion at various times into the crash. Certain SID upper and lower rib and lower spine responses are presented in Figures 17 through 19 in which they are compared with test data corridors computed from a series of five FMVSS 214 dynamic side impact tests (31). These simulations took 18 days on the IBM-Model 390 Power Server after being prematurely terminated due to extreme compression in certain volume elements in the rib damping material of the SID after 57.3 ms into the crash.

The much higher simulated peak accelerations observed in Figures 17 through 19 are attributed to the bare-metal contact of the thorax with the interior door surface in the vicinity of the B-pillar section (simulated: 119.3 g's and 139.4 g's for upper and lower ribs, and 81.8 g's for lower spine, respectively, and measured corridor data: 59.8 g's and 65.9 g's for upper and lower ribs, and 82.0 g's for lower spine). Also, it is noted that the SID's softer parts which first contacted the door, including the jacket, arm pad and the fabric surrounding the arm pad, penetrated through the interior door during the simulations. See Figure 16. A possible reason for this behavior is the use of the Type 13 automatic single surface contact sliding interface where there are access holes to the interior space. The presence of such holes created numerical inadequacies in the contact due to edge

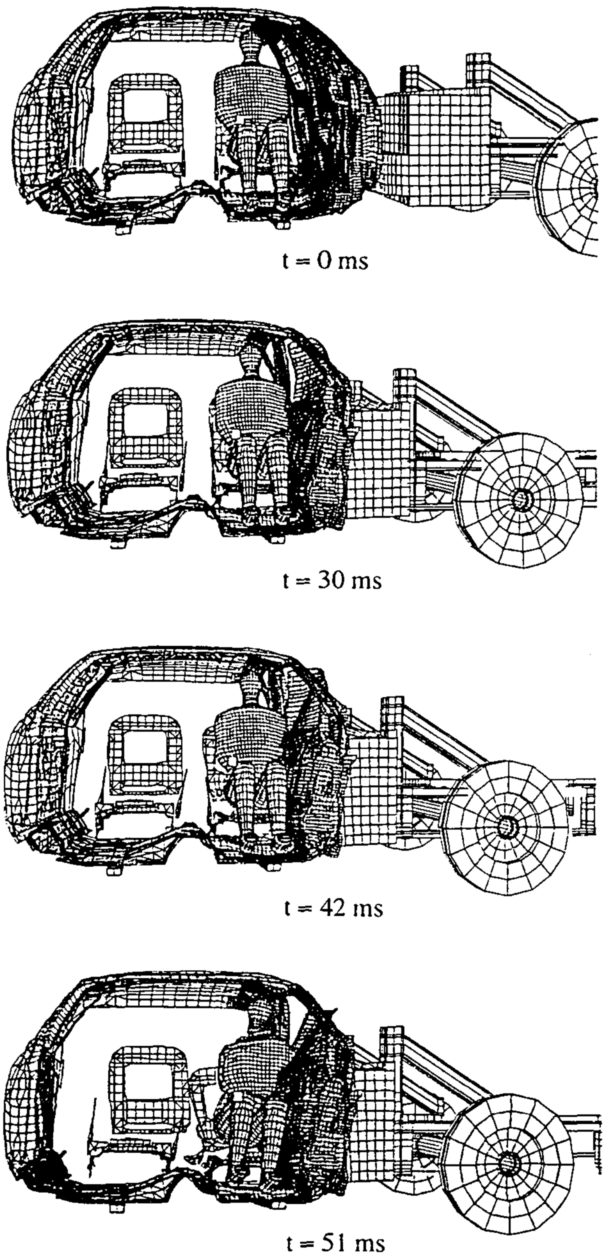

Figure 16. Simulated SID driver response for ford taurus vehicle in FMVSS 214 side impact.

effects. In this case, the Type 5 discrete nodes impacting the surface may have been a more reasonable slide line for overcoming these difficulties.

In this model, the SID was seated at an incline angle of 20.5 degrees, thus placing its upper torso parallel to the seat back. Since the seat cushions were not modeled, the SID was seated about 101.6 mm (4 inches) further rearward from the front window than in the actual vehicle. Also, in these computations, the SID was seated about 76.2 mm (3 inches) further from the door than in the test. This greater distance from the door may explain part of

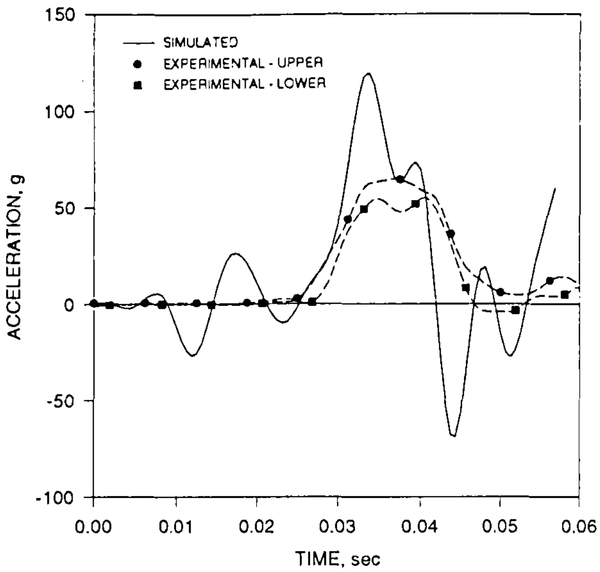

Figure 17. Simulated and measured driver SID upper rib accelerations.

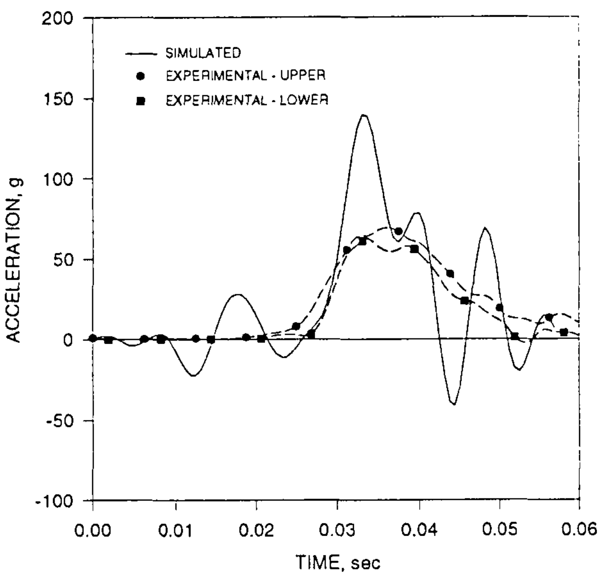

Figure 18. Simulated and measured driver SID lower rib accelerations.

the difference noted between the computed and measured SID g-responses. In addition, part of the difference between the computed and measured responses can be explained by the absence of the door trim.

Subsequent development of the FE model of the Ford Taurus is planned. This will include all of the interior surfaces and driver/passenger seat cushions, and will characterize corresponding materials and properly seat the SID in the FE Ford Taurus model. The door trim should help in reducing possible penetration of the SID's thorax through the door when using the Type 13 automatic single surface contact. Once fully developed, this model will be

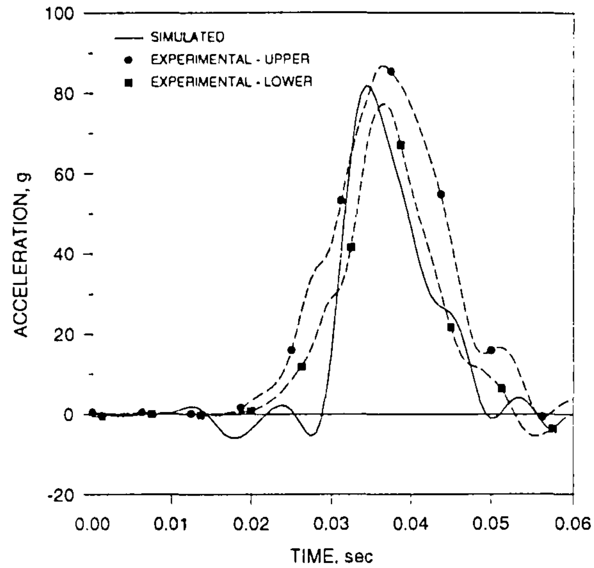

Figure 19. Simulated and measured driver SID lower spine accelerations.

used to assess vehicle design and material changes and countermeasures for occupant injury reductions.

Vehicle to Rigid Pole Impact Simulation

NHTSA, in collaboration with the FHWA, has recently conducted two side impact rigid pole tests of 4-door Ford Taurus passenger cars. The cars were laterally propelled on a rail at a speed of 36.5 km/h (22.7 mph) and 32.8 km/h (20.4 mph) into a stationary rigid half-cylindrical load measuring steel pole. These tests were intended to assess baseline test conditions and dummy performance in typical real vehicle to pole impacts, and to provide data to validate the FE Taurus model in a pole impact configuration. The lateral impact location was midway through the driver side door forward of the dummy's head. See Figure 20. In these tests, the car was tossed off the rail, thus causing the vehicle to contact the pole at a tilt angle of 4 degrees at the instant of impact.

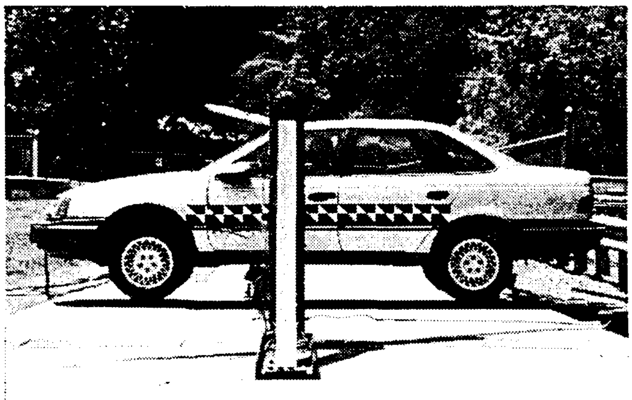

Figure 20. Pole impact crash test.

Prior to being tossed off the rail, the vehicle was about 1 inch above the ground and the vehicle's suspension was not loaded. However, just before contact, the tires were in contact with the ground for a period of 4 to 5 ms. The amount of vehicle tilt depends on the height of the vehicle above the ground and its axle size. The SID was positioned in the driver seat according to FMVSS 214 procedures.

Figure 21 depicts the FE model developed for the rigid pole impact. It contains about 57,000 nodes and 55,000 elements which are primarily shell elements. The pole, here, was modeled as a cylindrical "non-moveable" object. The side impact Ford Taurus FE model developed by EASi Engineering was used. The SID was not included in the model. The Taurus was positioned at time t=0 against the stationary rigid pole at a 4-degree tilt angle. The Type 13 single surface slide line was used to model the contact interaction with the pole and the contact interactions of the various parts within the vehicle.

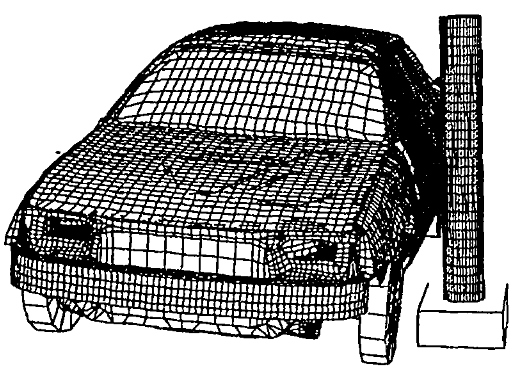

Figure 21. Rigid pole impact finite element model.

The vehicle interaction with the pole is shown in Figure 22 for a 36.5 km/h (22.7 mph) impact at various times into the crash up to 40 ms. In these simulations, the pole's centerline in the vertical direction was positioned 114.3 mm (4.5 inches) forward of the mid location of the wheelbase. As seen, the door immediately contacts the rigid pole and thus begins to crush from outside to inside. Several milliseconds later, the floor sill area also begins to crush with further door crush. This pattern continues until contact is made with the roof which begins to crush inward. Lateral penetration of the pole into the passenger compartment is established and this pattern continues until the vehicle comes to rest. Vehicle inertial forces cause an asymmetric rotation of the front and rear portions about the pole. These patterns closely replicate the patterns observed from the crash test.

Figures 23 through 26 compare several simulated velocity time histories with test data. These include velocities from different locations in the impacted area of

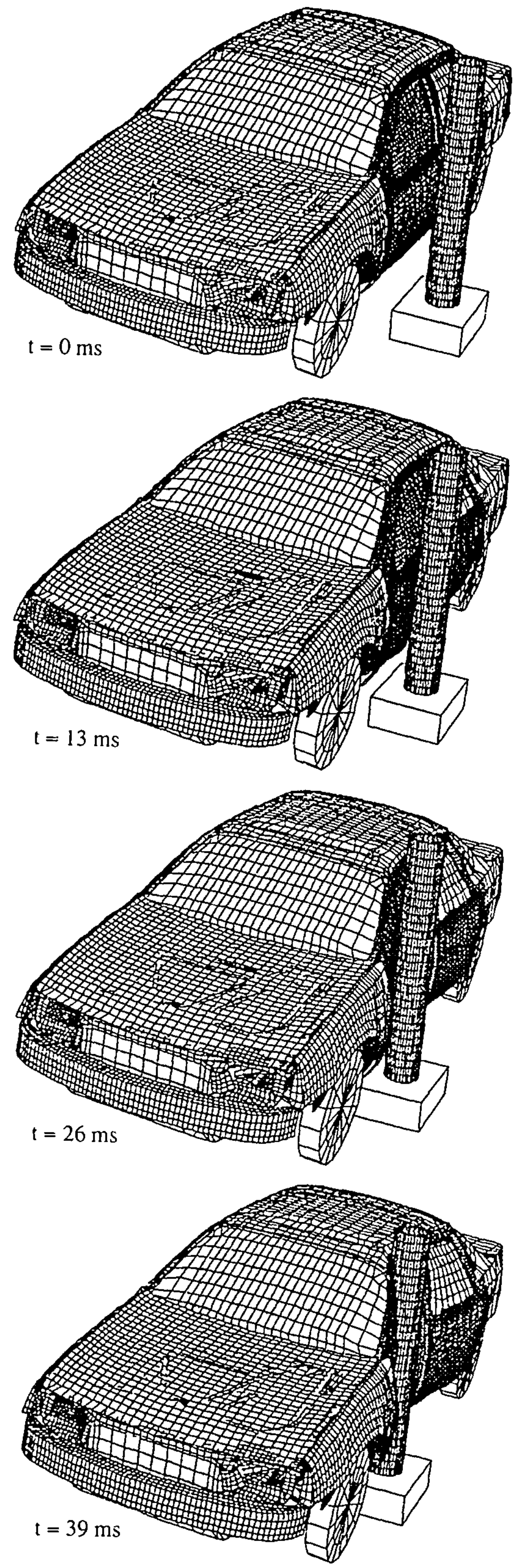

Figure 22. Simulated initial and deformed configurations of ford taurus in rigid pole impact.

the vehicle, specifically those on the door, pillars and floor structures. A good match between the FE simulation output and test data is noted for these responses. The simulated velocities used for comparisons here were defined in the accelerometer coordinates.

Once the interior side and roof header trim and seat have been developed, simulation studies with the SID driver included in the upgraded Ford Taurus FE model are planned. The goal is to evaluate the effect of various structural changes and the potential of air bags for mitigating head, thorax and pelvis injuries.

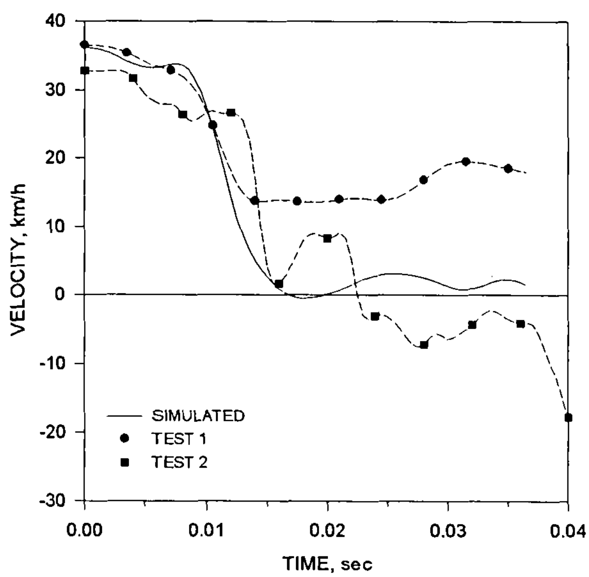

Figure 23. Simulated and measured driver door velocities at shoulder contact.

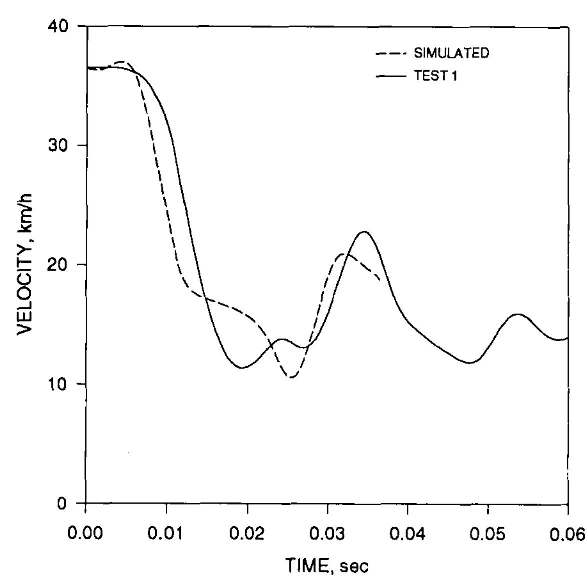

Figure 24. Simulated and measured driver door velocities at knee contact.

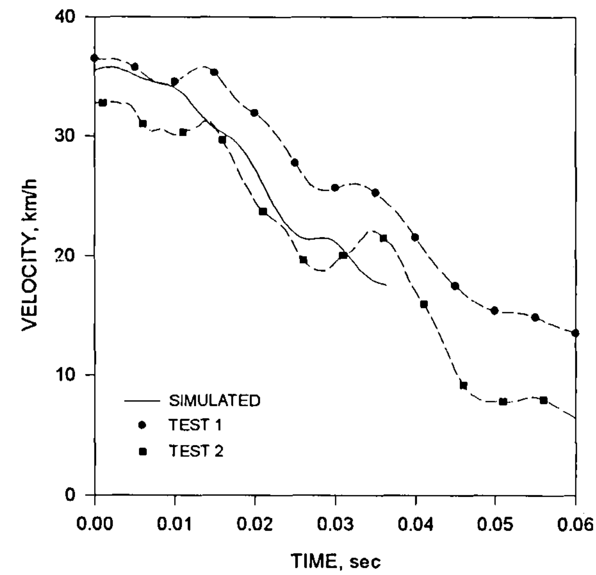

Figure 25. Simulated and measured left b-pillar velocities at mid location.

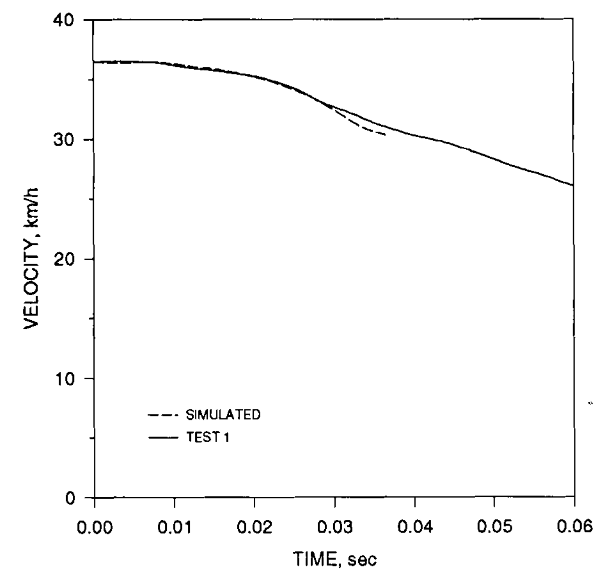

Figure 26. Simulated and measured rear deck velocities.

APPLICATION OF FINITE ELEMENT SIMULATIONS

The energy absorbed by the various structural members of the car was computed for the MDB, rigid pole, and the car to car impacts. The objective was to identify the structural members which absorb most of the energy during collision. This information aids in defining the major energy flow paths in the vehicle during crush and provides guidelines in establishing lumped mass models for safety performance studies. A second objective was to determine the absorbed crash energy distributions in the different car impacts and to compare such distributions as a function of collision type. Note, that the absorbed energy density, a measure denoting how efficiently the crash energy is absorbed by critical members during the crash, was not investigated in this paper.

The energy absorbed by the car during the three different collisions was grouped to represent the following five (5) major structural areas:

Group 1:	Front and Rear Driver Side Doors
Group 2:	Near Side Pillars (These include the A-, B- and C- pillars)
Group 3:	Floor and Near Side Floor Sill
Group 4:	Roof Structure
Group 5:	Remaining Structure

The structural members in Group 1 consist of the outer and inner panels, hinge plates, hinges, window, door beam, window mechanism, and attachments, among other members. Group 3 members include the driver side floor pan and reinforcements, side rail, and beam and sill elements. Group 5 members include driver and passenger seat, far-side door and pillar structure, far-side floor structure and sill, rear deck, quarter and wheel panels, parts directly behind the A-pillar such as the steering column and steering wheel, fire wall, part of the inner and outer hood structure.

Since the energies in each of the groups are varying with time, it is important to compare these energies at different times for the different crashes. Figure 27 displays the five groups of absorbed energies as a function of time for the MDB to Taurus at the 53.9 km/h (33.5 mph) closing speed. As a benchmark, the 15 ms time approximates the time of occupant contact with the door and the 40 ms time represents the simulation end time. In this figure, the impacted door shows the greatest amount of energy absorbed at the 15 ms occupant contact, 43% of the total absorbed energy compared to the energy absorbed by the other groups. The door continues to absorb a significant amount of the total energy, 33% even at 40 ms. It is also interesting that the rear door absorbed slightly more energy than the front door; 53% compared to 47% of the total energy absorbed by the impacted side doors. The second most significant energy absorbing group appears to be in the area where the bottom of the bumper contacts the vehicle; the rocker panel, floor and sill area, which absorbs between 20 to 23% of the total absorbed energy. The percentage of the total energy absorbed by the pillars remained constant over this time period, peaking at 20% at occupant contact and slowly decreasing to 16.5% at 40 ms. As expected, the energy absorbed by the remaining structural parts such as the dash, firewall, steering wheel and column increased from 8.4% at 15 ms to about 19% at 40 ms since these parts do not deform until later in the crash. Note, that the MDB absorbed between 7 to 10% of the total absorbed energy with lessor amounts of energy being absorbed during the initial stages of crush than in the later stages of the crush. Similar absorption capabilities of the MDB have also been observed in side impact studies (32, 33) previously performed by the authors.

In the rigid pole impact case the situation is different, as expected. Figure 28 displays the various groups of absorbed energies from 10 ms through 40 ms into the crash. Here, the doors show the greatest amount of energy absorbed, 78% at 15 ms compared to the other groups, and continue to absorb a significant amount of the total absorbed energy, 51% even at 40 ms into the crash. In this case, the front door, the part of the vehicle impacting the pole, absorbed most of the energy, approximately 80% of the energy absorbed by the doors. However, as the pole penetrates into the car's structure later in time, the near side floor sill area begins to absorb a greater amount

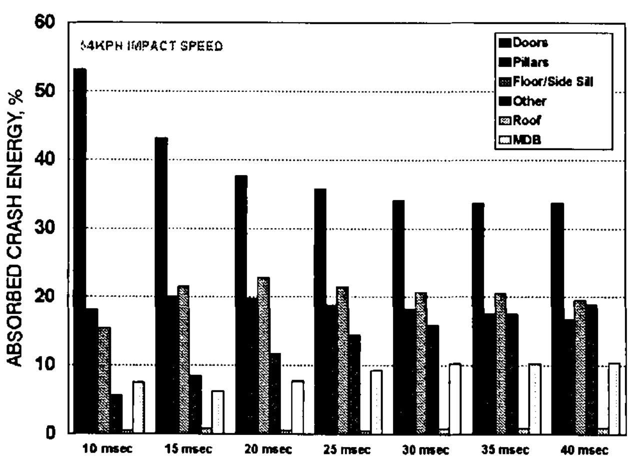

Figure 27. Absorbed crash energy (%) for MDB to ford taurus side impact.

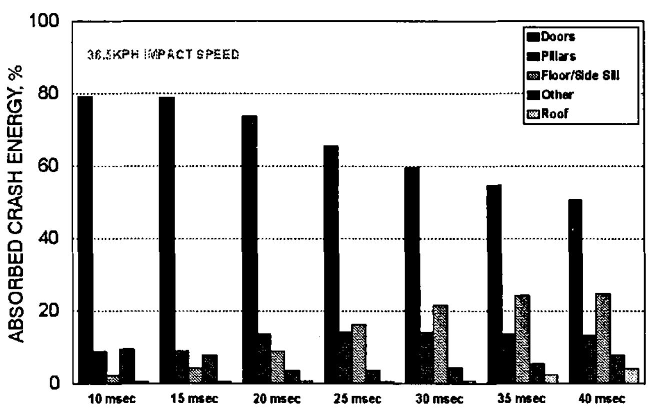

Figure 28. Absorbed crash energy (%) for ford taurus to rigid pole side impact.

of the total absorbed energy at 24.5% relative to the 7% at the time when the occupant made contact with the side structure. As expected, later in the event, the energy absorbed by the doors decreases significantly and more energy is absorbed by the other groups, particularly, the near side roof and floor sill. As in the previous case, the percentage of energy absorbed by the pillars appears to be constant around 13.5% for the simulation times considered here.

These energy distributions provide a valuable guide in assessing vehicle safety performance for different crash modes. It is apparent that the crash is more severe in the pole impact where the driver side door (i.e., the surface impacting the side of the driver's body) absorbed 55% of the total absorbed energy at 20 ms for the 35.4 km/h (22 mph) impact speed, whereas the driver side door in the perpendicular MDB impact absorbed 19% of the total absorbed energy at 20 ms for the 53.9 km/h (33.5 mph) closing "impact" speed. It appears that, in the pole impact case, the driver door is less compliant at the time of contact with the driver, thus potentially creating greater injury to the driver. Not surprisingly, these calculations show the car driver side door beam to be more effective in absorbing the crash energy in the pole collision than in the perpendicular MDB collision where the impact loads are distributed over a greater area of the side structure. For example, in the pole impact, the inner door panel and door beam absorbed about 20% more energy in the front door area than they did in the perpendicular MDB impact. See Figure 29. These distributions, also, suggest that for lumped mass modeling purposes, the MDB to car 90-degree collision should include energy absorbers and masses which take into account the door (inner as well as outer panels), floor sill, pillars and the remaining structure. The pole impact appears to be best represented by masses and energy absorbers which include the impacted door, floor sill and roof in addition to the remaining car structure.

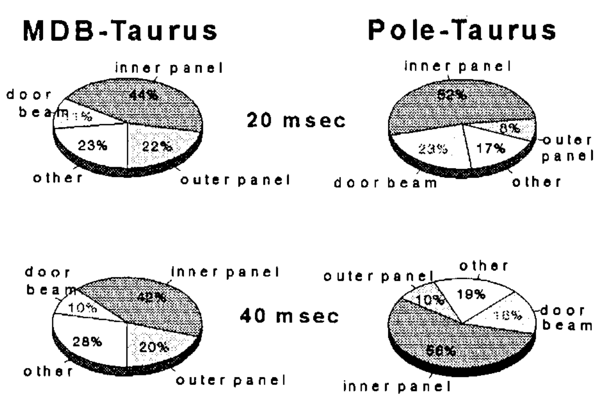

Figure 29. Absorbed energy for driver side door.

Lastly, the absorbed energies are presented in Figure 30 for the Ford Taurus to Ford Taurus side impact. The trend in the absorbed energies appears to be similar to those absorbed in the case of the MDB impact. However, the striking vehicle, here, appeared to absorb slightly more energy than did the MDB (about 10 to 15% compared to 6 to 10% for the MDB). Most of the energy absorbed by the striking car was during the initial rather than later stages of crush. Also, the door absorbed 10% more energy in this collision initially, and was the dominate structural group later in the crash.

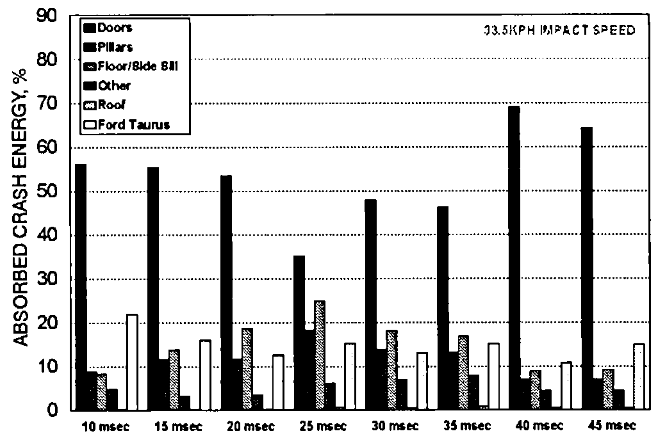

Figure 30. Absorbed crash energy (%) for taurus to taurus side impact.

LUMPED MASS SIMULATIONS

Past car to car lumped mass simulation models were 1-D with the ability to simulate the interaction of a striking MDB and struck car side structure (door) and seated dummy in the struck car in the lateral direction. A typical model is shown in Figure 31. These models were well suited to study the sensitivity of occupant responses

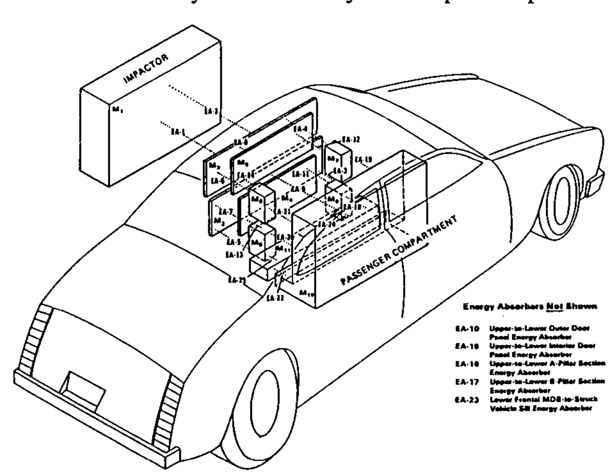

Figure 31. Earlier lumped spring/mass model for side impacts.

to parametric changes in occupant seating position and in certain aspects of car design (i.e., door padding, A-, B-pillar strength, door thickness, etc.). Since these models were 1-D models, they did not take into full account the detailed crushing of the struck car's side structure and the kinematics of the occupant during side impact. Thus, recent efforts were undertaken to develop advanced multi-dimensional lumped mass computer models for more detailed safety studies. Again, the goal was to investigate the effects of structural improvements and mitigation concepts in order to minimize the potential for occupant pelvis, thoracic and head injuries. Some of these advanced lumped mass models are presented below for car to car and car to pole simulations utilizing MADYMO procedures. A new methodology is introduced for modeling car to pole impacts which makes use of crash test and FE data extraction procedures for the development of the energy absorber's force deflection characteristics in a 1-D reference system.

Thus far, the models described include the front seated occupant on the struck side, however, they can be expanded to include the rear seated occupant as well as the adjacent structure on the struck side.

MDB to Vehicle Lumped Mass Model

Initial efforts concentrated on the refinement of the earlier 1-D side struck vehicle models to include a belted 3-D SID. The overall objective was to investigate various air bag and door padding concepts to mitigate thoracic injuries. The local rotational degrees of freedom were added to the struck car side body mass elements, in particular to the door and pillar sections. Also, a simple model for the vehicle's suspension and tires was introduced and the interaction of the tires with ground was included. See Figure 32.

The MDB is represented by seven rigidly connected masses (impactor front face, bumper, main body, and four tires) as a 6-degree of freedom "total" rigid body system. The tires are allowed to rotate and interact with the ground through ellipsoid-plane contacts. The Ford Taurus vehicle is also modeled as a 6-degree of freedom system

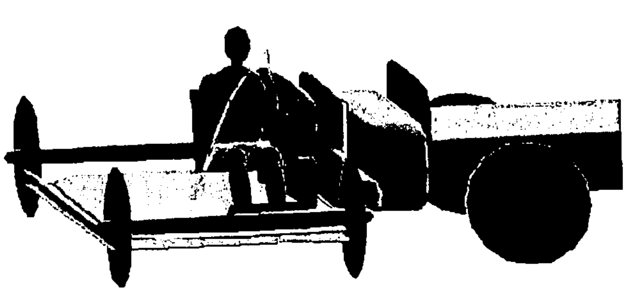

Figure 32. MDB-struck car lumped mass model.

and is represented by sixteen masses: a) main body; b) upper and lower outer struck door section; c) upper and lower inner struck door section; d) lower and upper A-, B-pillar sections; e) floor sill/board; and f) front and rear axles; and g) tires. Each mass in the vehicle and MDB is described by a rigid ellipsoid. The door and pillar sections are allowed to rotate about their roll axes through the action of planar free-joints. About 35 non-linear energy absorbers, represented by rate sensitive (dynamic amplification) Kelvin elements, were positioned between the various door, pillar and main body rigid ellipsoids of the struck car. The energy absorber characteristics were approximated from car crash and component tests and engineering judgment was used where data was not available for certain energy absorbers. A simple spring-damper arrangement was assumed for the suspension system. Planes were rigidly attached to the outer, inner upper and lower door ellipsoids to simulate MDB front honeycomb face and SID contacts with the stationary struck vehicle's door. The lumped mass SID used in this refined side impact model was the most recent Version 2.1 TNO-MADYMO SID. The MDB contacts are through two plane-ellipsoid contact; bumper-to-lower outer door mass and upper front face honeycomb section-to-upper outer door mass. SID body contacts with the door are between the lower torso to lower inner door, left arm to upper inner door, and left thigh, leg, post knee, lower leg and foot to lower door. The ground was modeled as a plane and the tires allowed to interact with the plane through a specified plane-ellipsoid force-deflection contact curve.

In this model, the 3-point belt system consists of a lap belt, shoulder belt, retractor, appropriate attachments (anchor points) and D-shape slip-ring. Two belt models were considered: a line segment belt model and a FE approach to simulate its response and coupling with the TNO-MADYMO SID. In the FE approach, the belts were modeled by line segments for the belt parts lying close to the attachment points and the D-shape slip ring, and by trapezoidal shell elements for the parts of the belt pressed against the SID torso and abdomen body parts. As a first try, a 2-element wide belt design was selected. Prior to simulations, the FE belt was fitted on the TNO-MADYMO SID using special procedures developed by TNO (10). This required positioning, dynamically, the belts around the torso and hip areas of a "hypothetical" stationary rigid SID occupant in the vehicle by slowly pulling the belts at the attachment points around the two body part ellipsoids. To achieve this, a multistage process was used which required pulling the belt in different directions until the belts fit snugly around the SID. Since the belts were modeled as two-element wide belts, the elements next to the line segments were made rigid in order to prevent the belt from creasing in the middle as it was being drawn around the SID.

Simulations performed for a 48.3 km/h (30 mph)/24.1 km/h (15 mph) striking/struck vehicle impact with the line belt model produced SID filtered peak g-levels of 104.9 g's for the pelvis, 66.9 g's for the upper rib and 83.6 g's for the lower spine. These results compare very favorably to those from the crash test (30): 103.4 g's for the pelvis, 67.9 g's for the upper rib and 84.6 g's for the lower spine. This resulted in a simulated TTI of 78.7 g's compared to the measured TTI of 76.3 g's. The simulated pelvis, lower spine and rib acceleration time histories are shown in Figures 33, 34 and 35 where they are compared to the test acceleration's time histories.

Future efforts will focus on incorporating various air bag geometries for countermeasure studies, and on further developments of the FE belt, since the authors' earlier efforts with the 2-element wide belt in another vehicle indicated higher TTI g-levels than those simulated without the belts and with the simple line belt model.

Rigid Pole Lumped Mass Models

Simple Lumped Mass Model - The simple lumped mass model shown in Figure 36 provides a 1-D structural simulation of the lateral impact of a passenger car into a rigid stationary pole. The model includes a belted 3-D SID. The purpose of this simple model is to provide a lumped mass representation for the development of force deflection characteristics for more complex simulation models. The passenger car is modeled by eleven masses interconnected by non-linear energy absorbing elements.

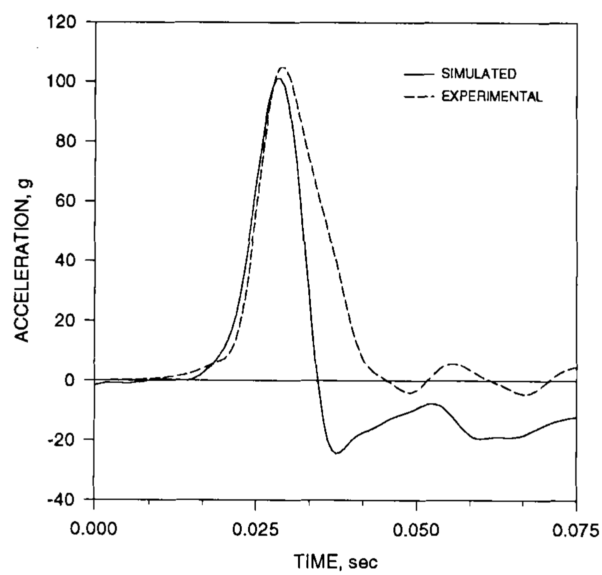

Figure 33. Comparison of simulated and measured pelvis g's.

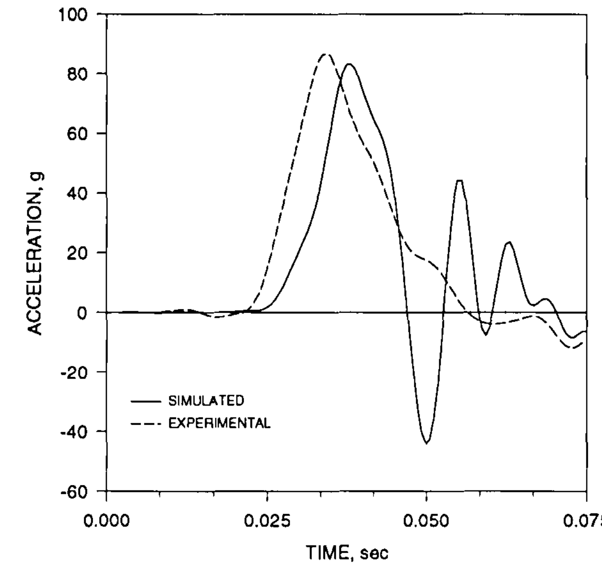

Figure 34. Comparison of simulated and measured lower spine g's.

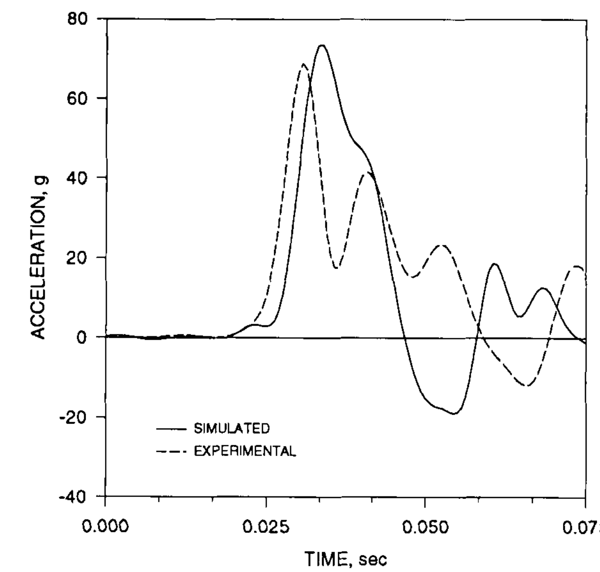

Figure 35. Comparison of simulated and measured lower rib g's.

Five masses are used to model the response of the door: the upper, middle and lower outer door sections, and the middle and lower inner door sections adjacent to the SID's pelvis and thorax body parts. The A- and B-pillar vertical sections are represented each by two masses, and the floor sill and roof rail sections are represented each by a single mass. The remaining three masses represent the vehicle's front, middle and rear sections (Note: In Figure 36, front and rear vehicle model masses are not shown). The pole is modeled as a single mass. The SID is modeled as a belted 3-D MADYMO-SID. The door, pillar, sill, roof rail, and vehicle's middle section are interconnected by non-linear absorbers. Shear non-linear absorbers interconnect the front and rear with the middle section vehicle masses, and the upper and lower door sections. Pole contact is modeled with non-linear energy absorbers interconnected to the outer door sections, floor sill, and roof rail. The energy absorbers in this model can be grouped as following:

Group A:	Outer Door to Pillar, Roof Rail, and Floor Sill
Group B:	Outer Door to Inner Door
Group C:	Pillar, Roof Rail, Floor Sill, and Inner Door to Compartment
Group D:	Inner Door to Pillar, Roof Rail, and Floor Sill
Group E:	Front and Rear to Middle Vehicle Shear Coupling
Group F:	Pole to Outer Door, Roof Rail, and Floor Sill

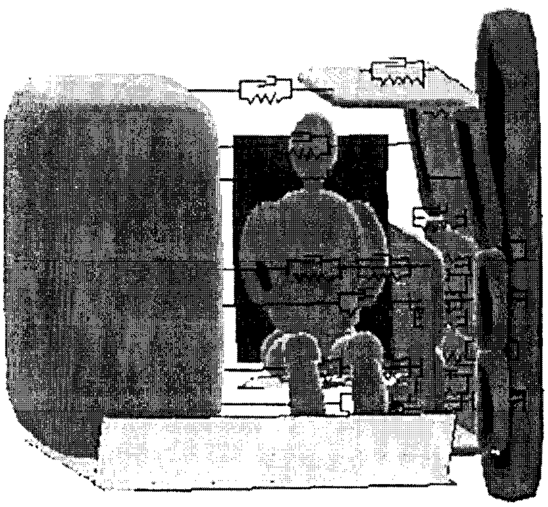

Figure 36. One-dimensional pole-impact lumped mass model.

A new approach will be adopted for the characterization of the energy absorbers in this model. Initially, crash test time histories will be used along with FE model outputs to derive the most optimal force versus deflections for the energy absorbers for lumped mass models of this type. The FE model simulation will be used to supplement the collected test data by providing time histories not captured by available instrumentation. In most instances, it is not practical or feasible to fully instrument the crash test vehicles, specifically near the impact zone. The energy absorber characteristics will also be derived solely from the FE model outputs and then compared to the combined crash test/FE model output derived characteristics. The energy absorbers with common characteristics will be identified and then used in successive iterations to establish the optimum characteristics for the remaining absorbers. In this approach, the Structural Impact Simulation and Model Extraction (SISAME)* system identification methodology, developed by Menzter, Radwan, and Hollowell (34), will be applied to derive the "optimum" force versus deflection characteristics for the model's energy absorbers. The objective is to best reproduce the measured occupant responses.

As a start, SISAME was used to derive the force deflection characteristics for the simple model shown in Figure 36, using available accelerometer time history data from the pole crash test. Data from various regions on the impacted door, sill, roof rail, A- and B- pillars and far side floor sill were used for an 8 mass, 19 non-linear energy absorber representation. The masses in the model represented the full outer door and inner driver door panels, the left floor sill, upper and lower sections for the impacted side A- and B- pillars, and the occupant compartment. The motion of the outer door was estimated from the inner door panel data and factors such as the depth of the door, the initial pole contact time, and the time of the door collapse (total collapse of the door was assumed). Figure 37 shows the extracted force deflections for the main energy absorbers. These represent the major force paths in a vehicle structure in this type of crash,

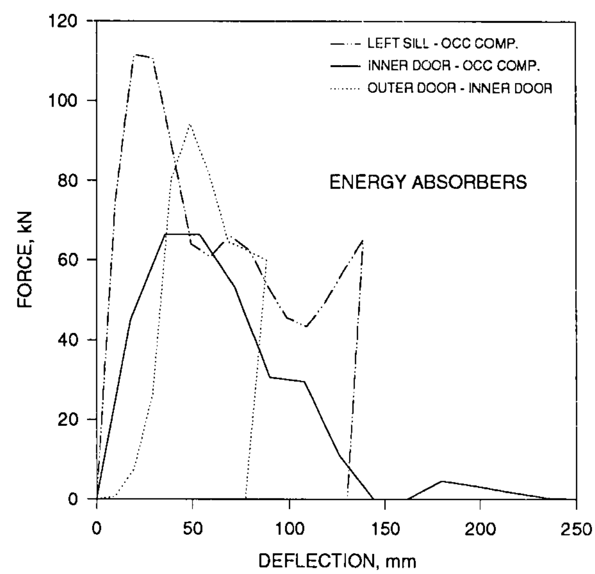

Figure 37. Derived force versus deflections from SISAME.

*SISAME is a general purpose tool for extracting and simulating large deformation crash events. It applies global, constrained least squares optimization techniques to extract the nonlinear modeling characteristics. Also as an option, it extracts the weight of the mass elements directly from motion time histories. Estimates for, and constraints on, the extracted parameters can be specified. To date, at NHTSA, full and offset frontal crash models have been extracted from barrier crash test data with the SISAME methodology (34, 35).

namely the floor sill and door areas. In this development, Figures 38 and 39 show the simulated and measured velocity time histories for the eight masses as a measure of the goodness of the approximated energy absorbers from SISAME.

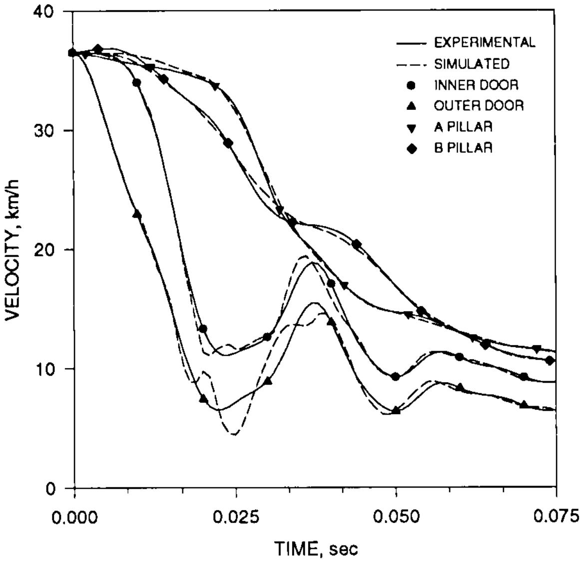

Figure 38. Comparison of calculated and measured velocity time histories from SISAME.

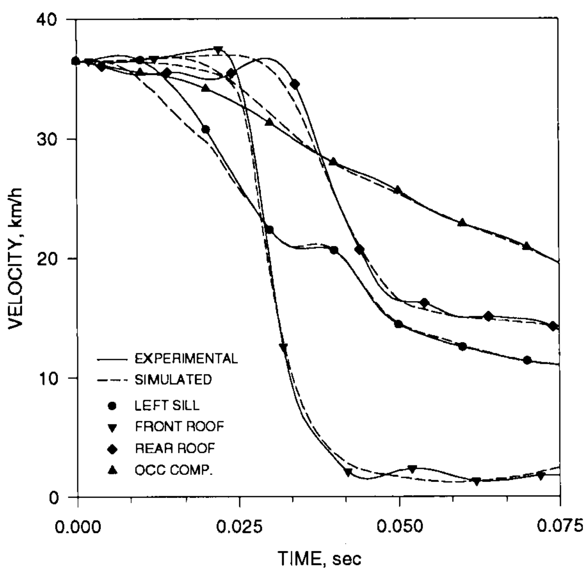

Figure 39. Comparison of calculated and measured velocity time histories from SISAME (Continued).

<u>Multi-Dimensional Lumped Mass Model</u> - A multi-dimensional vehicle lumped mass representation of a rigid pole impact is shown in Figure 40, characterizing the response of a belted SID driver in a passenger car. In this model, the impacted door is modeled in terms of an inner and outer door panel, where each panel consists of 28 partitioned masses. These masses are represented by non-

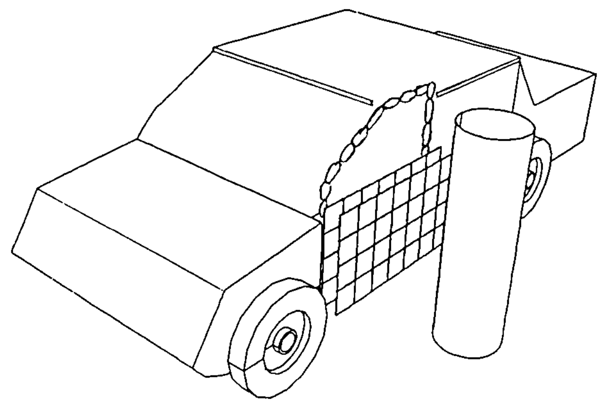

Figure 40. Multi-dimensional lumped mass model.

deformable ellipsoids in a matrix arrangement of 7 ellipsoids in the longitudinal direction and 4 ellipsoids in the vertical direction. Partitioned masses are also included as non-deforming ellipsoids in characterizing the response of the impacted side A- and B- pillars, roof rail and floor sill. All door ellipsoids are interconnected at their corners by non-linear energy absorbers through "hypothetical" beam elements and the masses and mass moments of inertia are lumped at the ellipsoid centers. This avoids interconnecting the various masses through a "analytically defined" network of beam-type elements, as in the shell framework method (36). Similarly, the pillar, roof rail and floor sill ellipsoids are joined through the "hypothetical" beam elements. The door is also interconnected to the pillars through the door hinges and lock. The pillars, roof rail and floor sill are interconnected to the main car by non-linear energy absorbers. Tire road interaction is also an important feature included in this model.

This lumped mass representation is currently being implemented on LS-DYNA3D based on a rigid body representation. The pole is modeled as a cylindrical structure. The SID driver in this model is the 3-D TNO-MADYMO SID which interacts through contact ellipsoids with the lumped mass multi-degree-of-freedom door. Since a lumped mass MADYMO-like representation of the SID is not available for the LS-DYNA3D program, the TNO-SID will be used through a special coupling arrangement of the MADYMO simulation model with the LS-DYNA3D FE model. In this model, the dummy to door contact forces determine the response of the door.

Once verified against test data, this model will be used to evaluate various side air bag designs including the BMW Inflatable Tubular Structure (ITS) for countermeasure purposes. It will also provide a tool for structural design changes for improvements in safety performance.

HYBRID MODELS

A useful mathematical model is a hybrid model which makes use of both the lumped mass and FE approaches for impact modeling. Such models should be designed with a certain degree of simplicity and yet show enough design detail for simulating as closely as possible the crash response for side impact parametric and countermeasure studies. To illustrate the usefulness of this approach for occupant safety simulations, the hypothetical impact model which simulates a pure MDB to door impact, shown in Figure 41, was assembled. Here, a simple door structure was considered with pillars rigidly attached to the floor instead of a real structural side design of a passenger car or light truck. Note that the structural responses of the near side door and pillars are modeled using the FE approach as well as the response of the struck occupant and its interaction with the door. The remaining structures, including the front and rear structures, in addition to the far side door and pillar structures, are modeled as a lumped mass rigid structure. These structures are modeled this way since, usually, they do not severely deform when the struck driver contacts the interior surface of the door in 90-degree side impacts. The SID was positioned 101.6 mm (4 inches) from the inner door surface. The "type 5" discrete nodes impacting surface (dnis) contact was used to model the contact between the SID's torso, hip and legs with the door. The mass of the rear seated occupant, SID, was included as part of the mass of this rigid body piece. The MDB is represented by its front face as a rigid piece with the 1360.78 kg (3000 lbs.) weight of the MDB evenly distributed over the entire front face piece. This model could be expanded to include a rear seated SID and adjacent contacting structures.

The attractiveness of this approach to modeling the safety of actual vehicles lies in its ability to develop efficiently a larger number of vehicle models, for use in first cut safety performance safety studies.

The simulated lateral velocity time histories from several locations on the struck door, compartment of the struck vehicle and the striking MDB for a 48.3 km/h (30 mph) impact are shown in Figure 42. SID rib and spine acceleration time histories are also displayed in Figure 43 and pelvic acceleration time histories are shown in Figure 44. All simulated time histories were 100 Hz filtered. In this example, a rigid rubber material was assumed for the SID's hip/leg assembly, since previous simulations at Quantum of a sled impact resulted in premature termination of the simulation run. Also, a non-deforming neck was assumed and the abdomen was not included in the model. However, the lower lumbar spine was modeled with the Mooney-Rivilin Rubber material model as defined by Quantum.

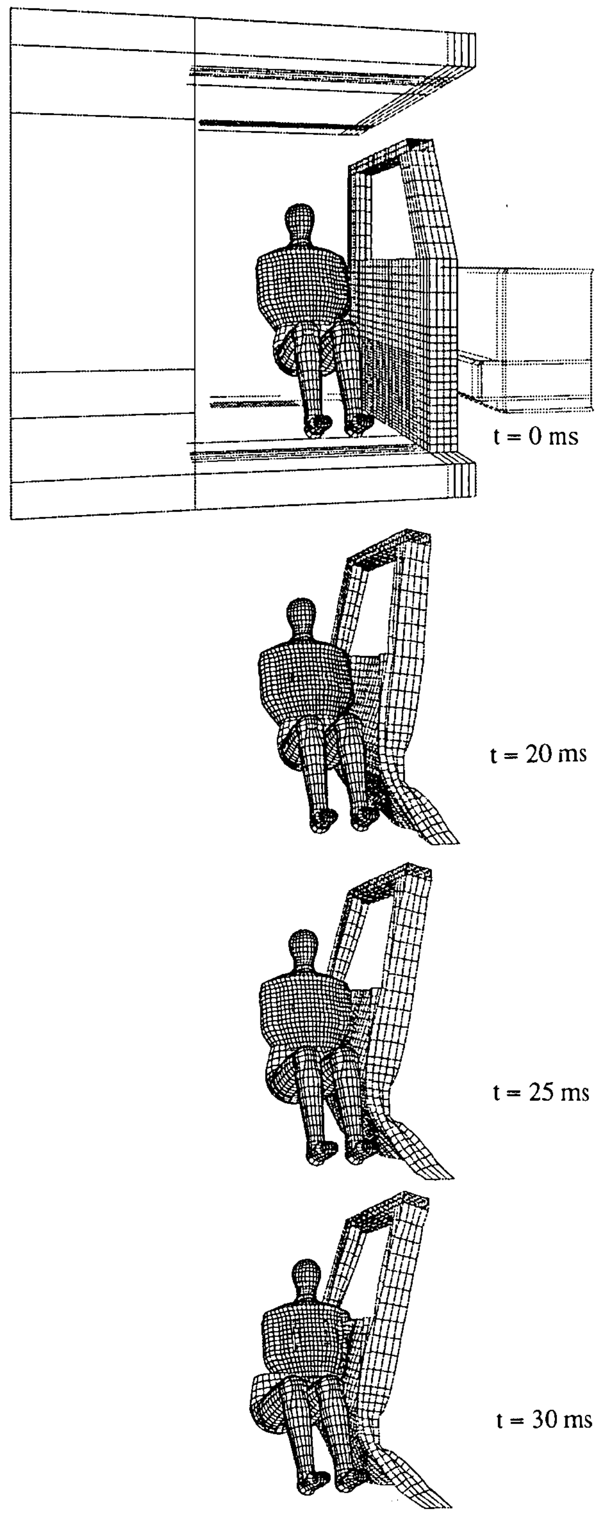

Figure 41. Simulated initial and deformed configurations from hybrid model.

As expected, the simulated velocities displayed in Figure 42 appear to follow the velocity trends observed in car crash tests. The hypothetical model's side structure is stiffer than a real car's side structure as noted in Figure 42,

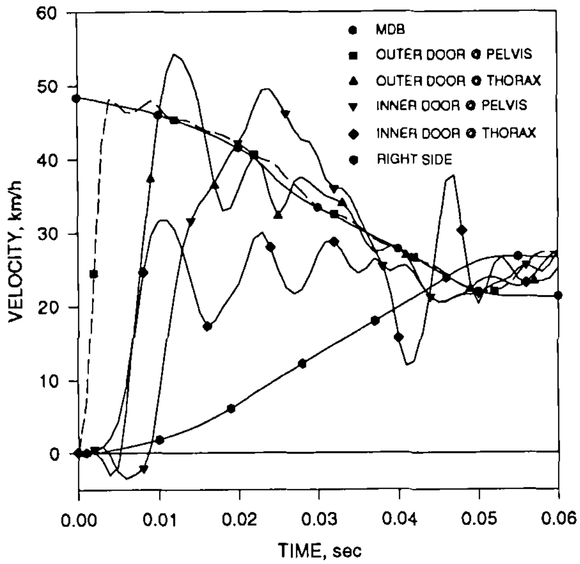

Figure 42. Simulated lateral velocity time histories from hybrid model.

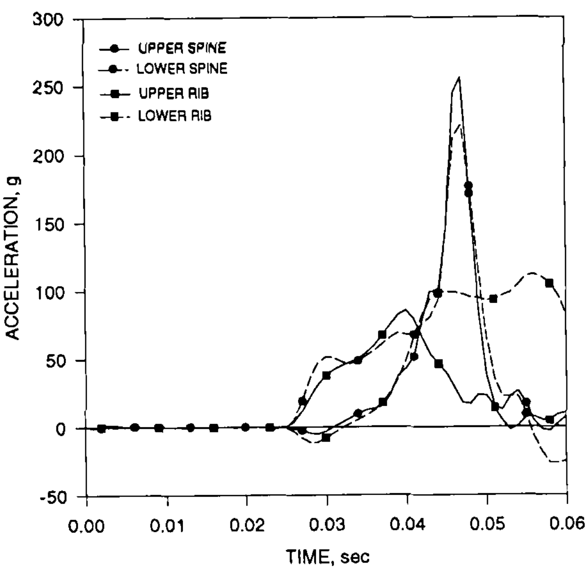

Figure 43. Rib and spine g's from hybrid model.

where the combined MDB-hypothetical vehicle velocity (mutual velocity) is reached at the end of 45 ms into the crash. Typically for passenger cars in 48.3 km/h (30 mph)/24.1 km/h (15 mph) side impacts, the mutual velocity is usually reached within 60 ms into the crash. Also, SID accelerations follow trends as expected for the case modeled here. The unreasonably high pelvic acceleration (four to five times higher than test) is directly attributed to the material assumption of a non-deforming hip/leg assembly. The use of a completely rigid MDB (impacting device) could indeed lead to higher SID responses in general. These same modeling assumptions also resulted in much higher g-levels in the lower and upper spine which peaked later in time. Unusual

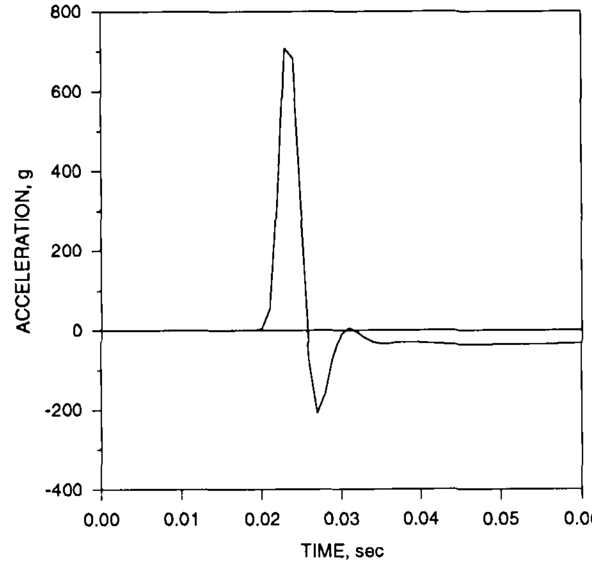

Figure 44. Pelvis g's from hybrid model.

excessive stretch in the lower lumbar spine was also observed in these simulations, indicating the need for further refinement of the SID's material codes.

SUMMARY

This paper presents an overview of NHTSA's application of advanced analytical tools in side impact research. The goal is to develop analytical models to evaluate occupant interaction with vehicle interior surfaces and assess potential occupant protection countermeasures in side crashes. Both FE and lumped mass modeling approaches, and a hybrid approach combining the two, are presented. The status of FE model developments is discussed. An FE side impact crash model of a passenger car is exercised in different impact configurations. Application of FE analysis to determine crash energy distribution under the different impacts is illustrated. Advanced side impact rigid body passenger car models, including dummy, which are currently under development are also presented.

Finite Element Model Development

A side crash FE model for a 1991 Ford Taurus was recently developed by EASi Engineering for NHTSA. This model is based on an earlier frontal model in which the side structure is refined with detailed definition for the doors, pillars, seat, and side floor pan. The structural response of the model was validated against data corridors at several locations within the vehicle in an FMVSS 214 dynamic impact configuration. The model was also analyzed in a break way luminaire support side impact. Further developments of this model are planned. These will include incorporating the dummy and interior components such as the seat, door panel, side header trim, and dash panel. The corresponding materials will be also characterized. A hybrid FE and rigid body SID will be developed and incorporated in the vehicle model. Corresponding occupant responses will be verified in an FMVSS 214 impact configuration. The resulting model will also be verified in an angled FMVSS 214 and a vehicle to pole side impact configuration.

The SID FE model was further developed by Battelle for NHTSA by primarily including the abdominal insert and anti-sag device, and introducing joints in the hip/legs area. The attachments were corrected for skull, neck, and legs, and contact interfaces were redefined in the thoracic areas. Material testing was performed and more representative foam models were incorporated. Model versus test data verifications of the refined FE SID have shown that the model is fully verified for pendulum impacts in the thoracic area. However, further refinement to the material codes and mesh geometry are needed for full verification of the pelvic response in pendulum impacts and overall body responses in sled impacts.

An FE model of the MDB for FMVSS 214 dynamic side impact simulations was developed in-house. The model was successfully verified in both 90-degree and angled 60-degree MDB to rigid wall impacts. Refinements to the front face honeycomb material codes and honeycomb rear-face to backing plate attachment are planned. Data from a planned series of rigid wall impacts at different angles and speeds using an advanced measurement MDB will be used for further model verification.

Finite Element Model Simulations

FE simulations of a variety of side impact configurations involving the Ford Taurus vehicle are presented: the FMVSS 214 MDB into the side of the vehicle, a vehicle into the side of another vehicle, and a vehicle sideways into a rigid pole. Experimental response data from the MDB and rigid pole tests are compared to the computed responses from the FE model simulations. Good correlation of the structural responses against the corresponding test data is obtained.

As a precursory analysis, a simulation of an MDB to Taurus impact incorporating an FE model of the SID is also presented. This demonstrates the authors' ability to simulate a full car crash test utilizing FE procedures by fully integrating an FE dummy in an FE vehicle.

Finite Element Model Applications

Analyses of energy absorption by structural members were performed using the FE simulation outputs for the MDB, rigid pole, and vehicle to vehicle side impacts. The purpose was to determine major energy flow paths and crash energy distribution under the different impact configuration. Results revealed that the front and rear doors on the struck side of the vehicle absorbed most of the energy, followed by the floor sill area and pillars in the MDB intersection type impact. Structural parts including the dash, firewall, steering wheel and column, and roof did not absorb energy until later in the crash. About 10% of the total energy was absorbed by the striking MDB vehicle. The striking vehicle's frontal geometry does affect the energy absorption distribution. For example, the simulation showed the door of the struck vehicle to absorb even more energy when the MDB was replaced by the Ford Taurus as the striking vehicle. The impact force thus appeared to be mainly concentrated on the door and not distributed across the entire side as in the MDB impact. These models simulated different crash energy absorption patterns when the vehicle was thrust into a rigid pole. The major energy absorbers in the pole impact were the driver door, floor sill and roof areas. Besides providing guidelines for areas of improvement in vehicle crashworthiness design, such information lends insight into the construction of potential lumped mass models for structural sensitivity and countermeasure studies.

Lumped Mass Model Developments and Simulations

Several advanced lumped mass models using the rigid body dynamics modeling approach were presented. The Ford Taurus was also targeted for these modeling efforts. The models included the MDB to car and the car to rigid pole side impacts. A 3-D lumped mass SID is incorporated in both models.

The MDB to car model takes into account the local rotational degrees of freedom of the door and pillar side structural members and the gross vehicle rotation of the struck vehicle about its roll axis. The dynamics of the tire/suspension system and interaction with the ground was modeled as well. The car to pole models are in various stages of development. Once they have been developed and fully verified with test data, plans are to incorporate and evaluate various countermeasures including air bags for mitigating occupant responses. Similar efforts are planned for vehicle to vehicle simulation studies utilizing the MDB to passenger car lumped mass model.

Hybrid Models

A hybrid model which makes use of both the lumped mass and FE approaches for impact modeling was also presented. This model simulates a pure MDB to door impact representing a hypothetical impact configuration. The simulated velocity responses appeared to follow the velocity trends observed in car crash tests. The attractiveness of such a hybrid approach lies in the ability to develop efficiently a larger number of vehicle models, for use in first cut safety performance safety studies.

CONCLUSION AND FUTURE RESEARCH

Overall, advanced side impact modeling will serve to enhance NHTSA's assessment and understanding of various side impact crash events. Analytical studies based on the models and modeling approaches presented in this paper are planned and should provide the basis for detailed vehicle safety performance and countermeasure characterization studies. Near-term efforts will focus on incorporating various side air bag geometries for mitigating head, thorax and pelvis injuries.

The initial FE simulations of the Taurus passenger car with the SID in the FMVSS 214 impact configuration and in the rigid pole impact demonstrate the utility of FE structural analysis. Good structural responses from this model are anticipated under various impact conditions such as angular impacts, higher speeds, and different impact locations. NHTSA will continue to focus efforts on the development and verification of FE models of various striking and struck vehicles, restraint systems, and occupants under different test conditions. Once a library of such models is constructed, vehicle safety performance studies under various impact modes can be performed. Simulation of these more complex models is now made possible with the increasing availability of low-cost powerful computer hardware. In certain cases, the coupling of FE techniques with rigid body dynamic techniques is an attractive feature. In these simulations, only parts of the vehicle or occupant are modeled by FE techniques, thus eliminating the need to process large scale models. Such models are a plausible approach for aggressivity and compatibility studies.

ACKNOWLEDGMENTS

The authors wish to acknowledge Dr. William T. Hollowell, Chief DOT/NHTSA Safety Systems Engineering & Analysis Division, and Mr. George Neat, Chief DOT/Volpe Center's Vehicle Crashworthiness Division. Special thanks are also given to Ms. Elizabeth D. Swanson and Mr. John Allspaw of DOT/Volpe Center's Vehicle Crashworthiness Division for their many helpful suggestions.

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EVALUATION OF THE PROTECTIVE EFFECTS OF SIDE AIRBAG SYSTEMS USING A MATHEMATICAL BIOSID DUMMY

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ABSTRACT

A two dimensional lumped mass model of the BIOSID dummy was developed and validated by means of pendulum and sled tests at various impact speeds. The mathematical BIOSID dummy consisted of eleven body parts; head, neck, arm, shoulder rib, thoracic ribs, abdmoinal ribs, spine and pelvis.The model was used to evaluate the potential injury reducing benefits of padding or airbags in side impacts. The side impact simulations were carried out using the crash victim simulation software MADYMO2D. The airbags were initially filled with compressed air and had varying initial (over) pressures and ventilation areas. The protective system resulting in the lowest injury measures for the dummy was thus determined.

The lowest TTI was obtained with the airbag with 0 kPa initial over pressure and 1500 mm^2 ventilation area, while the lowest chest deflection and chest VC were obtained with an airbag with 40 kPa initial over pressure and 2000 mm^2 ventilation area. The risk of the head impacting the side window was significantly reduced with the addition of an airbag. It was found that the airbag should have a thickness of approximately 120 mm when fully deployed to adequately protect the occupant. The arm and shoulder of the occupant have to be engaged in the impact. The model of the BIOSID dummy was found to be a very valid and valuable tool for evaluating the protective effect of padding and airbags in the side door.

INTRODUCTION

In the literature several investigations have shown that car-to-car side impacts are more severe than frontal impacts. Although the overall number of accidents is lower the relative number of seriously injured occupants is higher (Håland et al., 1993). In side impacts, injuries are about twice as common to struck side occupants as to non-struck side occupants (Håland et al., 1993; Langwieder and Bäumler, 1994).

In order to reduce injuries in side impacts, modifications to the passenger vehicles have been the common solution. The injury reducing benefits of these changes are then evaluated in sled and crash tests. At present there are two side impact crash test procedures: the American (NHTSA, 1990) and the proposed European procedure (ECE, 1993). In the tests, the risk of sustaining injuries is evaluated with a human surrogate, a crash test dummy. At present there are three side impact dummies available; the US-SID, EUROSID-1 and BIOSID. The US-SID is used in the American side impact procedure and the EUROSID-1 is used in the proposed European procedure. In a comparative evaluation, the BIOSID was found to be the most biofidelic of the three (ISO, 1990).

Both the EUROSID-1 and the BIOSID are distinctly different from the US-SID in their design features. The US-SID has no articulating shoulder and the arm and shoulder mass is incorporated into the mass of the thorax. The thorax is composed of five ribs attached to a shock absorber assembly (DOT, 1990). The US-SID is primarily intended to measure rib, spine and pelvic accelerations.

The EUROSID-1 on the other hand has stub arms and the shoulder-arm pivot allows two different arm positions with respect to the thorax. The thorax has three identical rib modules, consisting of steel hoops attached to a fluided spring-damper system, along with a damper running in parallel with the piston-cylinder (EEVC, 1985; Lowne and Neilson, 1987; Janssen et al., 1989; Roberts, 1989). In addition to the rib and spine accelerations, EUROSID-1 is capable of measuring rib deflections, abdominal penetrations and the forces associated with prescribed penetration levels, and forces at the pubic symphysis and the iliac crest.

The thorax of the BIOSID consists of six ribs. The top rib represents the shoulder while the three ribs in the middle simulate the human rib cage. The two bottom ribs represent the abdomen. The damping material is attached to the inside surface of the rib to provide viscous damping and dissipation of energy. The stub arms are attached to the shoulder rib through a clevis joint to permit arm rotation (Beebe, 1990). In addition to the rib, spine, and pelvic accelerations, BIOSID is capable of measuring rib deflections, and forces at the ilium, lumbar, pubic symphysis, and sacrum areas.

The American side impact procedure uses an acceleration-based criterion, the Thoracic Trauma Index (TTI) (Eppinger et al., 1984). TTI is the average of the maximum spine acceleration and the near-side rib acceleration in g's. The American regulation has TTI $\leq$ 85 g and TTI $\leq$ 90 g as injury criteria levels for four, and two door cars respectively. In the proposed European procedure chest deflection and a deformation-based injury measure, the Viscous Criterion (VC), are used for injury evaluation (Viano, 1989). VC is the instantaneous product of chest deflection velocity and chest deflection during impact. The proposed European procedure has chest deflection $\leq$ 42 mm and VC $\leq$ 1 m/s as injury criteria levels.

In a car-to-car side impact, the occupant on the struck side is hit by the intruding door. The likelihood for the occupant to sustain injuries is dependent on the velocity time-history of the door, the occupant location relative the door, and the shape and compliance of the interior (Lau et al., 1991). Reinforcement of the car structure (Mellander et al., 1989) to reduce the door-to-occupant impact speed, and to avoid a collapse of the B-pillar (de Coo et al., 1991) are the first and necessary steps in improving the occupant's protection. The next step is to use some suitable bolstering to make the interior more compliant. Padding is one type of bolstering; an airbag is another. When padding is used on the inside of the door, the impact between the occupant and intruding door occurs earlier and is of longer duration than if no padding is used. Padding can reduce rib accelerations and acceleration based injury measures like TTI. However, prolonged contact between the occupant and the door may increase the energy transfer to the occupant, resulting in a possible increase in chest deflection and deformation-based injury measures such as VC. If the padding material used is softer than the human chest, the intruding structures with the padding pushes the occupant away from the structures. Chest compression is avoided until the bag has bottomed out and a net reduction in chest deflection is obtained. If the padding is stiffer than the human torso, the chest deflection (and VC) increases instead of decreases.

Mathematical models have previously been used to evaluate the role of padding materials in side impacts. A lumped mass model of the anteroposterior thoracic impact response of the human chest was presented in 1973 (Lobdell, 1973). This model has been modified and used in pendulum simulations (Viano, 1987a). It has been found that with very hard padding materials, above 6 kN crush force, the material is so hard that it acts as a rigid wall. That is, no crush of the energy-absorbing material occurs during impact because the contact force is below the crush force 6 kN. Extremely soft materials are crushed with virtually no energy absorption and the impact when the material is bottoming out mimics that of a rigid wall impact. It has also been found that for energy absorbing materials of 100 mm thickness, the best reduction in peak viscous response is obtained with 3-4 kN constant crush force material or with material of 60-80 kN/m characteristics (Viano, 1987a; Viano, 1987b). The Lobdell chest model has been modified by including an energy absorbing interface (Shokoohi and Breed, 1991). Shokoohi used the modified model in pendulum simulations and obtained considerable reductions in both TTI and chest VC with air damped padding.

A 3D model of the US-SID dummy has been used to evaluate the effect of padding on TTI, chest deflection, and chest VC (Deng, 1987). Pendulum (free flight) and intruding door (velocity pulse) simulations were carried out. In the pendulum simulations padding reduced TTI, chest V and VC without significant chest rib deflection change. In intruding door simulations, padding also reduced TTI and thorax V, but considerably increased rib deflection and chest VC (Deng, 1987). In Deng's study padding stiffnesses of 0 - 150 kN/m were used.

The aim of the present study is to develop and validate a MADYMO 2D model of the BIOSID dummy.The model is to be used to evaluate the potential injury reducing benefits of various airbags and padding materials mounted between the struck side occupant and the door. The injury criteria chest deflection, chest VC, and TTI are used to evaluate the injury risk.

METHOD

The main focus of this study was to evaluate the injury reducing benefits of airbags or padding material mounted to the inside of the door. This was done by mechanical sled tests and mathematical simulations. A mathematical BIOSID dummy was developed and validated. Subsequently the dummy model was used in sled simulations. The sled tests consisted of a rigid door impacting a BIOSID dummy at various impact velocities. The padding material or airbag was mounted on the inside of the rigid door.

The mechanical BIOSID dummy was chosen because of its proven biofidelity (ISO, 1990). The mathematical BIOSID model developed for this study consisted of 12 body parts: head, neck, shoulder rib, arm, three thoracic ribs, two abdominal ribs, spine, pelvis and the legs. Pelvic foam was added in the upper leg mass (Figure 1). The spine and ribs were connected by a number of springs and dampers. The software used was the crash victim simulation program MADYMO2D (TNO, 1994).

The mathematical BIOSID dummy was validated by means of mechanical pendulum tests as well as mechanical sled tests (Pipkorn, 1996). The pendulum impact tests were carried out at 6.7 and 4.5 m/s impact velocity.

In the thorax, pelvis and shoulder calibration simulations, all parameters were within the performance corridor specified by First Technologies Safety Systems for pendulm tests at 6.58-6.84 m/s pendulum impact speed (Table 1) (BIOSID, 1991). In the abdominal calibration simulations, all parameters but the abdominal rib acceleration fell within the performance ocrridors specified by First Technologies Safety Systems (Table 2) (BIOSID, 1991).

Table 1.
Results from the model thorax and pelvis pendulum impact test, at 6.7 m/s compared to the performance specifications for the mechanical BIOSID (Without arm) at two different impact velocities (BIOSID user's manual, 1991)

	Mech. BIOSID Specification (6.58-6.84 m/s)	**Math. BIOSID** Model (6.7 m/s)
Pendulum Force (Thorax)	5.2 - 6.3 kN	5.3 kN
Thoracic Rib Defl (middle rib)	50-70 mm	63 mm
Thoracic Rib Acc (middle rib)	1305-1756 m/s^2	1636 m/s^2
Lower Spine Acc	118-162 m/s^2	149 m/s^2
Upper Spine Acc	191-240 m/s^2	212 m/s^2

Table 2.
Results from the model of abdominal pendulum impact test, at 4.5 m/s, compared to the performance specifications for the mechanical BIOSID (Without arm) at two different impact velocities (BIOSID user's manual, 1991)

	Mech. BIOSID Specification (4.41-4.59 m/s)	**Math. BIOSID** Model (4.5 m/s)
Pendulum Force Abdomen	2.9-3.5 kN	3.2 kN
Abdomen Rib Defl (lower rib)	38-51 mm	50 mm
Abdomen Rib Acc (lower rib)	530-854 m/s^2	926 m/s^2
Lower Spine Acc	78-105 m/s^2	94 m/s^2
Upper Spine Acc	54-80 m/s^2	69 m/s^2
Pendulum Force Shoulder	3.6-4.5 kN	3.8 kN
Shoulder Defl	21.2-29.1 mm	29 mm

In MADYMO2D there is an airbag module for modelling the interaction of airbag and penetrating objects. A model of the side airbag was developed using this airbag module. The side airbag was validated against the mechanical drop tests. The airbag force vs. deflection, and pressure vs. deflection from the mechanical drop tests were compared to corresponding results from the airbag model.

Sled tests

The mechanical sled tests simulated full-scale car-to-car side impacts (Håland and Pipkorn, 1991). In the mechanical tests, a reinforced door was mounted on a crash track sled (Figure 2). The dummy sat at a right angle to the crash track. The door approached the dummy at a constant speed, and after impacting the dummy, the door was braked at a constant deceleration (Figure 3). The door velocity time history simulated full scale conditions. The tests were run at two impact velocities: 12 m/s and 10 m/s. The higher velocity corresponded to a 50 km/h car-to-car side impact into a car that was not significantly reinforced. The lower velocity applied to conditions for a significantly reinforced car (Håland and Pipkorn, 1993).

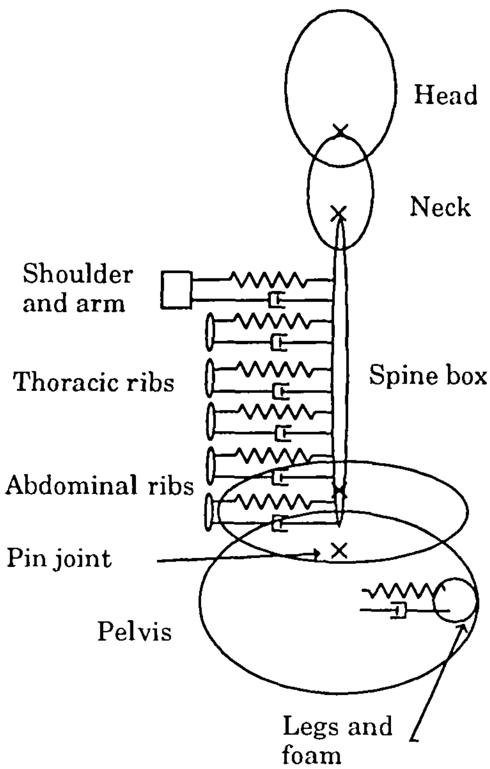

Figure 1. MADYMO model of the BIOSID dummy.

In the mechanical tests, 50 mm polyethylene foam or pre-inflated airbags were mounted to the door at chest and abdomen level. At the pelvic level, 75 mm thick polyethylene foam was mounted. It had previously been ascertained that automobile manufacturers considered padding materials of 50 mm for the chest area and 75 mm for the pelvis area as the limit for use in the vehicles (Håland and Pipkorn, 1991). The pre-inflated airbags had a volume of 12 l. In the present study the initial over pressure and the ventilation areas were varied

The padding materials or the airbag used extended over the chest and abdomen down to the level of the arm-rest. The three thoracic ribs and the two abdominal ribs were covered by the bag or padding. When inflated, the 12 liter airbag occupied the complete space between the door and occupant. Therefore increasing the bag size would not have improved the bag performance. The force deflection characteristics of the airbag was controlled by the initial pressure and the ventilation area. The airbag pressures and ventilation areas used in this study were considered to give the airbag force deflection characteristics of 60-100 kN/m (for an impactor area of 175 cm^2). In mathematical side impact simulations it was found that padding materials with this stiffness produced minimum chest VC (Viano, 1987b).

The mechanical tests and simulations were run at 12 and 10 m/s. The airbag had 0, 40 or 80 kPa initial over pressure and a ventilation area of 0 or 1500 mm^2. The 1500 mm^2 ventilation area was the largest possible ventilation area obtained with a fast opening valve used to achieve ventilation. The mechanical sled tests were carried out only once for each velocity and configuration. Earlier sled tests have ascertained that the repeatability of the results was acceptable (Håland and Pipkorn, 1991).

Simulations

The sled model including the BIOSID dummy was validated against the mechanical tests. In addition, simulations were run with only the rigid door. The validation and simulation matrix can be seen in table 3.

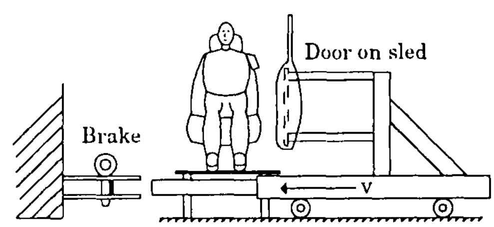

Figure 2. Mechanical sled test set up.

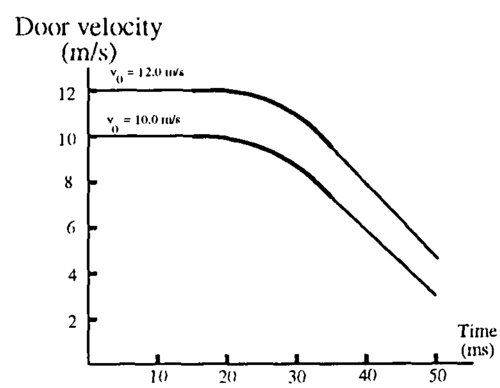

Figure 3. Door velocity time histories used in the sled tests.

Table 3.
Validation and simulation matrix

Door vel (m/s)	Config-uration	Initial bag pressure (kPa)	Vent area (mm^2)	Mechanical sled test
12	Rigid door	-	-	-
12	50 mm padding	-	-	x
12	12 l airbag	40	0	x
12	12 l airbag	80	1500	x
12	12 l airbag	40	1500	x
12	12 l airbag	0	1500	x
10	Rigid door	-	-	-
10	50 mm padding	-	-	x
10	12 l airbag	40	1500	x
10	12 l airbag	80	1500	x
10	12 l airbag	40	1500	x
10	12 l airbag	0	1500	x

Additional simulations were carried out in order to establish the airbag stiffness that produced the lowest TTI, chest deflection and chest VC. In these simulations the airbag stiffness was linear, up to 150 mm of bag compression. At that point the stiffness increased significantly, simulating the bottoming out effect (Figure 4). The force deformation characteristics that resulted in the lowest injury response is here referred to as the optimum airbag.

Simulations were also carried out at various impact velocities to investigate the effect of impact velocity on chest deflection and chest VC. The impact velocities used were 12, 10, 8, and 6 m/s. The simulations were carried out with a rigid door, 50 mm padding, and the force deformation characteristics of the bag that resulted in the lowest chest deflection and chest VC in the 10 m/s sled simulation.

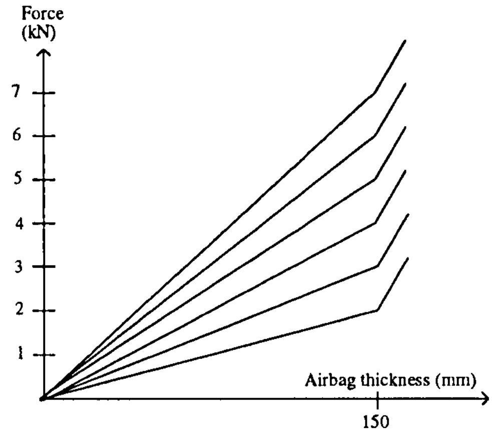

Figure 4. Airbag stiffness used in the simulations to find optimum occupant protection.

In the sled simulations, the force deformation characteristics of the airbag was used. The characteristics were obtained in drop tests using an impactor. The shape of the impactor was the same as the chest of the BIOSID dummy. The tests were carried out at 12 and 10 m/s impact velocities. The mass of the impactor was adjusted, so that the time from initial impact until bottoming out of the bag was the same in the drop tests as in the mechanical sled tests. Only one drop test was carried out at each impact velocity, bag pressure, and ventilation area (Figures 5 and 6).

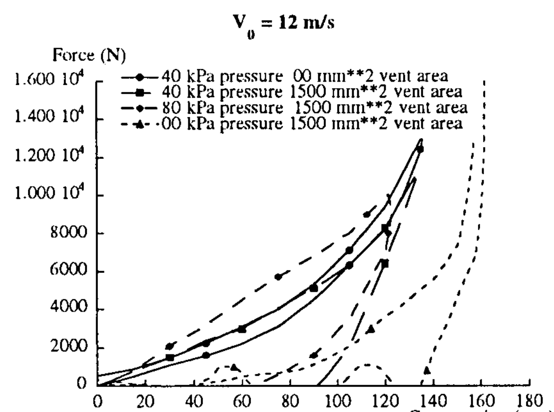

Figure 5. Force deformation characteristics of the airbag in the drop test at 12 m/s.

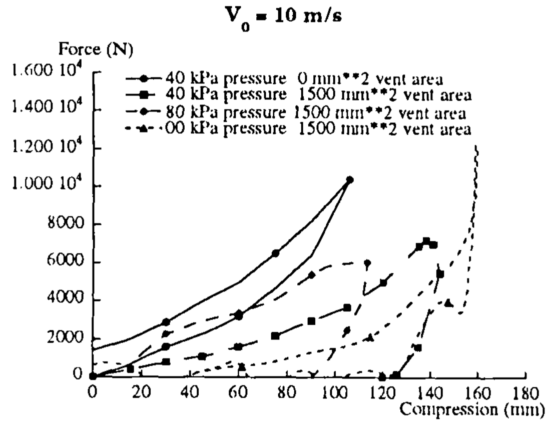

Figure 6. Force deformation characteristics of the airbag in the drop tests at 10 m/s.

The TTI, chest deflection, and chest VC measured in the sled tests were compared to the corresponding results from the simulations. TTI was computed by taking the average of peak spine acceleration and peak rib acceleration. Chest VC was computed by multiplying chest deflection by chest deflection velocity and dividing the result by 175 mm (half the thorax width). The average TTI of the three thoracic ribs of the BIOSID model was compared to the average TTI of the upper and lower thoracic ribs from the mechanical BIOSID (Figures 7 and 8). The maximum and minimum TTI shown in figures 7 and 8 were the highest and lowest TTI in the sled tests. The average chest deflection and chest VC of the three thoracic ribs of the BIOSID model were compared to the average chest deflection and chest VC of the three thoracic ribs for the mechanical BIOSID (Figures 9-12). The maximum and minimum chest deflection and chest VC shown in figures 9-12 were the highest and lowest chest deflection and chest VC in the sled test.

RESULTS

As can be seen from figures 7-12, generally good agreement between the results from the sled tests and the predictions from the model was obtained (Figures 7-12). In all simulations but one, the TTI of the model was somewhat higher than that obtained in the mechanical sled tests. In the simulation with 12 m/s initial velocity and 50 mm padding, the TTI in the mechanical sled test was somewhat higher than in the simulations (Figures 7 and 8). In both the 12 m/s and 10 m/s configurations, TTI was significantly reduced with the addition of padding or airbags.

In the simulations with 12 m/s sled velocity, the lowest TTI was produced by the airbag with 0 kPa initial over pressure and 1500 mm^2 ventilation area (Figure 7). The lowest chest deflection and chest VC were obtained with an airbag of force deformation characteristics 47 kN/m.

In the 10 m/s sled tests, the lowest TTI was produced by the airbag with 0 kPa initial over pressure and 1500 mm^2 ventilation area (Figure 8). The lowest chest deflection and chest VC were obtained with an airbag of 40 kN/m deformation characteristics.

In the 12 m/s sled simulations, chest deflection increased when 50 mm padding was added to the rigid door. The largest chest deflection in the 12 m/s simulations was obtained by means of the air bag with 40 kPa initial pressure and no ventilation (Figure 9). In the 10 m/s simulations, the chest deflection did not vary when 50 mm padding was added to the rigid door (Figures 9-12). The largest chest deflection was also obtained by means of the 40 kPa airbag with 0 mm^2 ventilation (Figure 10).

In the 12 m/s and 10 m/s simulations, the chest VC decreased when padding was added to the rigid door (Figure 11). When the airbag with 80 kPa initial pressure and a ventilation area of 1500 mm^2 was used, the chest deflection and chest VC predicted by the model was larger than those measured in the sled test. This was also the case for the 40 kPa airbag with 1500 mm^2 ventilation area. For the 10 m/s velocity there was good agreement between the results from the simulations and the impact sled tests for all configurations.

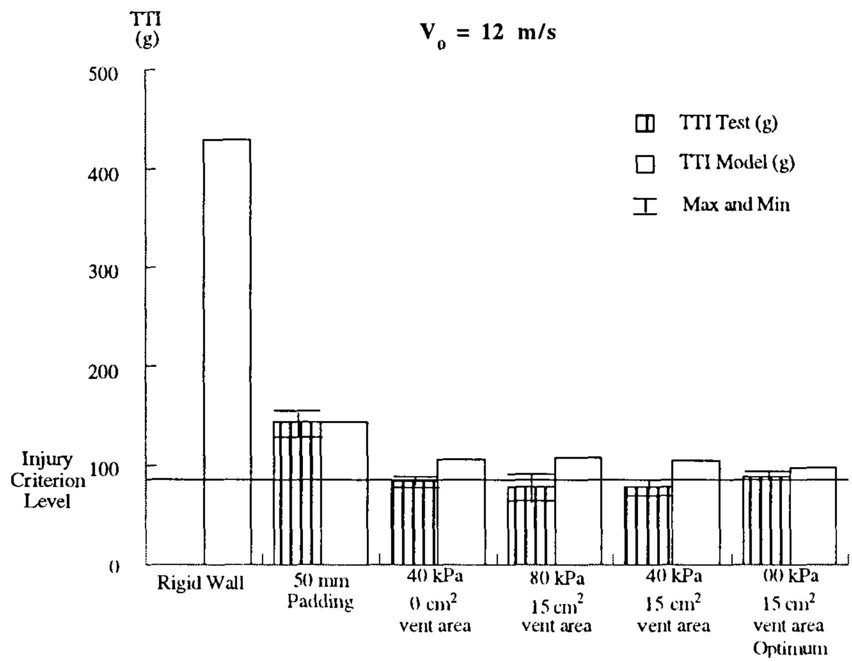

Figure 7. TTI for mathematical and mechanical BIOSID at an initial door velocity of 12 m/s (Mean values for the upper and lower thoracic rib).

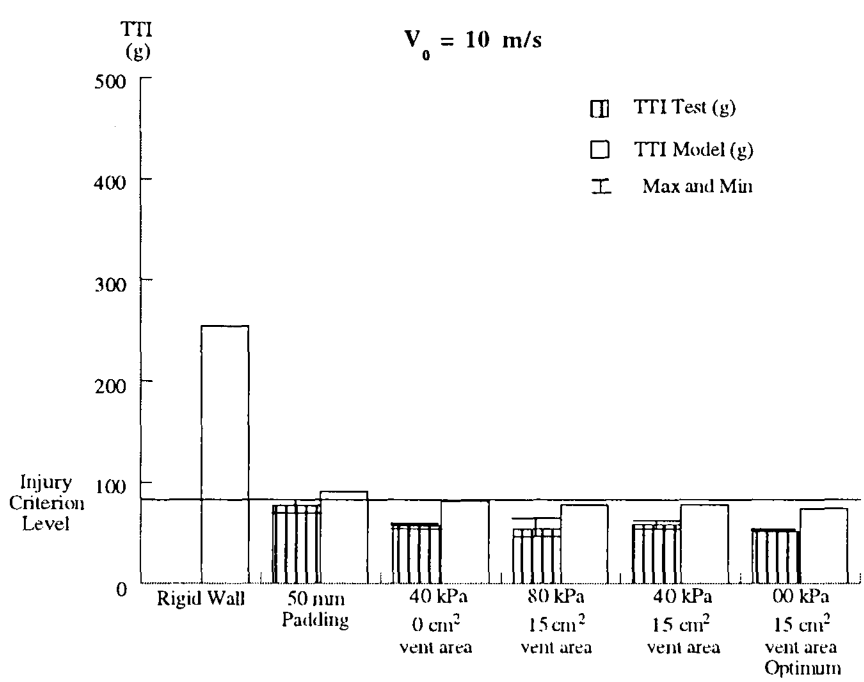

Figure 8. TTI for mathematical and mechanical BIOSID at an initial door velocity of 10 m/s (Mean value for the upper and lower thoracic rib).

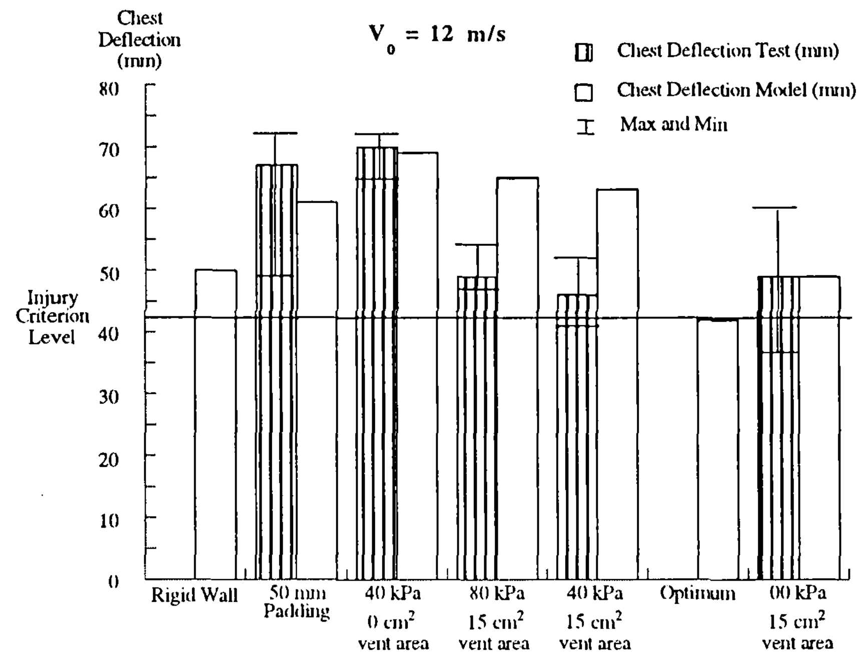

Figure 9. Chest deflection for mathematical and mechanical BIOSID at an initial door velocity of 12 m/s (Mean value for the three thoracic ribs).

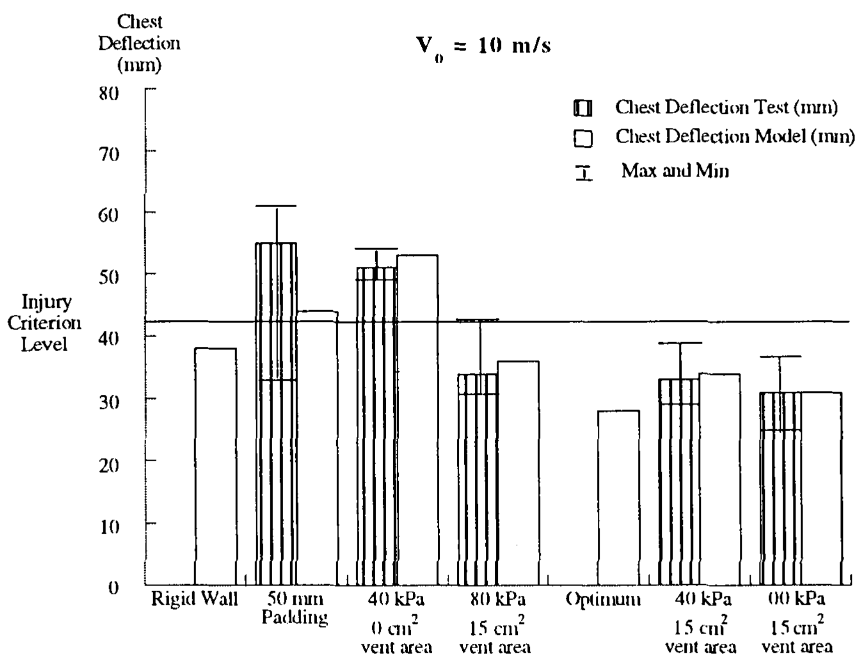

Figure 10. Chest deflection for mathematical and mechanical BIOSID at an initial door velocity of 10 m/s (Mean value for the three thoracic ribs).

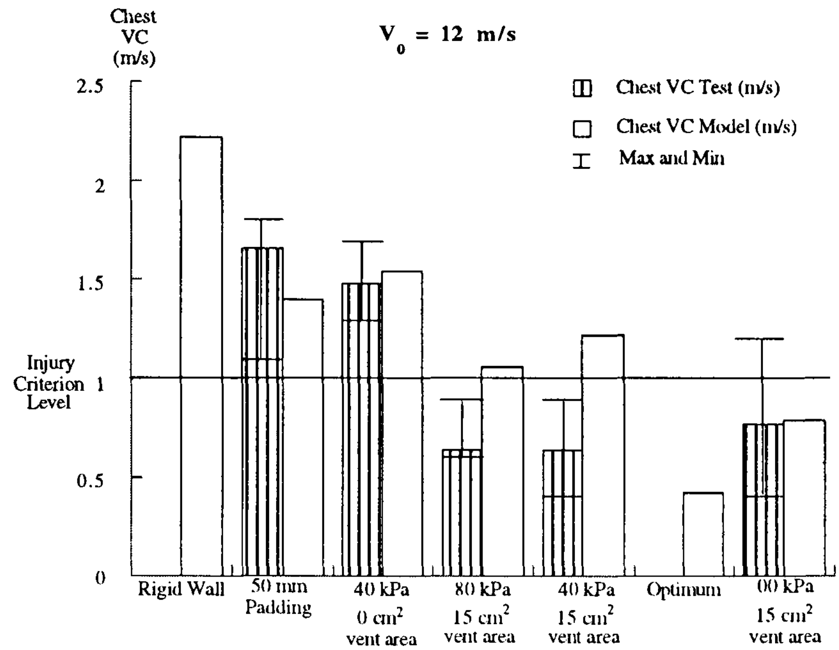

Figure 11. Chest VC for mathematical and mechanical BIOSID at an initial door velocity of 12 m/s (Mean value for the three thoracic ribs)

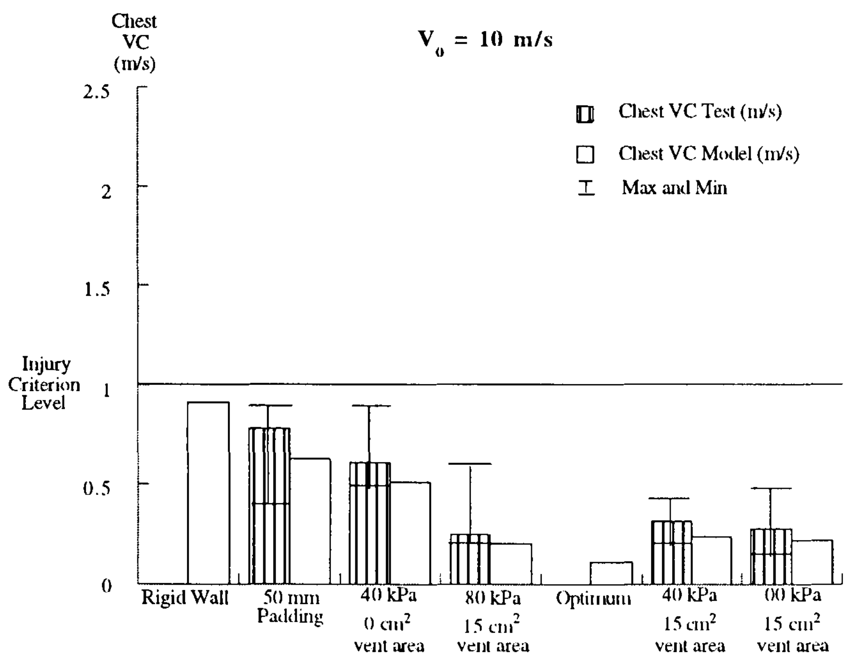

Figure 12. Chest VC for mathematical and mechanical BIOSID at an initial door velocity of 10 m/s (Mean value for the three thoracic ribs).

At 6 m/s door impact velocity, the lowest chest deflection was produced by the rigid door (Figure 13). At 12, 10 and 8 m/s impact velocities, the lowest chest deflection was obtained with the optimum airbag, while the largest chest deflection for all impact velocities was obtained with 50 mm padding. At all impact velocities only the optimum airbag produced chest deflection values that were lower than the proposed injury criteria level of 42 mm. For the rigid door, and for the door with 50 mm padding, chest deflection showed an approximate linear increase with door velocity (Figure 13).

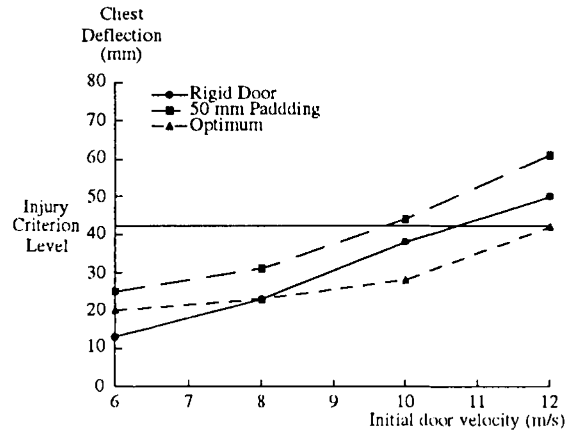

Figure 13. Chest deflection at door impact velocities of 6, 8, 10 and 12 m/s.

The lowest chest VC for all impact velocities was produced by the optimum airbag (Figure 14). The largest chest VC values for all impact velocities were obtained with the rigid door, while the 50 mm padding produced chest VC values ranging between those obtained by the optimum airbag and by the rigid door. The optimum airbag produced lower chest VC values at all impact velocities than the injury criteria level of 1 m/s. Chest VC increased exponentially with impact velocity for the rigid door, the door with 50 mm padding and the door with the optimum airbag.

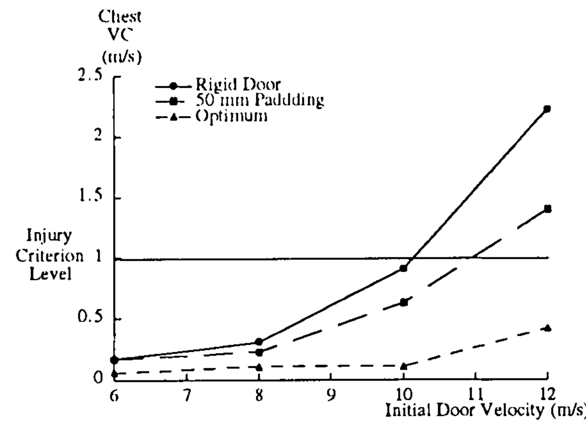

Figure 14. Chest VC at door impact velocities of 6, 8, 10 and 12 m/s.

DISCUSSION

This study has shown that, in a car-to-car side impact, optimum protection of the occupant (lowest chest deflection, chest VC and significantly reduced TTI) is obtained with an airbag with force deformation characteristics of 47 kN/m and 40 kN/m for the 12 m/s and 10 m/s impact velocities respectively. By using the MADYMO drop test model, the 47 kN/m and 40 kN/m stiffnesses were estimated to correspond to an airbag with an initial over-pressure of approximately 40 kPa and a ventilation area of 2000 mm^2. The maximum bag over-pressure obtained in the drop test simulations was 78 kPa for the 12 m/s impactor velocity. For the 10 m/s impactor velocity, maximum pressure was 50 kPa. Biomechanical data from Wayne State University (King et al., 1991; Cavanaugh et al., 1992) reported that crush strength padding of 20 psi (140 kPa) was tolerated by the thorax of people aged 20-40 and that 8-11 psi (55-80 kPa) was tolerated by the thorax of people aged 50-60. These estimates regarding suitable stiffness of a side airbag were found to correspond to the research data for elderly people.

With 50 mm padding added to the stiff door, TTI was reduced significantly, chest deflection was increased and chest VC was reduced somewhat. It was observed that the rib acceleration time curve had two peaks. The first peak, the largest, occurred during initial contact with the door. The first rib acceleration peak was significantly reduced by the addition of padding while the spine acceleration was not altered. TTI, which is calculated as the average of the impacted rib acceleration and the spine acceleration, was therefore reduced effectively by making the initial door (padding) to rib contact softer. Adding an airbag, or thicker padding than the 50 mm, did not significantly further reduce the initial door to rib contact force and TTI.

When padding or an airbag was placed on the inside of the door, the contact with the occupant took place earlier than if no padding was used. More energy was transferred to the occupant, and chest deflection increased. However, this padding somewhat reduced chest deflection velocity and chest VC. It was found in this study that to compensate for the increased energy transfer by the padding material or airbag, these have to be very soft. The optimum airbag was found to be thick and soft enough to efficiently reduce chest deflection and chest VC. However, 50 mm padding did not prove thick enough to efficiently reduce chest deflection and chest VC. To reduce chest VC it is important that peak chest deflection velocity and peak chest deflection do not occur simultaneously. It was also observed that in the simulations the largest reductions in chest VC were obtained when peak chest deflection velocity occurred prior to peak chest deflection.

A compliant airbag that is softer than the chest pushes the ribs and spine (chest) away from the intruding door without compressing the chest. Thus, the complete distance between the occupant and the door can be used by the airbag to push the chest away from the intruding door. If the airbag is stiffer than the chest, the ribs are pushed closer to the spine and the chest is thus compressed. Chest deflection and chest VC will then increase. If the airbag is too soft, the bag bottoms out without pushing the ribs and spine away, and the ribs impact the stiff intruding structures. Chest deflection will then decrease, but conversely chest deflection velocity and chest VC will increase.

A two-dimensional approach was chosen in this study because it was observed in the sled tests that the behavior of the mechanical dummy was in principle two-dimensional. An added reason for using MADYMO2D was to gain access to the simple MADYMO2D airbag module. The airbag module has been proven to work for driver side airbags but no evaluation of the airbag has been made as far as side impacts (Nieboer et al., 1988). In addition to the simple 2D airbag module, MADYMO has a 3D finite element airbag module. However, it is more complex and time consuming to use the 3D airbag than the 2D airbag. However, the airbag model was not used in the sled simulations because of the important assumption inherent in the MADYMO airbag module that penetration does not cause the bag shape to deform. It was observed in the mechanical sled tests that the deformation of the airbag was important for the performance of the bag. The bag deformed and distributed the load on all five ribs of the mechanical dummy. For this reason, the airbag module was not used in the sled simulations. In the mechanical drop tests, on the other hand, the deformation of the bag due to penetration was of minor importance because the chest impactor consisted of only one body representing the five ribs. A model of these mechanical drop tests was therefore developed using the MADYMO airbag module. The drop test model was validated against the mechanical drop tests. Generally good agreement was obtained between the results from the drop tests and the predictions from the model. The drop model was then used to estimate the airbag pressure and ventilation area that corresponded to the optimum airbag. It was also found, using the MADYMO drop test model, that the initial slope of the force compression curve was dependent on the pressure of the airbag, and that the latter part of the force compression curve was dependent on the ventilation area. In the drop tests, it could also be observed that the force deflection characteristics of the airbag were dependent on impact velocity (Figures 5 and 6).

The mechanical sled tests in combination with the mathematical sled model used in the present study were more economical and less time consuming than carrying out crash tests. The mathematical sled model proved an effective tool since a great number of parameters were evaluated in a very short period of time and at very little expense. In addition the mathematical model did not suffer from the repeatability problems often encountered in mechanical tests. All the important parameters that were included in the mechanical sled method were also included in the mathematical model. All important body regions, such as the shoulder, thorax, pelvis and legs were impacted since the motion of these body regions affect the thorax response (Deng, 1989). The important injury causing parameters, the door inner velocity at the time of contact with the occupant and the door velocity time history during contact with the occupant, were also included in the method (Nilsson, 1985; Deng, 1989; Watanabe and Yamaguchi, 1989).

The mechanical sled test method consisted of a rigid door mounted on a sled. In a passenger vehicle the door is not rigid and can be deformed by the occupant, therefore in the mechanical sled tests the door was covered with 20 mm stiff padding to compensate for this rigidity. The advantage of this approach is that the door can be used in repeated tests which results in good repeatability of the results.

It was observed that the results from the mechanical sled tests were in good agreements with the predictions from the simulations despite the fact that only one mechanical sled test was conducted for each configuration. However, the TTI values from the BIOSID in the sled model were higher than the corresponding measurements in the sled tests. The pendulum calibration test procedure for the BIOSID specifies a performance corridor for peak rib and spine acceleration and peak rib deflection (BIOSID, 1991). In pendulum calibration simulations previously carried out, the peak rib acceleration of the BIOSID model was close to the upper limit of the performance corridor (Pipkorn, 1996). The peak rib acceleration of the mechanical BIOSID, however, was found to be not as close to the upper limit. The peak spine acceleration was in the middle of the corridor for both the BIOSID model and the mechanical BIOSID. Rib and spine accelerations were used to compute TTI. Therefore TTI from the model was expected to be higher than the TTI measured in the mechanical tests. Note that the chest deflection predicted by the model in the pendulum calibration simulations was in the middle of the performance corridor.

The chest deflection and chest VC values from the sled tests were significantly lower than these predicted by the BIOSID model in two tests (Figures 9 and 11). In the 12 m/s sled test using the 40 kPa airbag with a ventilation area of 1500 mm^2, the airbag pressure dropped to approximately 30 kPa before the bag was impacted by the occupant. Therefore the airbag in this sled test must have been softer than in the corresponding simulation. In the 12 m/s sled simulation using the 80 kPa airbag with a ventilation area of 1500 mm^2, the force deformation characteristics used were too hard (Figures 9 and 11). These characteristics were obtained from the drop tests. There was uncertainty in these characteristics since only one drop test was performed at each impact velocity, bag pressure and ventilation area.

A few side airbag prototypes have been proposed in the literature. In a crash test with the Hybrid III dummy using a 60 liter side airbag prototype, modest head and chest accelerations were reported (Warner et al., 1989). In another side airbag system, TTI was lowered by 10% with a 40 liter airbag deployed from the armrest to the roof rail (Kiuchi et al., 1991). In a system with a 14 liter side airbag, TTI was also reduced by 10% (Schlopp, 1993). The relatively small reductions in TTI can be due to these bags being too stiff. In sled tests using a small airbag system similar to the one used in this study, Olsson et al., (1989) obtained significant reductions in TTI. The dummy used was the US-SID and the impact velocity was 9 m/s. In addition Olsson et al. (1989) observed that a small airbag system also reduces the risk of head ejection.

An evaluation of the injury reducing benefits of padding materials has been carried out by Zuby, (1991) who used 152 mm thick samples. The 12 l airbags used in the present study were of similar thickness as these padding materials. In sled tests at an impact velocity of 8.9 m/s, Zuby (1991) had found that Ethafoam LC 200 (103 kPa at 35% compression) produced the lowest TTI for the BIOSID. VC had been found to be reduced by 49% with ARCEL 310 (220 kPa at 35% compression). The stiffness of the ARCEL 310 was similar to the stiffness that was found to be the optimum airbag stiffness in the present study. However, it is not feasible to mount 150 mm thick padding materials in a passenger vehicle today. The airbag on the other hand is technically a feasible solution since it has the same effect as padding and is hidden in the door when not deployed.

In crash tests using the EUROSID-1, Hobbs, (1989) found that 150 mm padding did not reduce chest deflection. However, Chest VC was reduced by 23% with the addition of 150 mm padding. If intrusion was eliminated and padding was added, chest deflection was reduced by 51% and chest VC by 83%. Chest deflection and chest VC were thus significantly reduced when padding was added and intrusion eliminated. Eliminating intrusion reduces the door-to-dummy impact velocity. It was also found by Hobbs, (1989) that reductions in chest deflection and chest VC were obtained when the impact velocity was reduced.

How sensitive chest deflection and VC were to late deployment of the airbag was investigated by using the sled model. The thickness of the airbag was reduced to simulate late deployment. For the 12 m/s sled tests, a 20 mm reduction in airbag thickness corresponded to 1.6 ms later triggering time for the airbag. For the 10 m/s simulations, 20 mm reduction in airbag thickness corresponded to a 2 ms later triggering time of the airbag. For the 12 m/s simulations, chest deflection increased by 2 mm. Chest VC did not vary for the first 40 mm of airbag reduction, i.e. 4 ms later triggering time. Greater reductions than 40 mm resulted in an increase in chest deflection and a large increase in chest VC (Figure 15). These results indicate that the side airbag should have a thickness of approximately 120 mm when fully deployed to adequately protect the occupant.

For the 10 m/s simulations, chest deflection and chest VC did not change in terms of reduction in airbag thickness of less than 40 mm. Reductions in airbag thickness greater than 40 mm resulted in an increase in chest deflection and chest VC (Figure 16).

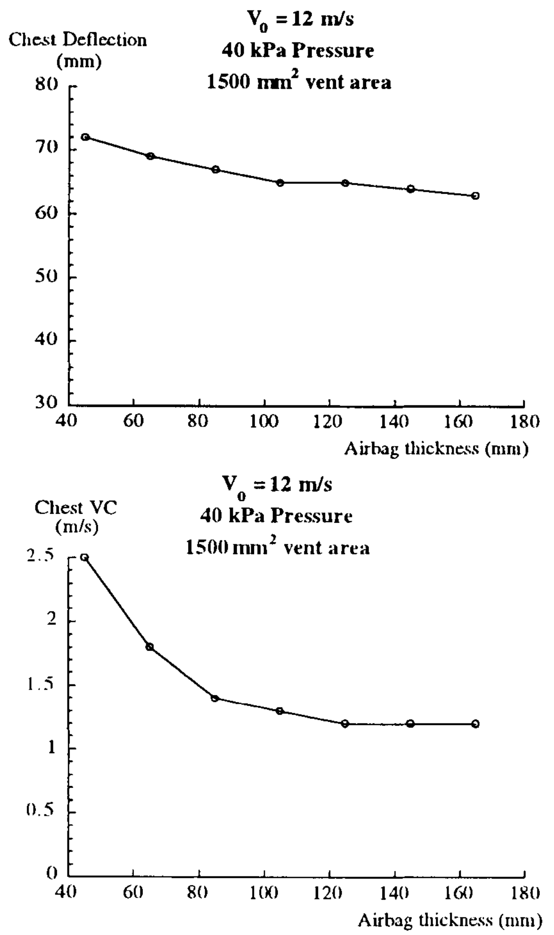

Figure 15. Chest deflection and chest VC for late deployment of the airbag at 12 m/s impact velocity.

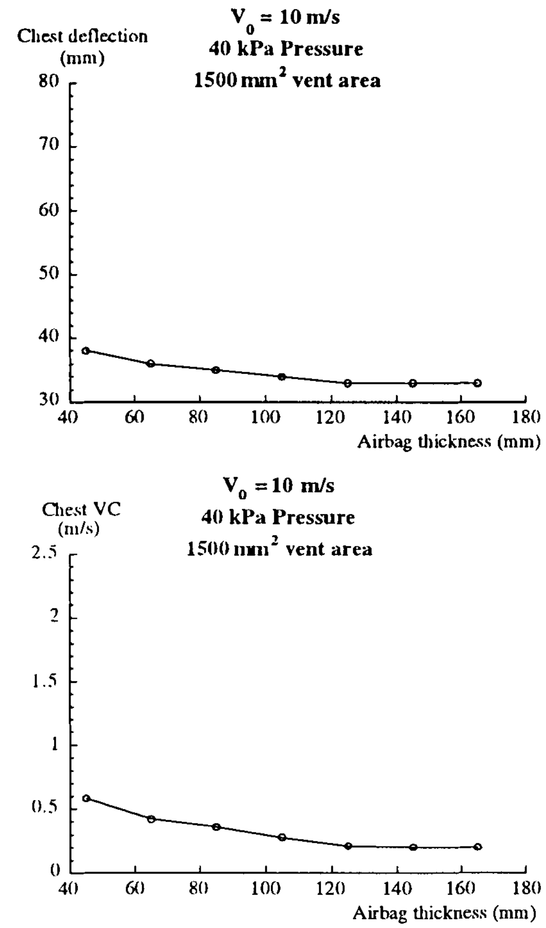

Figure 16. Chest deflection and chest VC for late deployment of the airbag at 10 m/s impact velocity.

Since the arm position of an occupant in a car-to-car side impact can vary considerably, the arm and shoulder of an occupant may not always be actively engaged in a side impact. In the mechanical sled tests, the arm of the BIOSID was positioned at a 45° angle to the chest. Hence, in the mechanical sled tests, the arm and shoulder of the dummy were engaged in the impact. The importance of engaging the arm and shoulder in the impact was investigated by doing simulations at 12 and 10 m/s sled velocities using the optimum airbag. It was found that both chest deflection and chest VC increased when the shoulder and arm were not engaged. In the 12 m/s sled simulations, chest deflection increased by 33 % and chest VC by 86 % when the shoulder and arm were not impacted by the intruding structures. In the 10 m/s sled simulations, chest deflection increased by 46% and chest VC by 219% when the arm and shoulder were not engaged in the impact. Therefore had the arm and shoulder of the dummy not been engaged in the mechanical sled tests, the injury measures would have been significantly greater.

This study has shown that, in a car-to-car side impact, optimum protection of the occupant (lowest chest deflection, chest VC and significantly reduced TTI) is obtained with an airbag with an initial pressure of 40 kPa and a ventilation area of 2000 mm^2. The thickness of the bag has to be at least 120 mm when fully inflated. In addition the arm and shoulder of the occupant have to be engaged by the intruding door.

The mathematical model of the BIOSID dummy has proven a cost effective and useful tool for quick evaluations of the injury reducing benefits of airbags or padding mounted in the side door. The BIOSID model is suitable to use for parametric studies. The model produces reliable results for both pendulum and sled simulations at various impact velocities, padding materials and airbag configurations.

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SIDE IMPACTS IN AUSTRALIA

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Paper Number 96-S6-O-14

ABSTRACT

Approximately thirty percent of fatal car crashes occurring in Australia each year involve side impacts. Recent international research has focused on the development of Standards for side impact testing.

This paper presents preliminary data obtained from an indepth vehicle factors crash study in respect to side impact crashes. Data was collected through on-scene investigations and vehicle inspections. Information is presented in terms of distribution of damage, striking vehicle and injury severity.

BACKGROUND

Several reports have presented information on impact distribution, striking object, and injury severity. In 1995, Ginpil et al reported on crashes in Australia resulting in car occupant fatalities in side impacts. The study found that 34% of all cars with an occupant fatality sustained a side impact. In the United States it has also been reported that 34% of all car occupant fatalities were side impacts (IIHS). Lestina reported that 29% of fatal crashes in the United are side impacts.

Ginpil also conducted analysis on area of impact from the Federal Office of Road Safety's Fatality File. Ginpil found that over 70 % of side impacts were to the centre side portion of the vehicle.

In 1992 Huelke et al reported on the damage distribution of side impacts recorded in the United States National Accident Sampling Scheme (NASS) between 1980-88. Huelke used the Collision Deformation Classification (CDC) for the coding of damage (SAE 1980). He found that left and right impacts involving the passenger compartment account for approximately 66% of side crashes.

Previous studies reviewing the striking object in side impact crashes are summarised in Table 1.

Table 1
Review of Striking Objects

	Ginpil (fatalities)	Lestina (all side impacts)	Mackay (all side impacts)
Car	30%	55%	59%
Heavy Vehicle	18%	14%	22%
Fixed Objects	40%	19%	16.7%

METHODOLOGY

This paper is based on an analysis of 234 crashes. The crashes investigated are part of a larger on-going indepth crash study into vehicle factors.

The Crashed Vehicle Study (CVS) will examine 5000 vehicles (approximately 3000 crashes) that have been involved in crashes that occurred throughout the State of New South Wales (NSW) in Australia (Duignan et al). The investigations are conducted by trained vehicle inspectors who randomly attend crashes. The crashes investigated, involve both Urban and Rural crashes, and include all vehicle types. The inspections are conducted both on-scene with further inspections conducted at repair yards. The primary aim of the study is to investigate the role of defects in crash causation and severity, but inspectors also collect comprehensive information and photographs on vehicle damage and crash scenes. Coding of the vehicle damage is by the Collision Deformation Classification. (SAE 1980).

Information on each crash and vehicle is coded and entered into database software. Information coded relevant to this analysis includes:

- Crash location,
- Type of Crash (RUM Code),
- Intersection type,
- Vehicle number and type involved,
- Injury severity of crash, and
- Damage codes of vehicles.

The crashes extracted for this paper have at least one vehicle with either left or right side damage. This equates to an "L" or "R" in column 3 of the CDC in either the primary or secondary impact codes.

Fatal crashes were individually reviewed to determine the vehicle in which the fatality occurred crash dynamics and possible countermeasures.

RESULTS

Injuries/Striking Vehicle

151 vehicles in 128 crashes had side impact damage from a total sample of 371 vehicles and 234 crashes. This represented 40.7% of vehicles and 54.7% of the crashes.

15 of the 234 crashes studied (6.4%) had at least one fatality. 9 of the 15 fatal crashes had at least one vehicle with primary or secondary damage to the right or left side. This represented 3.8 % of the total sample, and 60% of the fatal crashes investigated. Tables 1 and 2 show the breakdown of injury severity and vehicle type for the side impact crashes involved in this analysis.

The 9 fatal crashes involved 28 vehicles. The general findings were:

- 10 vehicles had side impact damage as primary damage
- 7 of the 10 vehicles with side impact damage were passenger cars or derivatives. One vehicle was a utility.
- 5 of the 9 striking vehicles** had a gross vehicle mass in the range of 8 to 24+ tonnes
- 3 of the 9 striking vehicles** were a car, car derivatives or 4WD.
- only one vehicle, a rigid truck, received the damage from an impact with roadside furniture.
- The 9 fatal crashes involved 12 fatalities. There were 4 fatalities in one crash.
- The 8 remaining fatalities were drivers of the vehicles which had side impact damage

Annexure 1 details a sample of the fatal crashes.

29 of the 234 (12.3%) crashes had at least one serious injury. Of these 29 crashes, 11 (38%) were coded for primary or secondary damage on the left or right side.

Table 2
Number of Crashes by
Crash Severity

Fatality	9 (7%)
Serious Injury	11 (8.6%)
Minor Injury	31 (24.2%)
No Injury*	3(57.1%)
Unknown	4 (3.1%)
Total	128

Table 3
Number of Vehicles (with side impact damage)
by Crash Severity

	Car	4WD	Van	M/cy	Hev Veh
Fatality	8	-	1	-	1
Serious Injury	7	1	-	2	1
Minor Injury	22	2	2	1	10
No Injury*	61	1	6	-	21
Unknown	2	-	1	1	-
Total	100 66.2%	4 3.3%	10 6.6%	4 2.7%	33 21.8%

* No Injury includes minor injuries treated at scene.
** Vehicle causing the major impact

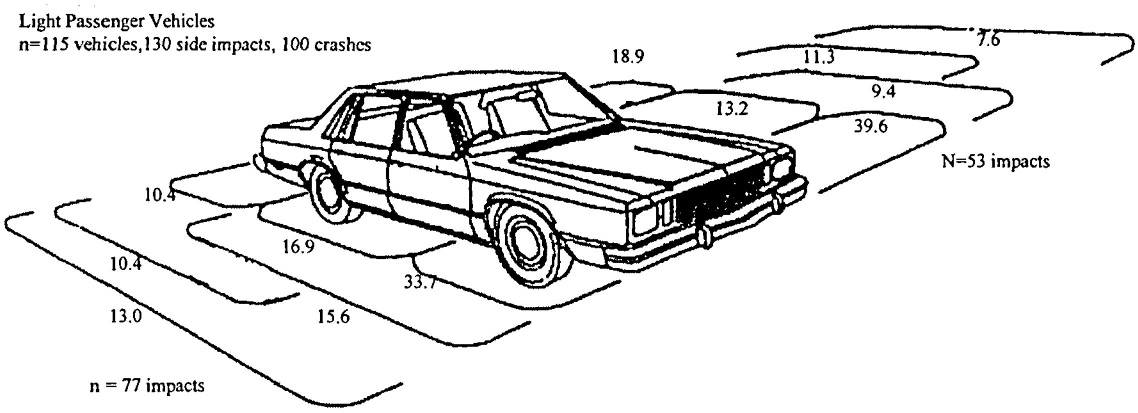

Figure 1: Distribution as a percentage of the left and right side impacts.

Damage Distribution

Of the 234 crashes in the study so far, there were 88 crashes in which 100 light vehicles received initial impacts to the left or right side. If secondary impact damage is included, 115 light vehicles in 100 crashes received side damage. There were a total of 130 impact sites on these vehicles. Analysis was conducted on the distribution of damage codes of the specific longitudinal areas as per the CDC coding column 4. There were 77 and 53 impacts to right and left sides respectively. 56% and 58% of the impacts to the right and left side respectively, involved a direct impact with the occupant cell. 34% and 40% of the impacts to the right and left side respectively were concentrated to the front quadrant of the vehicle. Figure 1 shows the distribution according to CDC coding.

Type of Crash

Analysis conducted on the type of crash. (Table 4) indicated that 44.5% of crashes occurred at intersections. 7 of the 9 fatal crashes occurred in locations other than an intersection.

Analysis on the Road User Movement (RUM) groups against crash severity, shows vehicles from adjacent directions (intersections only) and vehicles from opposing directions to be predominant.

Table 4
Injury Severity by
Intersection Type

	Cross	Tee	Y	Nil	Rlwy	Rndbt	U/K
Fatality	1	1	-	7	-	-	-
Serious Injury	4	-	-	7	-	-	-
Minor Injury	9	2	2	16	1	1	1
No Injury	15	22	-	31	-	2	2
Unknown	2	1	-	1	-	-	-
Total	31 24.2%	26 20.3%	2 1.6%	62 48.5%	1 0.8%	3 2.3%	3 2.3%

Table 5
Crash Severity by
Road User Movement
(Grouped)***

	10	20	30	40	50	60	70	80	90	99
Fatality	2	6	-	-	-	-	1	-	-	-
Serious Injury	4	2	1	1	-	-	1	2	-	-
Minor Injury	8	4	2	2	1	2	4	7	-	1
No Injury	21	15	12	3	2	2	7	8	1	2
Unknown	1	1	-	-	-	-	1	-	-	1
Total	36 26.2 %	28 21.9 %	2 11.7 %	6 4.7%	3 2.3%	4 3.1%	14 10.9 %	17 13.3 %	1 0.8%	4 3.1%

Discussion

The results presented in this paper are preliminary results obtained from the RTA's Crashed Vehicle Study. Although the number of crashes at this stage is relatively small (234), at the completion of the study there will be approximately 5000 vehicles (3000 crashes) to analyse. If the current trend continues, a database of over 1000 side impact crashes will ultimately be in the sample.

Conclusions should not be made from the data provided to date. Numbers are small and are not representative of the overall NSW statistics. Fatal crashes generally represent 1% of the total crashes in NSW each year. The Crashed Vehicle Study has 6 % at present. This is due to the under reporting of the "no injury" crashes. Heavy vehicle crashes are also over represented in comparison to Mass Crash Statistics. At the completion of the study a weighted analysis will be performed The study has been designed on a quota sampling system so that ultimately represented reliable analysis can be performed.

Not withstanding the above, there are some interesting points to note:

* 48.5 % of the total side impact crashes did not occur at an intersection. Ginpil et al found that 48% of fatal crashes did not occur at an intersection. If an analysis was being conducted on the RTA's Mass Crash Database, a majority of these non-intersection crashes would not be retrieved as side impact crashes because side impacts are assessed on pre-crash manoevours not crash dynamics.

* 50% of impacts were to the occupant cell of the vehicle. This compares to Huelke who found that for side impact crashes in the United States, approximately 68% were impacts to the occupant cell.

* 7 of the 10 vehicles involved in fatal crashes were cars or car derivatives. Ginpil found 57% of vehicles in fatal crashes were cars or car derivatives.

* In fatal crashes an oblique impact is the most common.

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Acknowledgments

The authors which to acknowledge the input to the Crashed Vehicle Study by the RTA vehicle inspectors. It is their enthusiasm and input that assists in learning more about crash dynamics and injuries.

The Authors would also like to thank staff from Taverner Research Pty Ltd, who are assisting in project management, data entry and analysis. Thanks also to Bill Bailey from Bailey Consulting for his input into the analysis.

***Road User Movement Groupings

10 - Vehicles from adjacent directions (intersections only)

20 - Vehicles from opposing directions

30 - Vehicles from same direction

40 - Manoeuvring

50 - Overtaking

60 - On-Path

70 - Off path on straight

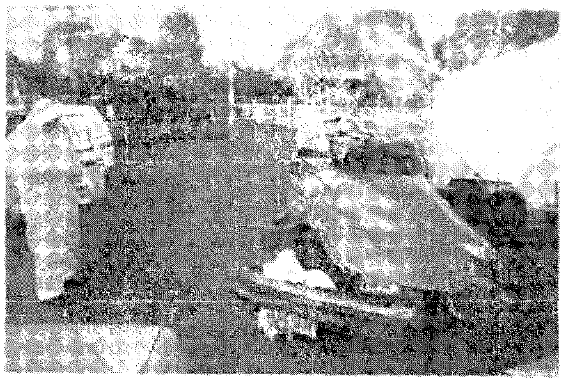
80 - Off path, on curve or turning

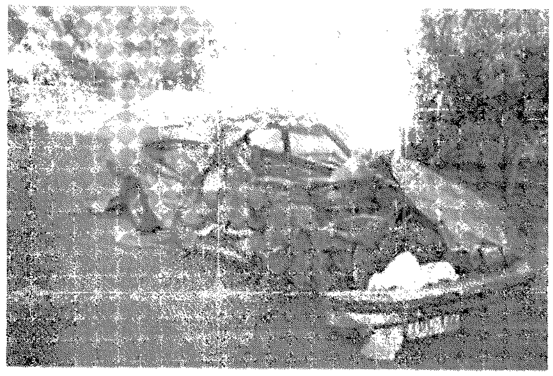
90 - Passengers & Miscellaneous

99 - Unknown

Annexure 1

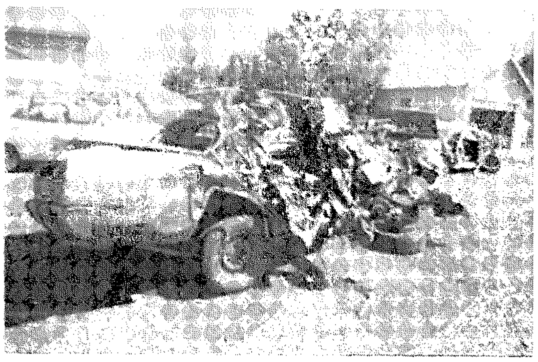
Fatal Crash Descriptions

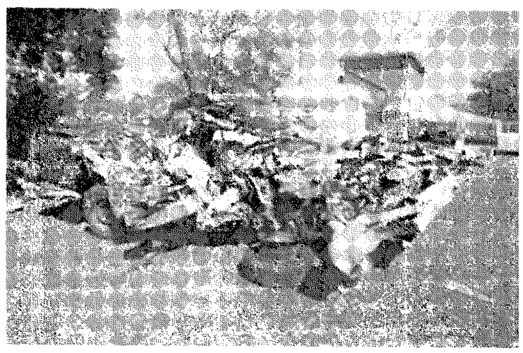


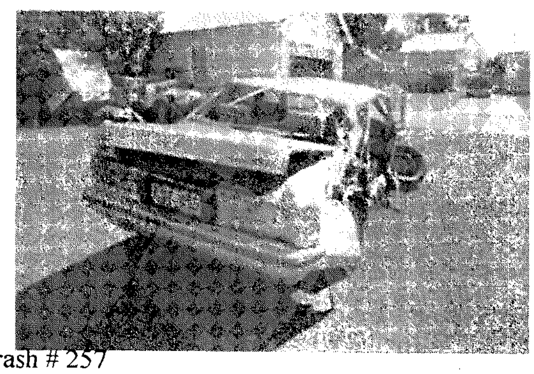


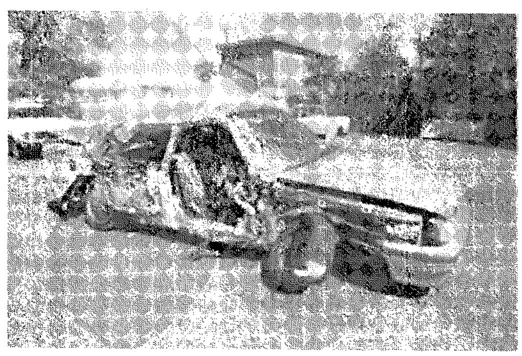
Crash # 016

Driver of Toyota Celica crossed median on expressway and right rear just clipped left side of small Honda sedan (minor damage). Right front area of laden light truck with trailer (Isuzu NPR) then collided heavily with right front of Celica at about the level of the right front wheel producing heavy intrusion rearwards to the level of the drivers seat and laterally to approx the centre of the cabin. The floor pan of the Celica failed to engage with the truck . Front passenger of Celica fatally injured. Possible strategies include strengthening in the area where the floor pan joins the firewall and base of A pillar to fully engage floor pan during main impact. This critical vehicle aspect in the crash was the intrusion into the occupant cell via the Front quadrant of the vehicle. Underrun protection on the truck was also an issue.





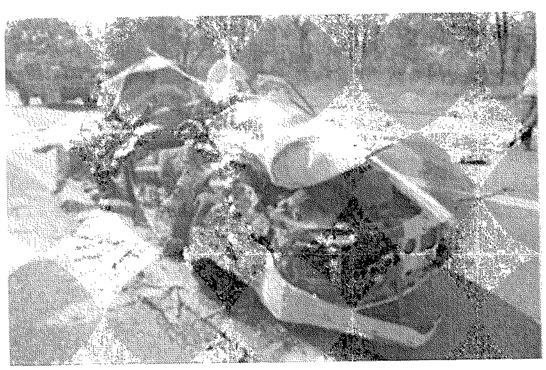


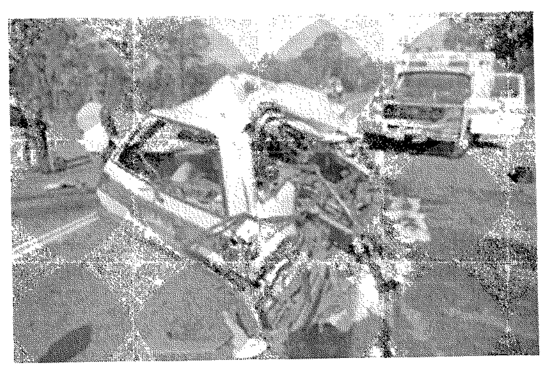


Crash # 257

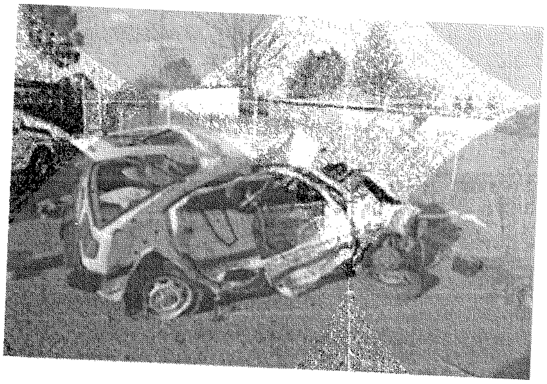
Four fatalities in large passenger car which sustained a frontal offset (~50%) collision with laden semi trailer which veered to incorrect side of major highway. Closing speed 185 km/hr per police records. Severe under ride of the car produced complete intrusion of the complete right side of the cabin and resulted in fatal injuries to driver, right rear occupant, left front occupant and a child in the lap of the left front occupant. Strategies for improved occupant protection would be front underrun improvements to the truck and again a more rigid "A" pillar region.

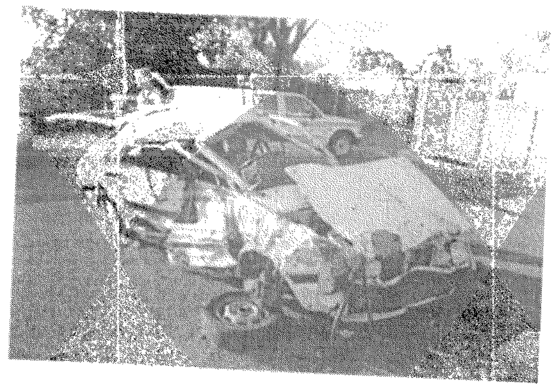
Crash #281
Elderly (> 70rs) single occupant driver of Toyota utility (not car deriv) turned right onto major highway from side road into path of laden semi trailer. Driver of semi trailer swerved right and left front of rigid bull bar collided with front right mudguard and cabin of Toyota producing heavy intrusion and buckling of the A pillar and rearward deflection of the steering wheel. Significant override component in vehicle interaction with the bull bar which was not significantly damaged in impact.



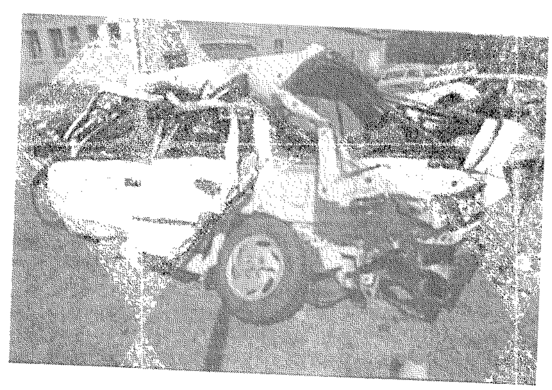


Crash # 400
Single occupant driver of Mitsubishi Sigma station sedan was killed when his vehicle veered to the incorrect side of a main road in a rural area and collided in a frontal offset with a laden rigid truck (>24 t GCM). Severe under ride with structure above window sill level collapsed to the level of the rear axle. Heavy engagement between engine/front suspension of car and right axle/wheel of truck.
Unsurvivable intrusion to right side of cabin from under ride. Possible vehicle strategies include i) underride barrier on truck, ii) deflection barrier on truck.

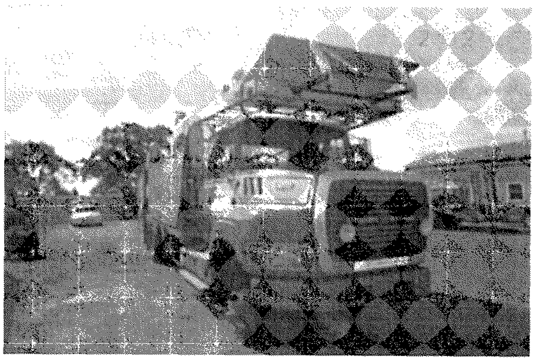
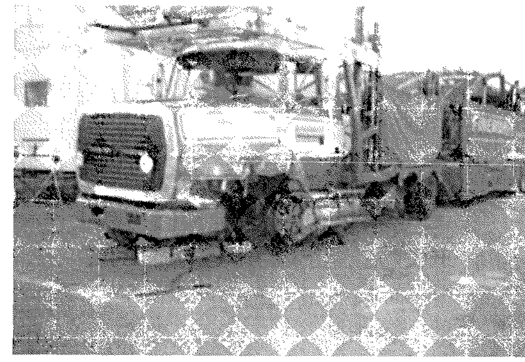




Crash # 812
Single occupant driver of Mitsubishi Colt sedan sustained fatal injuries in heavy side impact from Toyota Corona station sedan at cross intersection after Colt sedan passed through control sign.
Heavy distributed intrusion to complete right side of cabin extending to the midline (Dam7=6/9)
Possible strategies include increased lateral strength to retain cabin integrity.



Crash # 944
Single occupant driver killed in Hyundai Excel which veered across major highway, sideswiped Nissan Patrol and trailer producing moderate sideswipe damage to Hyundai (probably not fatal injuries). First impact rotated Hyundai CW into path of heavy coach with bull bar. Heavy distributed impact with complete left side producing general intrusion to the centreline of the cabin. Driver wearing seatbelt. Little permanent deformation of right side of cabin. Possibly survivable if driver was effectively restrained against lateral movement across the midline of the cabin.

Crash # 1039
Rigid Truck lost control crashing in Armco Railing.. Truck received minor damage, but driver was ejected and fatally injured

SIDE IMPACT REGULATION BENEFITS FOR AUSTRALIA

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ABSTRACT

This paper reports the benefits of Australia adopting either the current FMVSS 214 or the proposed ECE Regulation 95 side impact regulation. A Harm analysis was undertaken following a one-day meeting in Munich, Germany, involving a panel of international specialists to determine the likely injury reductions of both standards. Using the Harm reduction method, the total benefit for FMVSS 214 was subsequently estimated to be A$136 million annually or A$147 per car using a 7% discount rate and a 25 year fleet life. The equivalent figures for ECE Reg 95 were A$147 million with a unit Harm benefit of A$159 per car. Given an implementation cost of A$100 per car, it is evident that either standard is likely to be cost-effective for Australia.

INTRODUCTION

The responsible authorities in the USA and Europe have decided on <u>two</u> fundamentally different test procedures and test dummies for dynamic side impact regulation that cannot be harmonised. The US introduced a revised Federal Motor Vehicle Safety Stanndard 214 dynamic side impact standard in September 1993 with a phased introduction for full implementation by the end of 1996. The European regulation is due to be introduced for all new vehicles manufactured after October 1995. In May 1995, the Federal Office of Road Safety (FORS) released for comment a draft ADR for dynamic side impact protection which allows compliance to be demonstrated for either the US FMVSS 214 or the ECE Regulation 95.

The Monash University Accident Research Centre (MUARC) conducted a study on behalf of FORS to examine what the benefits would be if either standard was to apply to vehicles sold in this country.

FMVSS 214 Regulation

The US standard calls for a moving "crabbed configuration" deformable barrier to be propelled into a stationary test vehicle at 33.5mph (54km/h) at an angle of 27deg. Two Side Impact Dummies (SID) as defined by the National Highway Traffic Safety Administration in Washington DC are positioned in the front and rear near-side seating positions and instrumented to record peak accelerations of the spine and ribs, [TTI(d)] and pelvis (g force). Acceptance criteria specify dummy measures not exceeding a TTI(d) value of 85 and pelvic accelerations of 130 g's. The impacting barrier is essentially a homogeneous construction of 45 psi honeycomb material with a harder protruding bumper section. It is 3000lb (1360kgm) in weight, 66" (1.68 metres) long and has a ground clearance height of 11" (280mm).

ECE Regulation 95

The European procedure requires a perpendicular test of a movable barrier into a stationary vehicle at 50km/h. The barrier is to be 950kgm in weight, with a 1500mm wide barrier face made up of 6 variable density sections which supposedly represent the stiffness values of European cars. The barrier height was originally 300mm but was dropped to 260mm when the regulation was pronounced.

The specifications call for a single dummy (European designed EUROSID model) positioned in the front seat on the struck side of the test car. Injury measures include a maximum Head Performance Criterion (HPC) of 1000Hz, a peak chest deflection on any rib of 42mm maximum with a peak viscous response (V*C is to be recorded for 2 years as a transitional arrangement, but not counted as pass/fail, after which 1.0msec is likely to apply), and peak abdominal and pelvic force criteria.

The US standard was first implemented for 1994 model vehicles and prescribed a phased introduction of 10% in the first year, 25% for 1995 vehicles, 40% for 1996 vehicles and 100% for 1997 models and beyond. The European procedure was been promulgated for introduction for all European vehicles manufactured after September 1995.

INJURY REDUCTIONS

There were practically no published data available reporting injury reductions associated with either standard. The US government had published a Notice of Proposed Rulemaking prior to the implementation of FMVSS 214 in which they had argued for certain injury reductions in the USA for their standard. However, these figures had been strongly criticised by a number of reports, both outside and within the USA. Moreover, there was a dearth of information from Europe for ECE Reg 95. It was decided, therefore, to assemble a panel of overseas experts to advise on the likely injury benefits that would accrue to Australia for either standards.

MUARC subsequently organised a one-day workshop in May 1994 in conjunction with the Enhanced Safety Vehicles conference in Munich, Germany involving staff from these two international agencies as well as a number of other international specialists from research organisations and automobile manufacturers. The workshop provided an up-to-date account of side impact regulation developments and involved a lengthy discussion of the likely injury benefits if Australia adopted either of these two standards. The major findings from the workshop are listed below:

- While neither of the two standards currently seems optimal for occupant protection, there is little doubt that either one would be better than none at all. Thus, introducing a dynamic side impact test requirement that allows manufacturers to meet either standard seems desirable for Australia at this time.
- The two standards are different in almost every respect. Both standards appeared to have their individual strengths and weaknesses.
- The US regulation seemed best at mimicking car-to-car crash patterns and intrusions, at least for US vehicles. The crabbed crash configuration was felt to be more punishing on the vehicle and ensured a degree of protection for rear seat occupants.
- The SID dummy was disadvantaged by its reliance on measures of acceleration alone and by its inability to measure abdomen injuries. Both the EUROSID and BIOSID dummies appear to offer improved measurement capabilities and higher biofidelity.
- It was felt that the FMVSS 214 under certain circumstances could allow inappropriate counter-measures (more stiff structures), although this view was not universal.
- The proposed European standard appeared to have the better crash dummy in terms of its biofidelity. However, at least for US vehicles, its barrier and crash configuration did not always simulate real world crash patterns.
- Lowering the barrier height to 260mm was felt to be a backward step and disregards most of the development work behind ECE Reg 95. Early testing suggests around a 30% drop in sensitivity in V*C and pubic force from this height reduction. This is likely to have negative implications for side impact countermeasures.
- It was felt that the European standard (at least for European vehicles) was likely to lead to cars with more rigid surrounding structures but with weaker highly padded doors.

The one-day workshop also highlighted various sources of biomechanical data, test results and specific injury studies for use in the Harm analysis to follow. The authors are grateful to all those who participated at the workshop and for their valuable contributions.

CALCULATING HARM REDUCTIONS

The *Harm Reduction* method has been used previously for estimating the likely benefits of new occupant protection countermeasures (see Monash University Accident Research Centre 1992). Harm is a road trauma metric which contains both frequency and cost components and is therefore able to express the likely reductions in injuries from the introduction of a new measure into financial benefits.

The systematic building block approach used in this study permitted a body region by contact source analysis of benefits which provided an objective estimate of the consequences of Australia adopting either of the two candidate regulations.

Data Sources Available

An Australia-wide database was necessary to assess the likely injury reductions for both standards. A detailed database was constructed in 1991 of national injury patterns by body regions, restraint conditions and contact sources, along with a series of resultant Harm matrices using BTCE human capital cost estimates (Monash University Accident Research Centre 1992). This comprehensive trauma analysis, based on over 500 real-world crashes examined in the Crash Vehicle File by the Monash University Accident Research Centre, along with statewide TAC no-fault insurance and FORS fatal files, offered a baseline trauma pattern upon which estimates of Harm reductions could be made.

While this database was several years old, it nevertheless was still the most up-to-date source of baseline information available. Moreover, while the numbers of crashes (and hence injuries) have reduced over the last 4 years, their costs have risen such that the overall cost of trauma is probably still similar to that estimated for 1991. Thus, this database was judged suitable for use in this study, too.

Relevance Assumptions

A number of assumptions were necessary for determining the likely benefits of a dynamic side impact regulation for Australia. The findings from the one-day workshop were subsequently used as a basis for these assumptions and these are fully detailed and justified in Fildes et al (1995). They are summarised below:

1: The standard which requires a test at a crash severity of around 27km/h delta-V will provide benefits at crash speeds up to 64km/h. No benefits are assumed above this speed.

2: Near-side occupants who sustain AIS 5 or 6 fatal head injuries are excluded from any benefit from the standards. Reductions in chest injuries to occupants who sustain a non-fatal head injury are included.

3: The benefits will apply to both car-to-car and car-to-fixed-object in side impact collisions

4: The benefits will apply to occupants involved in both non-compartment and compartment side impacts.

5: The benefits will apply to hard thorax (chest including liver, kidney and the spleen), pelvic, femur, shoulder, upper extremity, head and face injuries caused by contact with the door panel, hardware or armrest.

6: The injury risk curves for TTI and V*C apply to the range of impact speeds for side crashes at severities less than 64km/h for injuries of AIS 3 or greater.

7: The effectiveness of an incremental reduction in TTI on chest injuries from interior door contacts to near-side occupants is as outlined above. The AIS 3+ curve is used for calculating injury reductions involving AIS 1 to 4 injuries while the AIS 5+ curve is used for calculating AIS 5 and 6 injury reductions.

8: The average TTI for Australian cars can be estimated by comparing the hard thorax injury distribution for near-side occupants in Australian crashes with those in the NCSS and adjusting MUARC values to include more low severity crash

9: The hard thorax injury reductions to occupants in side impact crashes in Australia are as listed in Table 2.4 and detailed above.

10: Based on the above analysis, a relevance factor of 0.45 is expected for AIS 3 to 6 hard thorax injuries over the crash severity range of 0 to 64 km/h. For the more minor AIS 1 and 2 injuries, a relevance factor of 0.90 would be expected, based on the conservative evidence of NHTSA which suggests these low level injuries are at least twice as frequent as the more severe ones.

11: The reduction in hard thorax injuries by the use of SID and TTI measures are 2 - AIS over the crash severity range listed above. The hard thorax injury reductions possible using EUROSID and V*C measures would be 3-AIS across the same delta-V range.

12: The pelvic fracture relationship published by Haffner is valid and the crash performance in terms of the risk of pelvic injury of US and Australian vehicles is similar. Injury reductions to AIS 1 are expected for a relevant percentage of side impact crashes.

13: The relevance factors for abdominal injuries are the same as those expected for the hard thorax. However, an overall injury reduction of AIS 1 is expected for SID and AIS 3 for EUROSID, assuming abdominal injury criteria is applied when using this test dummy.

14: All head injuries in side impacts from contact with the door panel are reduced by 2 AIS and face injuries by 1 AIS over the range 0 to 64km/h. For EUROSID, an additional benefit of 2 AIS applies to head contacts with the side rails.

15: All upper extremity, shoulder and lower extremity injuries in side impacts from contact with the door panel and fittings will be reduced by 1 AIS over the impact range from 0 to 64km/h.

16: A dynamic side impact standard will result in the elimination of all injuries with exterior contacts for far-side occupants ejected through the far-side door over the severity range 0 to 40 km/h.

17: The crash severity relevance figure at each AIS level for hard thorax to door panel contacts is the ratio of those injured at each AIS level at delta-Vs below 64 km/h to all injuries at each AIS for which delta-V is known. Similar relevance figures can be for other body regions where sufficient data exists.

RESULTS OF THE ANALYSIS

A detailed system of spreadsheets was assembled for calculating the benefits of both standards. Relevance figures were assigned by body region and seating position (near- or far-side of the vehicle) and the subsequent Harm units removed were computed. The savings by body region and seating position were then summed to arrive at the total estimate of savings for both standards.

Annual Harm saved was converted into Unit Harm benefits using a 7% discount rate and historical vehicle sales and write-off figures. A 7% discount rate is used in the majority of similar Commonwealth Government feasibility studies and is generally regarded as a rather conservative estimate of unit benefits. The results of the analysis are illustrated in Table 1 below.

Table 1
Summary Table of Harm Benefits for FMVSS 214 and ECE Reg 95.

BODY REGION INJURED		U.S. STANDARD FMVSS 214 $million	EUROPEAN ECE Reg. 95 $million
HEAD INJURIES	near-side	9.7	10.8
	far-side	17.7	18.1
FACIAL INJURIES	near-side	0.7	0.8
	far-side	0.1	0.1
HARD THORAX INJURIES	near-side	54.4	61.7
	far-side	3.2	3.6
ABDOMINAL INJURIES	near-side	6.5	8.4
	far-side	0.1	0.1
PELVIC INJURIES	near-side	4.4	4.4
	far-side	0.1	0.1
UPPER LIMB INJURIES	near-side	17.0	17.0
	far-side	3.6	3.6
LOWER LIMB INJURIES	near-side	17.6	17.6
	far-side	1.2	1.2
TOTAL NEAR-SIDE HARM SAVED ($million)		**110.3**	**120.7**
TOTAL FAR-SIDE HARM SAVED ($million)		**25.8**	**26.7**
TOTAL HARM SAVED ANNUALLY ($million)		**136.1**	**147.4**
UNIT HARM ($'s per life of a vehicle)		**$147.20**	**$159.40**

For the US standard, FMVSS 214, the total Harm saved annually in Australia, assuming all vehicles were to comply, would amount to A$136 million based on 1991 crash patterns and costs of injuries. This represents a 4.7% reduction in vehicle occupant trauma annually if FMVSS 214 were to apply in Australia. The unit benefit per car would be $147 based on a 7% discount rate and a 25 year fleet life.

The equivalent figures for ECE Reg. 95 is estimated to be A$147 million each year assuming all vehicles in the fleet complied. This would yield a slightly higher 5.1% reduction in occupant trauma annually over the US standard with a unit benefit per car of $159.

IMPLEMENTATION COSTS & BCR

The Federal Chamber of Automobile Industries (FCAI) represents all vehicle manufacturers in Australia. They were asked to provide estimates of the average cost to manufacturers to meet the standard. Their estimate of the additional cost that manufacturers would have to spend on each vehicle to comply with either FMVSS 214 or ECE Reg. 95 was A$100 per vehicle. This cost includes additional materials and processes as well as design and testing charges.

On this basis, it is estimated that either standard would be cost-effective, yielding a Benefit-Cost-Ratio of between 1.4 and 1.5 assuming a 7% discount rate. The BCR would be even greater for a lower discount rate of 5% which is becoming more widely accepted among the safety regulators in this country.

DISCUSSION OF THESE RESULTS

These findings are quite impressive and suggest there would be merit in ensuring that all Australian vehicles meet either standard.

It should be stressed that there was a high degree of confidence in calculating the benefits of FMVSS 214 as there were data available on the likely effectiveness based on US crash patterns and vehicle population which was able to be converted into equivalent Australian figures using the Crashed Vehicle File data and other sources of available information. However, unlike the US standard, the was little or no information available on the likely effects of ECE Regulation 95 in reducing injuries. Hence, these figures can only be viewed as indicative estimates at this stage. It is fair to say, though, that these estimates have some validity, given input from European experts and comparative research performed by Transport Canada.

The likely consequences of recent amendments to this proposed regulation to reduce the barrier height from 300mm to 260mm has not been thoroughly researched overseas. However, a recent study by the Federal Office of Road Safety suggests that this may not be all that significant in this country.

Whether either of these two standards will optimise the benefits available in side impact protection is still an open question. There was a suggestion that some form of a hybrid standard involving the US crash test procedure but with a more sensitive test dummy would lead to even larger benefits. The need for greater attention to head injuries in these standards was also noted.

CONCLUSIONS AND RECOMMENDATIONS

The results of this study show that there are likely to be modest benefits to Australians in reduced vehicle trauma of around 5% per annum if all vehicles on the road were to meet either the existing US dynamic side impact standard FMVSS 214 or the proposed European equivalent ECE Regulation 95. With BCR's of 1.4 and 1.5 respectively, there seems to be a possible marginal advantage for the European standard.

Three recommendations therefore seem warranted from this study:

1. *That the Australian Design Rule system include a new or revised regulation mandating that all vehicles sold in Australia be required to meet either FMVSS 214 or ECE Regulation 95. A suitable lead time would need to be negotiated with the manufacturers.*

2. *That further research be undertaken to examine whether a hybrid standard would lead to further improvements in occupant protection. In the event that there are sizeable benefits, these results be used to bring about a harmonised side impact standard in the long-term.*

3. *That further research be carried out into the need for additional regulations aimed at reducing head, neck and spinal injuries further in front and side impacts.*

ACKNOWLEDGMENTS

The authors are grateful to the Federal Office of Road Safety for sponsoring this work and to the expert panel members who willingly gave up their time to help with this research. The views expressed are those of the authors and not necessarily those of the sponsors or Monash University.

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REQUIREMENTS OF COMPREHENSIVE SIDE PROTECTION AND THEIR EFFECTS ON CAR DEVELOPMENT

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Paper number 96-S6-W-16

SUMMARY

In recent years more attention has been given to the side impact following the reductions made in the severity of frontal collisions in the past. The need for vehicle designs giving occupants a higher level of safety in the event of a side impact has also been recognised by the legislators and resulted in introduction of the standard FMVSS 214 dynamic in the USA in 1993. In 1998, the directive R95 is to be introduced in Europe, meaning that there will be two different international standards for the dynamic side impact test in the future. These standards differ enormously not only in their test conditions but also in the construction of the dummies and the biomechanical assessment criteria for the latter.

In this respect, a vehicle manufacturer with world-wide activities such as Volkswagen AG is obliged to develop its new vehicles to meet the requirements of both the existing laws. At the same time, the requirements to be met by the vehicle which are revealed by the effects analysis of real accidents must, of course, not be overlooked. The development process is confronted with a large number of conflicting objectives resulting from different requirements. The changing basic conditions for vehicle development, such as reduction of development times, systematic use of lightweight designs etc., give rise to further requirements, which can only be met with the use of modern development tools.

This paper will discuss in detail design requirements for comprehensive side protection and necessary development methods and tools. In addition, the conflicting objectives of the various crash requirements encountered in the design process will be described. Particular attention will be paid to FEM structural computation and occupant simulation, and to components and vehicle testing.

INTRODUCTION

For some years now, the development departments of vehicle manufacturers, and the legislators and consumers' associations world-wide have attached increased importance to the side impact. The reason for this is well known and can be found in the relevant accident statistics. Although side collisions occur less frequently than frontal impacts, the safety optimisation of the front vehicle structure and of the restraint system carried out in the past had the effect that the major potential threat to passengers shifted from frontal impacts to side impacts. This can be seen in Figure 1, which is based on evaluation of accident data from the Hanover area (data basis: 6178 vehicles involved in accidents in the period 1975 to 1993) /1/.

This demonstrates the fact that the proportion of frontal collisions decreases with increasing injury severity, whilst the proportion of side collisions increases correspondingly. It can thus be said, for severe and critical injuries, that frontal and side collisions are of approximately equal importance, and all other types of collision play only a subordinate role.

In the mid to late 1980s the National Highway Traffic Safety Administration in the USA tightened up the legal standards for protection in side collisions because of the major potential threat posed by side collisions. As a result FMVSS 214 (quasistatic door intrusion test) was augmented by a dynamic impact test with a deformable barrier. The fulfilment criterion for crush resistance was augmented by an occupant assessment using specially developed side impact dummies (US-SID) /2/.

In Europe the ECE and European Union also developed a dynamic impact test at the beginning of the 1990s /3/. Substantial parts of it differ from the American law. It is to be obligatory for new type approvals as of October 1998.

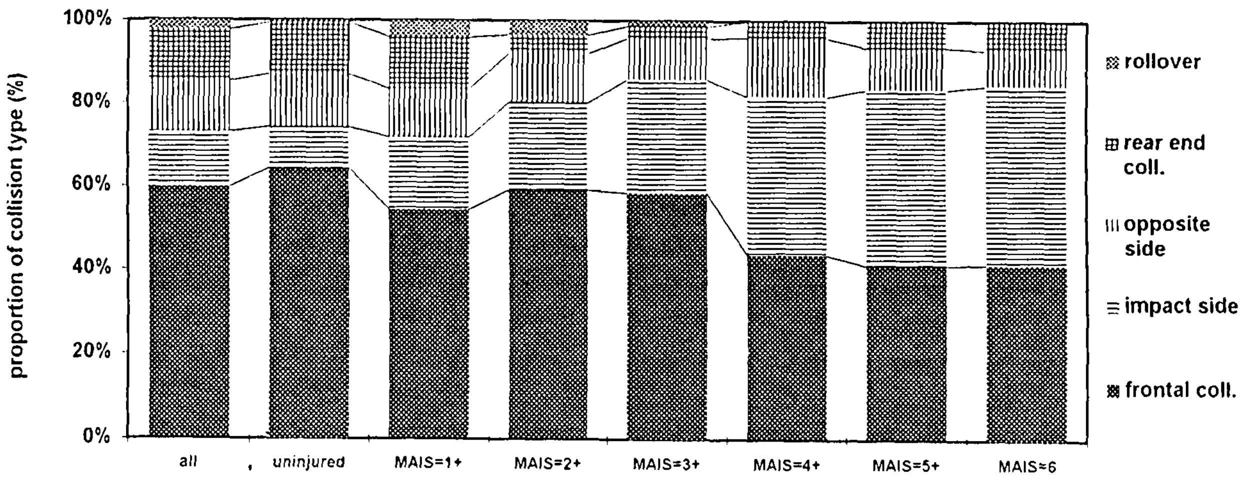

Figure 1: Proportions of the types of collisions in the overall number of accidents with belted-in car occupants suffering injuries of varying degrees of severity /1/

In addition to these two side impact tests, there are further laws and guidelines which concern the protection of occupants in side collisions. FMVSS 206, for example, describes the requirements and testing methods for side-door latches and hinges. Doors and tailgates which cannot be opened could decide between life and death in multiple collisions and in collisions involving occupants who are not belted in. Furthermore, in the US side impact configuration, the requirement for fuel system integrity described in FMVSS 301 must be met. The major potential threat of head injury in side collisions as observed in real accidents can, to a certain extent, be reduced by the future FMVSS 201 (head impact in the vehicle interior).

These considerations are the basis of every development in side impact research. At Volkswagen AG, findings made from real accidents are also taken into account in development, as is the adherence to legal basic conditions for the protection of occupants in side collisions. An accident database provides valuable assistance here. The safety engineers responsible, however, find themselves in a difficult position. They must take different, and in some cases contradictory, requirements into account which can be categorised on the one hand as global, and on the other hand as customer-specific boundary conditions, and beyond this also as safety-relevant requirements. The requirements are depicted in Figure 2. It is obvious that the complexity of these additional requirements leads to new paths in the development of motor vehicles. In addition, the tendency of vehicle manufacturers to develop so-called world cars demanding the simultaneous fulfilment of both US and ECE side impact requirements will lead to conflicting objectives, even if only the side impact is considered. These two existing vehicle tests often give contradictory assessment results for the same vehicle. Even within ECE-R 95, the use of the different deformation elements presently available on the market in front of the moving barrier can lead to differing results - even to the extent of non-fulfilment - although they all meet the test requirements set down in ECE-R 95 to a greater or lesser extent.

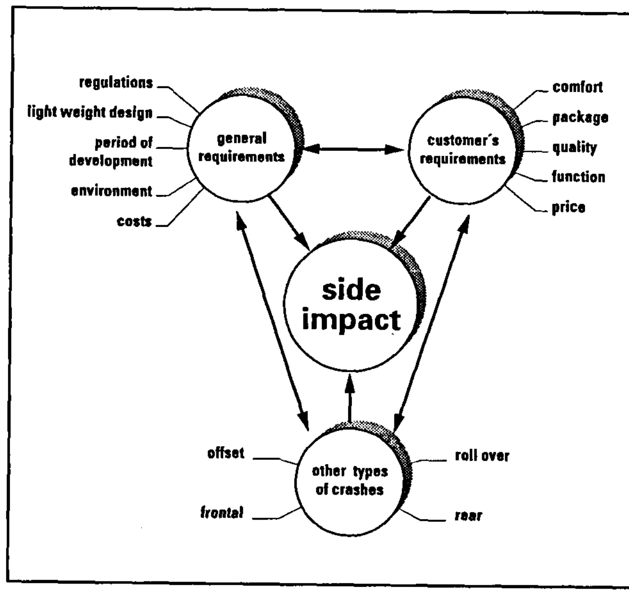

Figure 2: The side impact in the field of tension between additional requirements

For the development engineers, this dilemma means that it is first necessary to find the common denominators for the requirements of comprehensive side protection. For this purpose, Volkswagen AG engineers can refer to a large number of side impact tests carried out in accordance with FMVSS 214 and ECE-R 95, and vehicle to vehicle crash tests, which were performed in past years.

REQUIREMENTS OF COMPREHENSIVE SIDE PROTECTION

A comprehensive side protection concept must satisfy all the requirements made on a vehicle by real accidents and by FMVSS 214 and ECE-R 95 . In order to determine this general design, we shall first consider individual results for one vehicle type from the various crash tests required by law and results from vehicle to vehicle crash tests which will be used to represent the results from real accidents.

FMVSS 214 - The general principles of the side impact carried out in accordance with FMVSS 214 are well known (Figure 3). In this article, only the decisive influencing parameters are summarised so that they can later be compared with ECE-R 95. In addition to the occupant loading criteria, certain strength requirements, e.g. for the latches and hinges, must be fulfilled and the removal of the passengers after a crash must be assured.

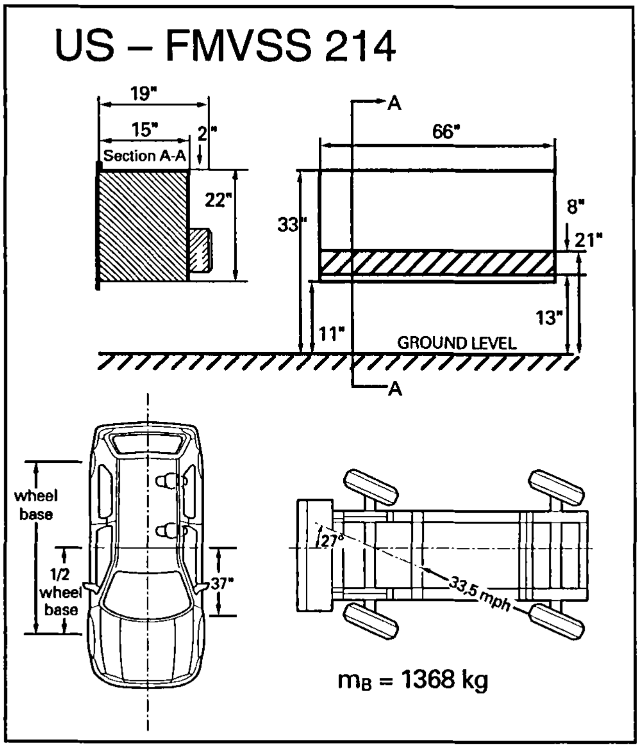

Figure 3: FMVSS 214 test arrangement

In a large number of US side impact tests with different vehicles, it was shown that the impact velocity of the side structure or door is decisive for the loading criteria of the US-SID. The occupant results are around 30% worse in SINCAP tests (38 mph) than in FMVSS 214 crashes (33.5 mph).

In order to reduce the penetration velocity, the side structure of the vehicles must be stiffened accordingly. The influence of targeted bodyshell reinforcement on the force-deflection behaviour of the vehicle structure and on the penetration velocity can be seen in Figure 4 and 5.

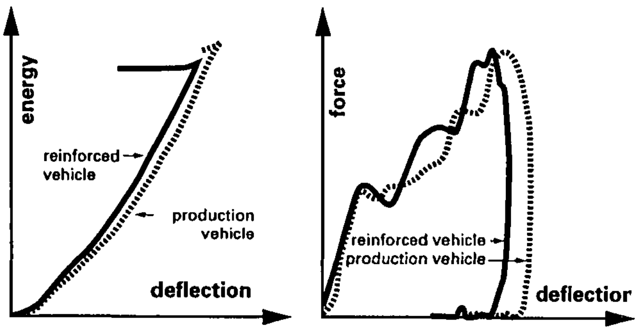

Figure 4: Force-deflection and energy-deflection characteristics of the vehicle structures

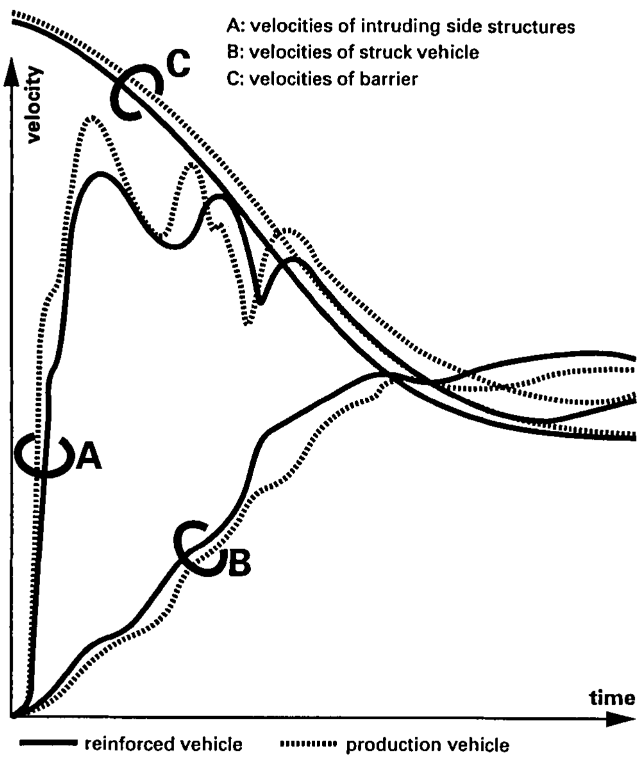

Figure 5: Penetration velocities versus time

In a real accident, a stiffened side structure leads to increased energy absorption by the impacting vehicle which is only subjected to relatively low stress anyway.

The maximum deformation of the reinforced, as compared with non-reinforced, vehicle has decreased by 11%. Securing an adequate survival space is an important development aim at Volkswagen. Only in this way injuries resulting from intruding components can be prevented, as a mere assessment of the measurable occupant loadings cannot guarantee adequate occupant protection. The penetration velocity of the reinforced vehicle structure is approx. 9% lower. The occupant values were reduced by approx. 20% in this case.

A further important influencing parameter is the force-deflection characteristic of the side structure striking the dummy. As the accelerations are limited for the US-SID, the force level of the internal side structure hitting the occupants (characteristic of the door trim with padding) must not exceed a certain limiting value. In order to make optimum use of the acceleration distance, the padding must have a high energy-absorption efficiency with low compressed dimensions. If a side airbag is used as an additional restraint system, this acceleration distance can be increased by approximately the free space between the occupant and the door trim.

The effectiveness of harmonised padding measures is shown in Figure 6. The padding reduces thoracic loading by 27% and pelvic acceleration by 37%.

The main requirements to be met by a vehicle in order to fulfil FMVSS 214 can thus be summed up in the following general design principles:

1. The side structure must have a high initial resistance to the impacting body in order to guarantee the lowest possible penetration velocity at the time of impact of the occupant.
2. Suitable restraint systems, such as padding and/or side airbags, which effect a targeted limitation of occupant acceleration, can be used for further reduction of the occupant loadings.

ECE-R 95 - ECE-R 95 is a much more recent law (Figure 7), and the EuroSID-1 and ECE deformation elements have not been available to the vehicle manufacturers for as long as the US-SID and the US deformation element.

In addition, the first ECE deformation elements did not comply with the legal requirements /4/. Elements which meet those requirements have only been available from AFL and Fritzmeier since the end of 1994.

A long period also elapsed whilst the law was still at a preliminary phase before an agreement was reached on the barrier height. Until the final decision was made to use ECE-R 95 as of 10/98 with a barrier height of 300 mm, it had been 260 mm for a long time. In addition to former evaluations /5/, Volkswagen carried out a study to determine the influence of the test parameters in the

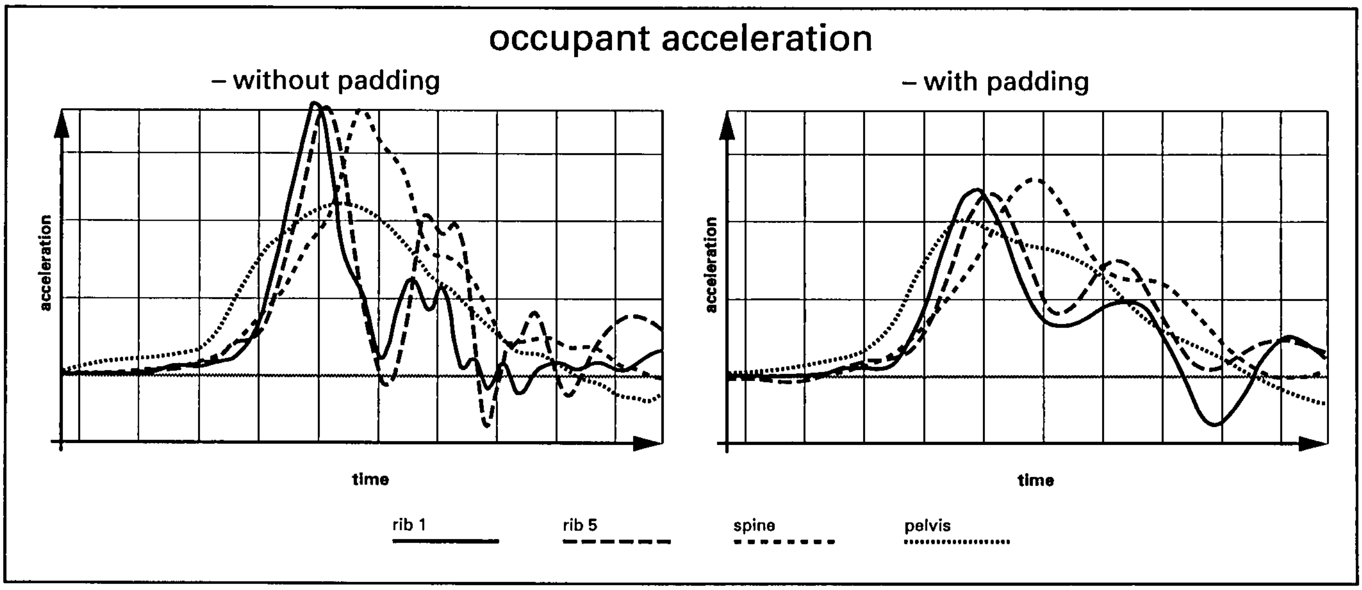

Figure 6: Occupant loadings for a vehicle with and without padding measures

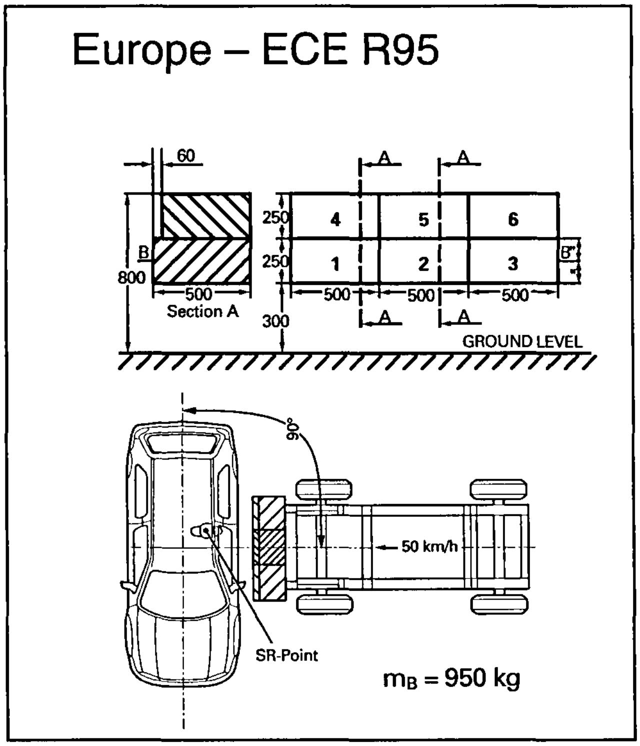

Figure 7: ECE-R 95 test arrangement

ECE-R 95 side impact. For this purpose, tests were performed with different deformation element types at different velocities and different heights in order to understand and describe the behaviour of vehicles and of the EuroSID-1 in the ECE crash, and to determine those features in common with the US tests necessary for development. This was complemented by computational work.

Although there is a building regulation for the NHTSA deformation element, the operational regulation for the impacting body described in ECE-R 95 led to very different deformation element types in which both the internal construction and the materials vary. Thus, Kenmont and Fritzmeier have developed a polyurethane element, and Hexcel and AFL construct the ECE element from aluminium honeycombs.

At Volkswagen, tests were carried out with Kenmont, Fritzmeier (old and new specifications), Hexcel and AFL elements. The results showed pronounced differences in penetration velocity, vehicle deformation and, above all, in occupant loadings. The influence of the deformation elements is the most obvious of these results if the energy quantities absorbed by the vehicles and the barriers are considered using the same vehicles and barrier heights but with three different deformation elements (Figure 8). This leads to greatly differing distributions of the energy absorbed between vehicle and barrier as represented in Figure 9. Figure 9 also shows the typical energy distribution between the one NHTSA element and the vehicle in a side impact carried out in accordance with FMVSS 214. In the US side impact, approximately 84% of the energy must be absorbed by the vehicle; only 16% of the energy is absorbed by the NHTSA element. In the ECE tests, the energy absorbed by the deformable barrier is between 19% and 34%. That is to say, the vehicle must absorb between 66% and 81% of the deformation energy, depending on the choice of deformation element. Differences of up to 20% in the intrusion depths occur as a result.

These considerable differences in deformation element stiffness in the vehicle crash can also be seen in the penetration velocities of the side structure (Figure 10). With equal test velocities, differences of up to 35% were determined for the three deformation elements.

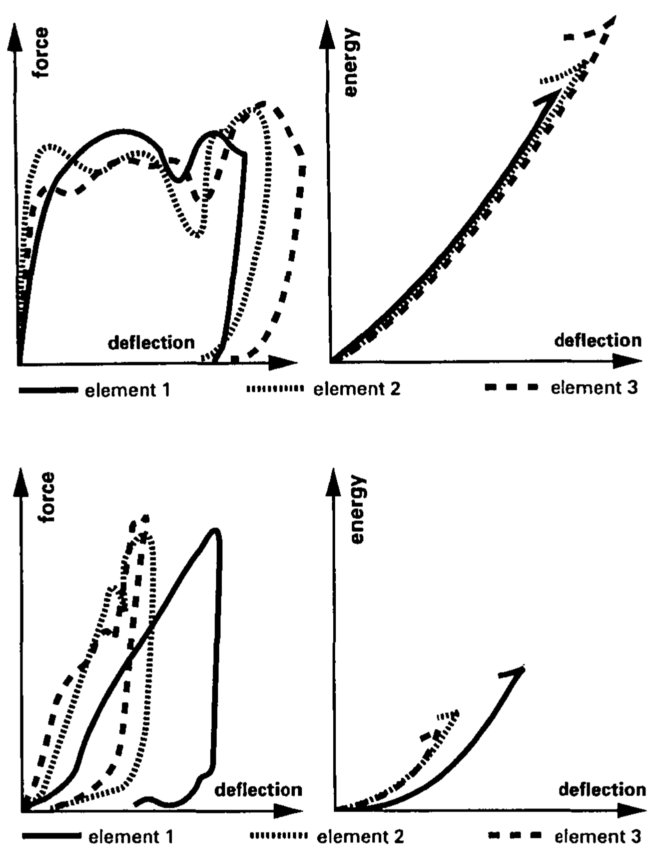

Figure 8: Energy absorbed by a 4-door vehicle (top) and barrier (bottom) for three different deformation elements

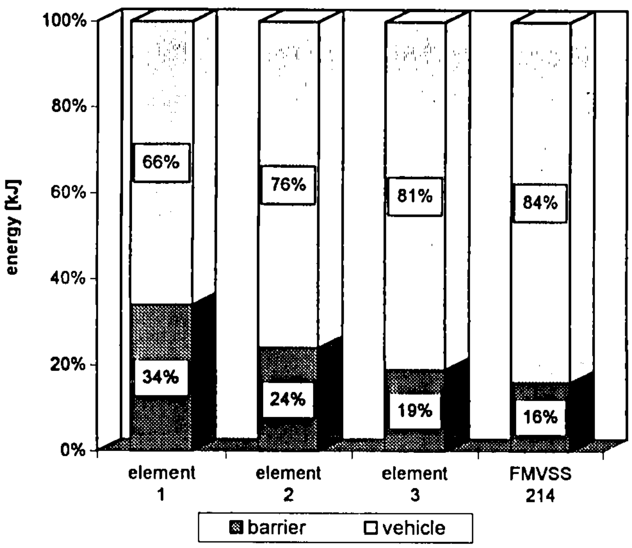

Figure 9: Energy distribution between barrier and vehicle (4-door) for three deformation elements and for a US side impact

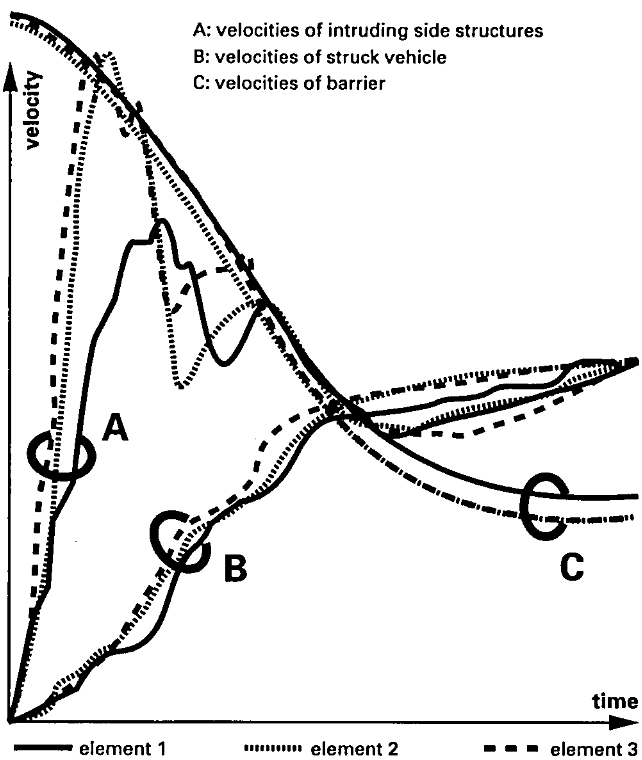

Figure 10: Penetration velocities for the 4-door vehicle with three deformation elements

As a result of the segmentation of the ECE deformation element, the influence of the number of doors of a vehicle is much greater than in the US deformation element. In contrast to a 4-door vehicle, the B-pillar on a 2-door vehicle is not struck by the central rigid block of the deformable element in most cases. On a 2-door vehicle, therefore, the largest amount of force is exerted on the door and is transferred to the pillars and the sill. The penetration kinematics thus differ significantly between 2 and 4-door vehicles.

There was, during the ECE-R 95 tests, a wide spread in the occupant values as a result of the elements used, the two barrier heights of 260 mm and 300 mm and the differing numbers of doors. The decisive correlations between occupant loadings and vehicle behaviour are to be detailed below.

If the mean occupant loadings and mean maximum B-pillar velocities of a vehicle class (the mean was determined for all barrier heights and all deformation elements, see Figures 11 and 12) are compared, it can be seen that the Rib Deflection Criterion (RDC) and the Viscous Criterion (VC) in the 2-door vehicle are significantly lower than in a 4-door vehicle. The Abdominal Peak Force (APF) and the Pubic Symphysis Force (PSPF) are lower in the 4-door vehicle than in the 2-door vehicle. No correlation could be determined for the Head Protection Criterion (HPC). The maximum B-pillar penetration velocity is significantly higher in the 2-door vehicle and thus does not have any direct influence on the thoracic loadings.

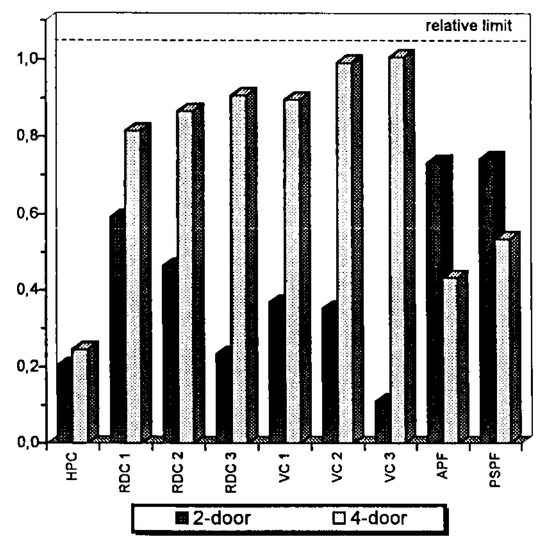

Figure 11: Influence of the number of doors on the occupant values

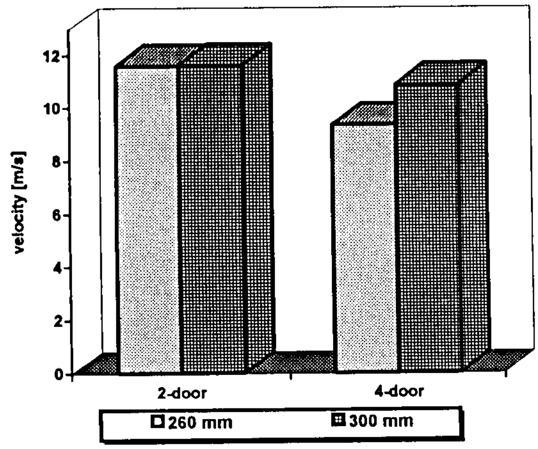

Figure 12: Mean maximum B-pillar velocities for 2- and 4-door vehicles for barrier heights of 260 mm and 300 mm

If the mean deformations in a vertical section through the vehicle at the level of the H-point (point position based on FMVSS 214) are compared, a significant tilting movement of the door about the x-axis of the vehicle can be seen (Figure 13).

In the evaluation of this deformation behaviour in various vehicles, a good correlation can be determined between the rib deflection / VC criterion and the difference in y-direction penetration between the upper door beam (MP 29) and the level of the H-point (MP 31) on the door outer panel. Figures 14 and 15 show the

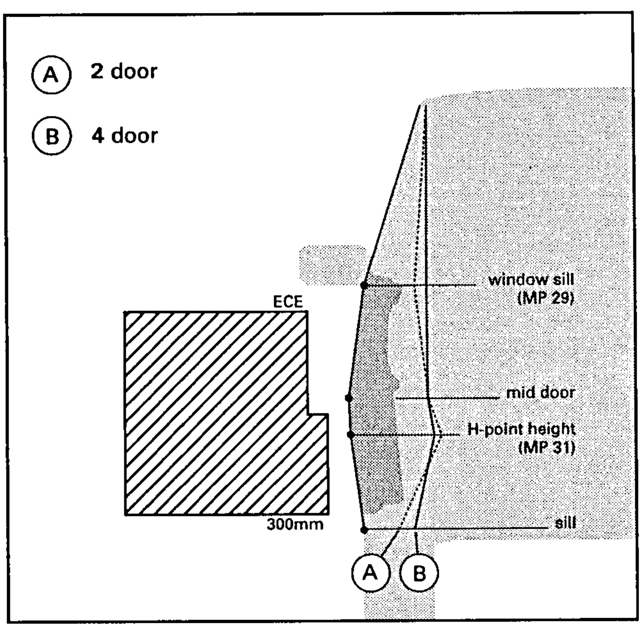

Figure 13: Comparison of the mean deformation of the door through the H-point in 2- and 4-door vehicles for 300 mm ground clearance and all barriers

significant reduction in thoracic loadings with decreasing penetration of the upper door beam **relative** to the penetration of the door at the level of the H-point. No correlation was found, however, between the absolute penetration depth and intrusion velocity and the

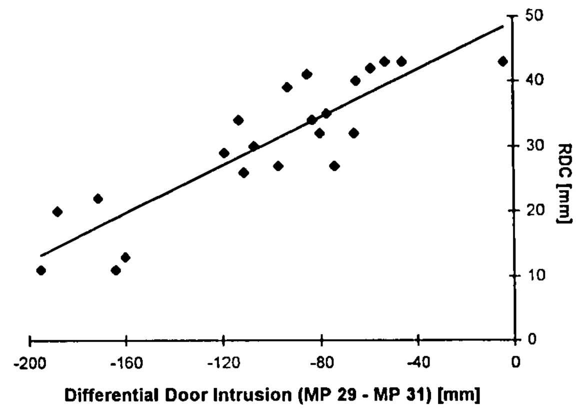

Figure 14: Correlation between the maximum RDC and the difference in door intrusion

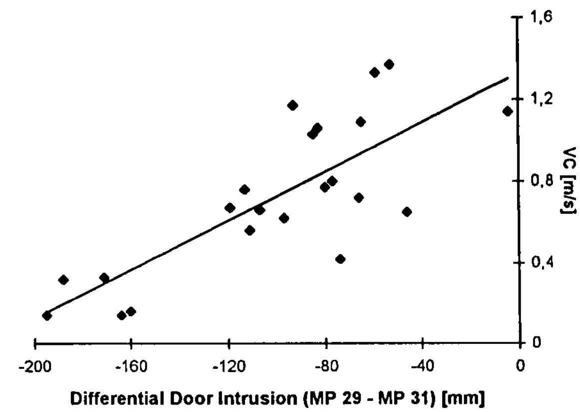

Figure 15: Correlation between the maximum VC and the difference in door intrusion

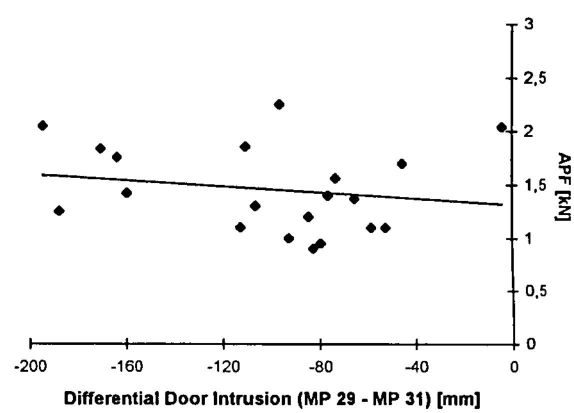

Figure 16: Correlation between the APF and the difference in door intrusion

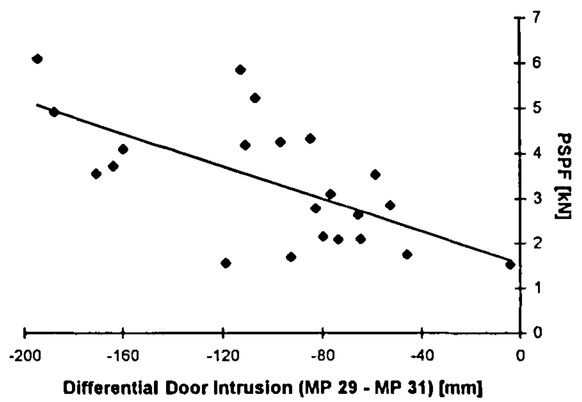

Figure 17: Correlation between the PSPF and the difference in door intrusion

EuroSID-1 response, see also /6/. There is a much weaker correlation tending slightly in the opposite direction for the abdominal forces and the Pubic Symphysis Force, as shown in Figures 16 and 17.

In contrast to the US side impact according to FMVSS 214, these results have produced the following general design principles in order to fulfil the requirements of the ECE side impact with EuroSID-1 dummies:

1. The side structure must be designed in such a way that the lower body is struck with limited force in a predetermined manner. It is important that this impact occurs before the upper body is struck in the upper door beam area. A weak correlation of the penetration velocity was observed only with the VC.
2. The restraint system must have a force-limiting effect in the area of the abdominal and pelvic impact. In the thoracic region, the restraint system must guarantee uniform distribution of loading on the ribs. Padding in the thoracic region must not lead to a reduction in the thoracic loading. Use of a suitable side airbag, on the other hand, leads to uniform, load-reducing introduction of force into the thorax.

Vehicle to vehicle side impact tests - Many vehicle to vehicle side impact tests were performed at Volkswagen because of the conflicting objectives resulting from the different design principles of FMVSS 214 and ECE-R 95. It became evident that the deformation patterns in the vehicle to vehicle impact can be compared neither with those in the ECE tests nor with those in the FMVSS 214 tests. A vertical section through a vehicle in the H-point (Figure 18) shows that the real vehicle structure allows the upper door beam region to intrude much less than the lower door region because of its shape and impact height.

In all the vehicle to vehicle tests, therefore, the EuroSID-1 loadings were not only far below the limit values but also below the values of corresponding ECE-R 95 tests, although the masses of the impacting vehicles were in some cases considerably higher than the 950 kg of the ECE barrier and the fronts of the vehicles were not lowered to simulate brake diving.

As the vehicle to vehicle tests showed, the loading of the struck vehicles is especially severe in the lower region. The penetration resistance must therefore be increased (safety catch, increased door-sill overlap area, etc.) for real accidents. Such a vehicle design, however, leads to increased thoracic loadings in the ECE-R 95 test. In the development phase of new vehicles, it has been shown that a vehicle can be well above the injury limiting values in FMVSS 214 but exhibit very low occupant loadings in ECE-R 95. These conflicting objectives indicate the necessity of more extensive fundamental biomechanical investigations and the incorporation of the results in further development of the side impact testing method.

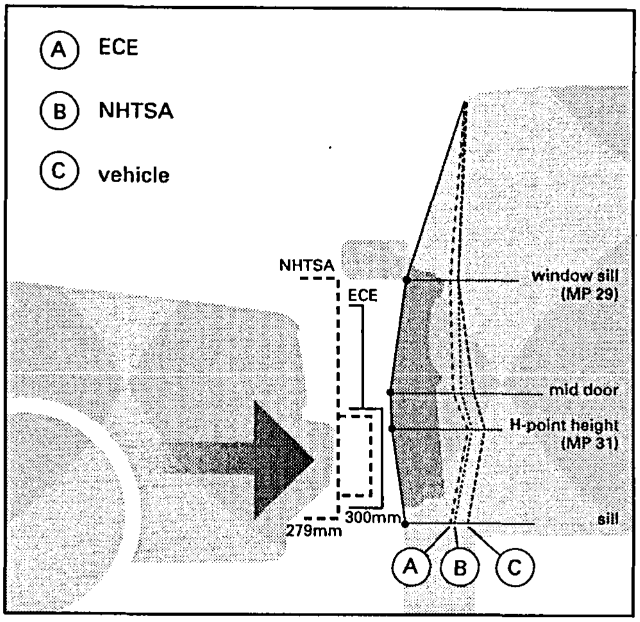

Figure 18: Vertical section through the H-point for a vehicle with FMVSS 214, ECE-R 95 and a vehicle-vehicle test

DEVELOPMENT TOOLS

FEM Structural Computation

In the first phase of development of a vehicle, information on the behaviour of the structure in the event of a side impact is of great significance for different concepts. The relevant components for side impact, such as the doors, A- and B-pillar concepts, sills, seats, roof and connection of these components should be examined and optimised as early as possible.

Computation with powerful super computers and modern computational methods (crash simulation) offers many possibilities and opens new perspectives for analysis of the impact and thus for optimisation of the vehicle structure. Crash simulation means that reliable information on lateral stiffness, deformation behaviour and penetration depth of the vehicle side structure can be ascertained even at a very early phase of development of a vehicle.

The descriptions of the geometry of the components of the vehicle are generally on hand in the form of CAD data records. Additional information (e.g. welded connections) may also be available in the form of a technical drawing. In the pre-processing, the components in the form of CAD data are modelled in the finite element network via the VDA interface by the ANSA network generator or the PATRAN pre-processor. The individual components are then assembled to form the complete vehicle and the material characteristics, boundary conditions and loadings are generated.

For the crash simulation, the LS-DYNA3D finite element package is used on a CRAY Y-MP 8/464 supercomputer (with 8 processors) or an NEC SX3. A whole series of software packages which are either commercially available or Volkswagen-developed are available for representation of the results. These range from representation of curves (e.g. energy balance as a function of time, cross-sectional forces as a function of time, penetration velocities as a function of time), sectional views and deformation patterns for the complete vehicle and individual components, to the animation of the crash on the screen of the work station. Figure 19 gives an overview of the results of the crash simulation.

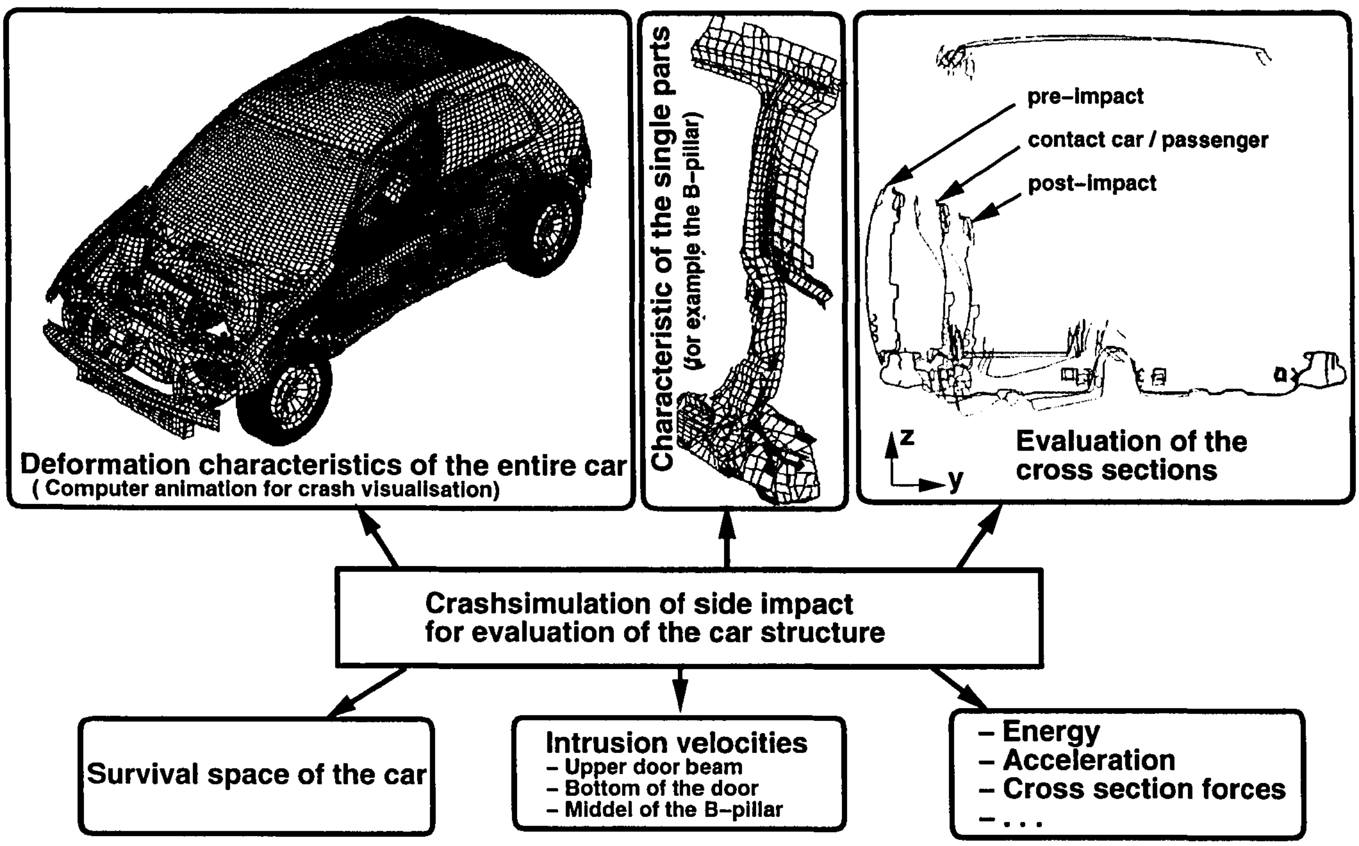

Figure 19: Overview of the results of the crash simulation

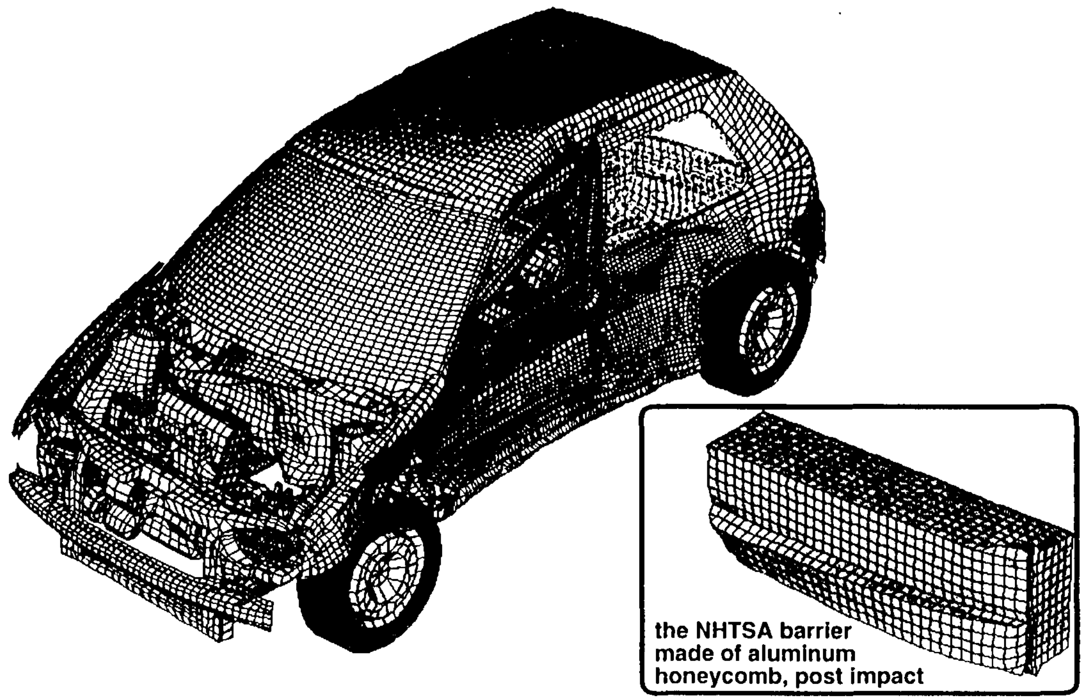

Figure 20: The deformed vehicle structure, post US side impact

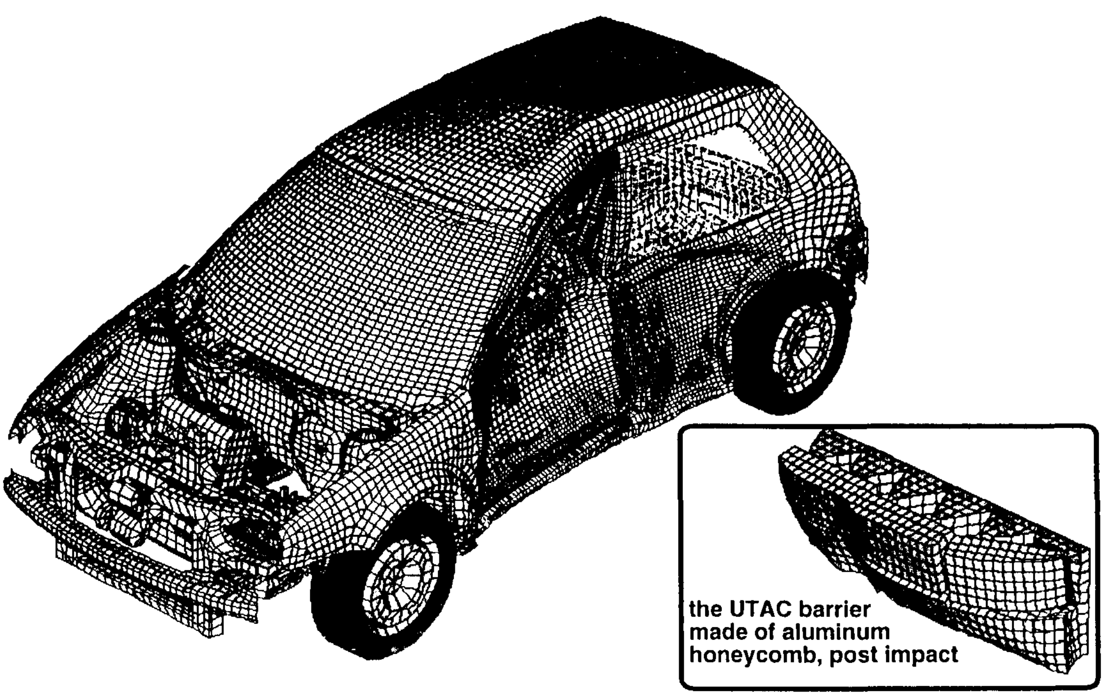

Figure 21: The deformed vehicle structure, post ECE side impact

The computational model for the simulation of a USA side impact test carried out in accordance with FMVSS 214 with an NHTSA barrier and a complete vehicle structure was discretised with 65,858 shell elements, 1,069 beam elements and 3,634 volume elements. For the bodywork panels, an isotropic material behaviour with a piecewise linear stress-strain curve was used, and for the honeycomb structure of the US barrier, the "metallic honeycomb material" in LS-DYNA3D. Account is taken of the high-strength material and of the dependence of the strain velocity-dependent behaviour of the panel. Figure 20 shows the deformed vehicle structure from the computational results after the US side impact. The vehicle structure has deformed uniformly.

The computational model for simulation of a European side impact test carried out in accordance with ECE-R95 with a UTAC deformation element and a complete vehicle structure was discretised with 67,587 shell elements, 1069 beam elements and 3110 volume elements. Figure 21 shows the deformed vehicle structure from the computational results after the European side impact.

A vehicle with comprehensive passive safety for side protection must have a high degree of side structure stiffness and a uniform deformation behaviour in the event of crash. At the same time, an adapted deformation characteristic in the interior (padding) is indispensable for the protection of the occupants.

The structure of the vehicle (sills, A- and B-pillars, cross members, etc.) can be designed initially on the basis of the requirements in respect of the survival space (penetration depth), the penetration velocity and the uniform deformation behaviour.

The "padding" can then be studied to a large extent on the basis of the maximum occupant loadings requirements, as defined in FMVSS 214 or the ECE-R95, with an FEM-US-SID (US side impact dummy on finite element basis) or an FEM-EuroSID-1 (EuroSID-1 on finite element basis). The term "padding" here means the combination of deformation behaviour of the door and cushioning in the usual sense of the term.

Computational Occupant Simulation

Optimisation of the door trim and padding on the basis of an optimised vehicle structure from the structural computation should, however, first be realised with a model of a part of the complete vehicle structure (e.g. door and a few relevant components). By this means, the complexity of computation and modelling in the computational investigations can be greatly reduced and a large number of variants of the cushioning can be examined (parameter study).

Figure 22 shows a computational model with a door, a seat and a US-SID on a finite element basis. Exact modelling of the dummy with finite elements accurately reproduces the kinematics of the dummy. The influence of the door position on the occupant loading at the time of door/occupant contact can be investigated by this means. The computational model is then extended to account for a door trim so that the influence and the occupant loading might be analysed.

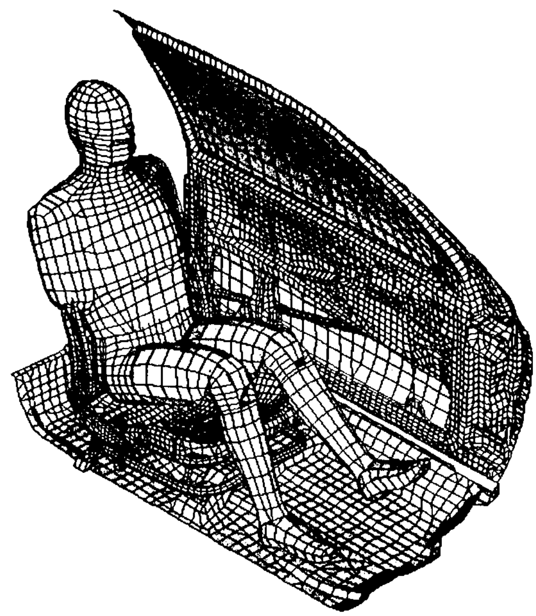

Figure 22: Parameter study for occupant simulation with a US-SID and a part of the vehicle structure

In this manner, padding foams with various properties and shapes are then studied further. The computation serves to predict the results of the subsequent tests and parameters studies in order to accelerate development work. The advantage of these investigations is the fact that the model is relatively simple and that different concepts can thus be evaluated more rapidly. The overall system comprising vehicle and occupant is then considered and harmonised as a whole in order to obtain information on occupant loading and vehicle deformation. This provides a calculation with an FEM model for the vehicle, the barrier and a validated FEM dummy made up of approximately 100,000 elements. Since the barrier, and in particular the link between the occupant, the vehicle interior and the structure play a significant role, accurate computation of this interaction requires that the material properties of the barriers and of the cushioning materials, and the dynamics of the occupants, can be described adequately and at a justifiable cost.

Component Tests

Individual components are investigated in component tests parallel to the computational activities. These components include bodyshell components such as door side impact beams, add-on parts and built-in parts such as door latch systems and hinges and parts incorporated in the restraint system such as padding and door trims. At this point only the dynamic testing of the latch systems and the dynamic padding investigations will be taken as an example.

In addition to the tearing-out tests carried out in accordance with FMVSS 206, additional dynamic drop tests are performed on the door latches and handles. The aim of these investigations is to ensure the acceleration resistance of the handle and the latch in the relevant spatial direction and thus prevent the doors from opening under the influence of inertia. The physical quantities are measured at the decisive points by means of measurements made on the latch system during the crash test. In the drop tests, the latch systems are first mounted in the associated door apertures and dropped onto special energy absorbers. Further to evaluation based on high-speed video recordings, the accelerations and displacements of individual components (handle, balance mass, actuating lever, etc.) are recorded. The parameters of drop height, weight of the latch carrier and stiffness of the energy absorber are derived from the measurement data from the vehicle tests. In order to model the maximum acceleration of 200 g over approximately 7 ms, Volkswagen uses, for example, aluminium honeycomb deformation bodies. These investigations are also used to validate computational simulations with the MADYMO multi-body system.

Various dynamic drop tests were performed to facilitate correct selection of padding /7/ in an comprehensive investigation into padding materials. The main emphasis was placed on comparison of polypropylene and polyurethane materials. It was shown that there is an optimum material thickness and density for a predetermined quantity of energy to be absorbed. This optimum is characterised by a low force level and thus by low occupant loading values. Figure 23 demonstrates this point. The maximum forces in drop tests on polyurethane padding are plotted as a function of their density and thickness. The force increase to left of the minima results from the sample foam parts bottoming out and the force increase to the right of the minima results from the higher deformation resistance as a result of the higher density. These sample parts are thus unsuitable for absorption of the corresponding quantity of energy at a low force level as the material thickness is not exploited.

In addition to the maximum force level, the efficiency of the energy absorber is also of decisive significance. If the efficiency is being determined the material thickness should be taken into consideration and not the maximum deformation as the foam parts can be crushed only until they fully bottom out and not until they are reduced to zero thickness.

The characteristics of the entire restraint system, that is to say the door trim, padding, door inner panel and any door fittings in the impact area are decisive factors for the design of the padding parts. The complete system is tested at Volkswagen on the dynamic door test stand, which is described below.

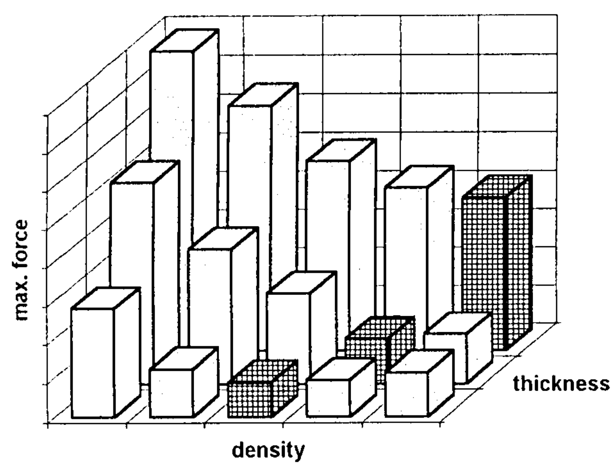

Figure 23: Maximum forces in drop tests on polyurethane as a function of foam density and thickness /7/

Door Test Stand On Horizontal Sled

Literature currently available describes various sled designs for the side impact simulation /8/. Problems in the reproduction of various test results occur with all sled systems published so far. Each sled is suited for only one test case (i. e. side airbag or padding development).

Volkswagen has developed a four-sled system which is suited equally to all development activities relevant to side impact. The design is conceived for installation on the acceleration sled system (Bendix) installed at Volkswagen. The design of the Side Impact Simulation Sled (SISS) is shown in Figure 24.

Three sled elements, which can be moved independently of one another, are arranged on the barrier sled where impacting bodies such as US or ECE deformation elements, a pole or a rigid barrier can be mounted. Two of these sleds serve for A- or B-pillar simulation and thus for mounting of the door. The connection to the pillars can, as in the vehicle, be effected by hinges and locks.

The driver's seat is mounted on the third, central sled where a side impact dummy is positioned. The retaining plate for the seat frame is mounted on a turnstile so that the seat direction of the dummy can be varied according to the sled axis or impact axis. As the pillar sleds can be moved independently of one another, the door can also be aligned obliquely with the barrier. In this way, with appropriate oblique positioning of the barriers, angular influences can also be investigated.

The seat position and the distance from the door trim are arranged as in the vehicle. The experimental sequence is analogous to that for the full-scale crash. The sled is fired and accelerated to the velocity v_0. The door is struck at v_0, deformed and impinges on the dummy positioned on the seat. Figure 25 shows the high correlation between the penetration velocities at the door in the full scale crash and those in the sled test as well as the occupant values.

In addition to the high level reproduction of occupant values, the SISS has the following advantages:

1. The door deformation is carried out in the sled test as in the full-scale test. The influence of the door characteristics and of the deformation on the occupants can thus be assessed directly.
2. The door penetration velocities during the crash can be simulated. The door structure with the installed parts (window lifts, door beams, latch systems, etc.) can be assessed directly as the loadings and accelerations are identical with those in the vehicle test.

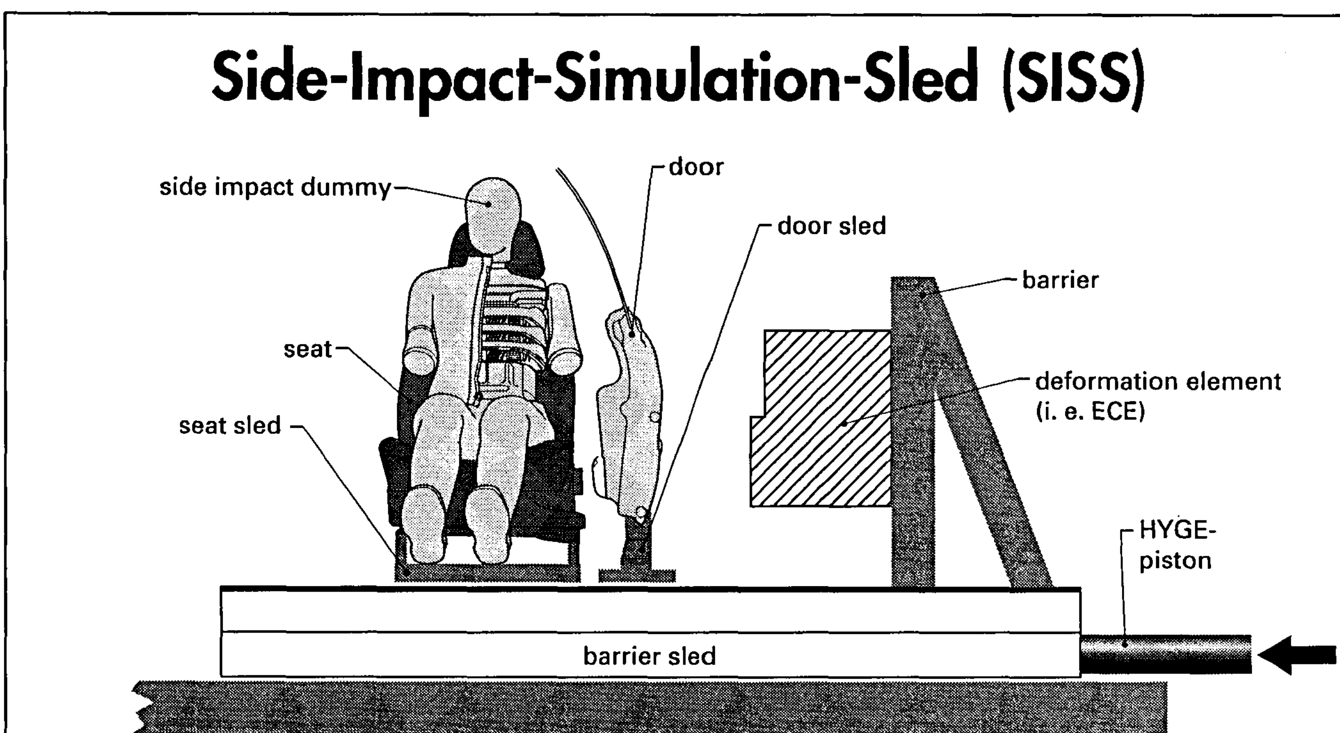

Figure 24: Fundamental design of the Side Impact Simulation Sled

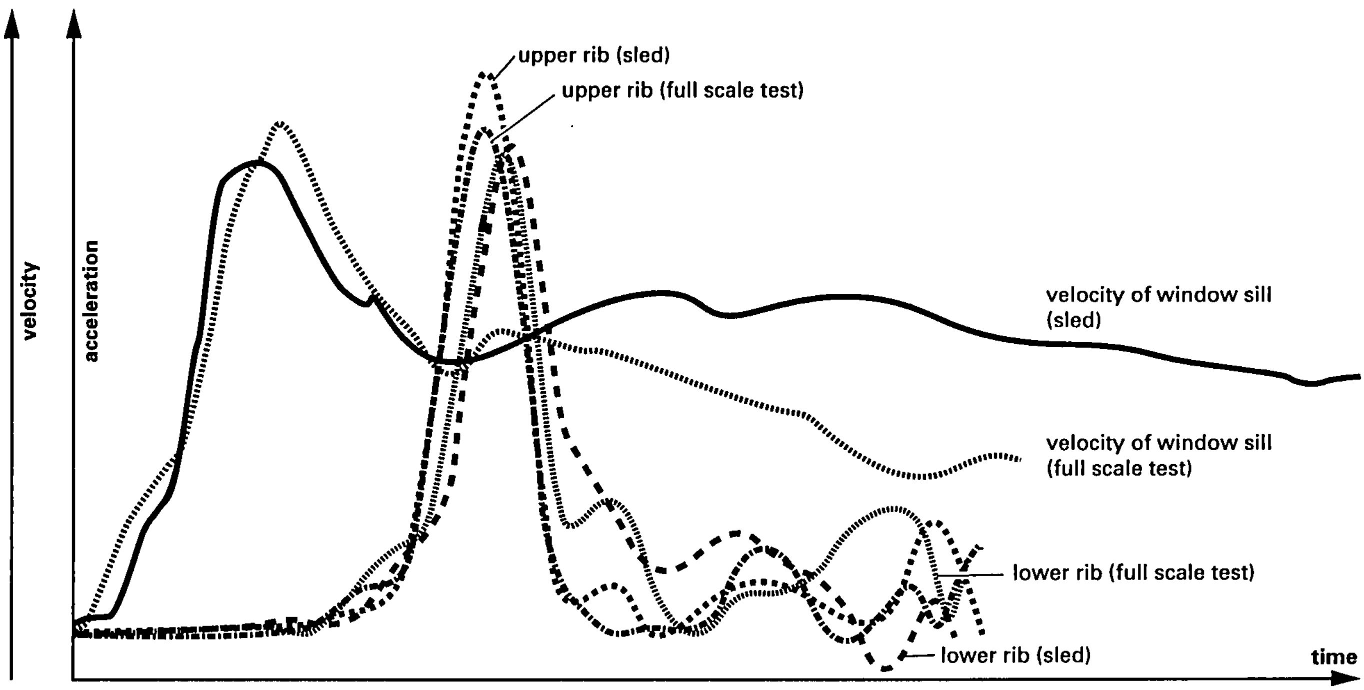

Figure 25: Comparison of the penetration velocities and the thorax accelerations in FMVSS 214 and sled test

DEVELOPMENT METHODOLOGY AT VOLKSWAGEN

A decisive requirement for the best possible planning, organisation and control of vehicle development is, alongside adherence to deadlines, the early integration and, at the same time, cooperation of all specialist areas. In addition to continuous pre-development work, the project-related design work begins with concept development, which starts when the project is initiated. Upon establishment of the catalogue of objectives, concept verification begins with the first testing processes. The catalogue of requirements produced in accordance with these results initiates production-vehicle development.

For the safety development engineers, the development sequence for a new vehicle project begins with the FEM simulation of the bodyshell concept which takes experience gained with the preceding model as well as the most up to date findings of materials and manufacturing engineering into account. The subsequent structural computation (side impact carried out in accordance with FMVSS 214 and ECE-R 95) gives an initial indication of the crash performance of the concept selected. At the same time, a suitable testing method and corresponding computational models are developed for new, side impact-relevant components. These components include, for example, new types of padding materials, complete door or side trims, door side impact beams, latch systems, catch systems, etc. If the assessment is positive it is followed by the modelling of a first concept vehicle. In the crash simulations, an FEM dummy is set up in the structure for the first time. Following positive assessment, the first full-scale test is performed. The findings from this test are taken into account both in the computational model and in the first prototype. Further optimisation is then carried out on individual components. Thus, the entire restraint system (door trim and padding) is optimised in tests on the SISS dynamic door test stand. The influences of different seat positions can also be visualised here. After these results are incorporated in the computational model of the first prototype, further crash simulations are performed. If positive, the prototype will be completed and evaluated in a full-scale crash test. After a further optimisation loop, the release test then takes place. Figures 26 and 27 summarise these side impact-relevant activities in the development process.

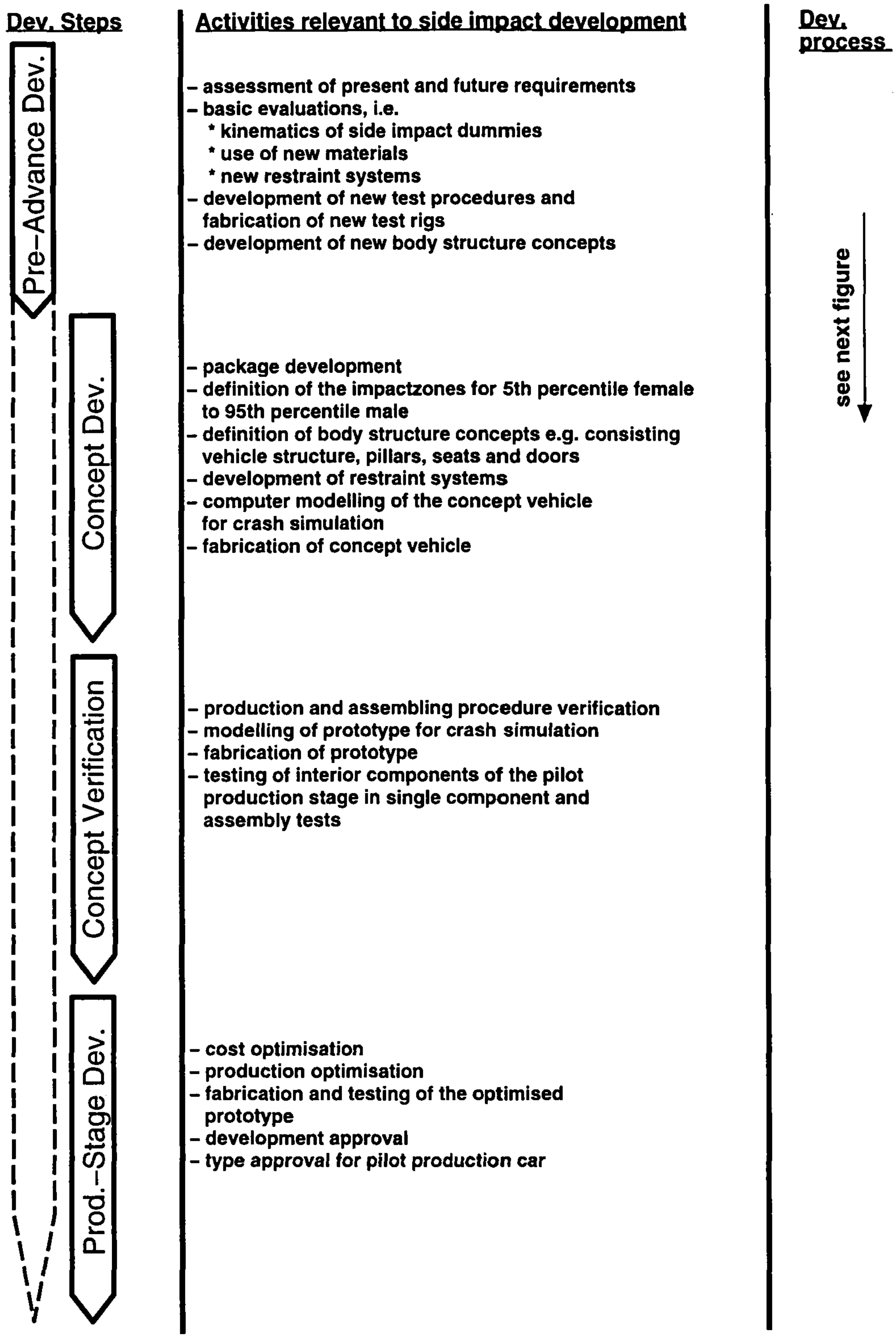

Figure 26 a): Side-impact relevant activities in the development process

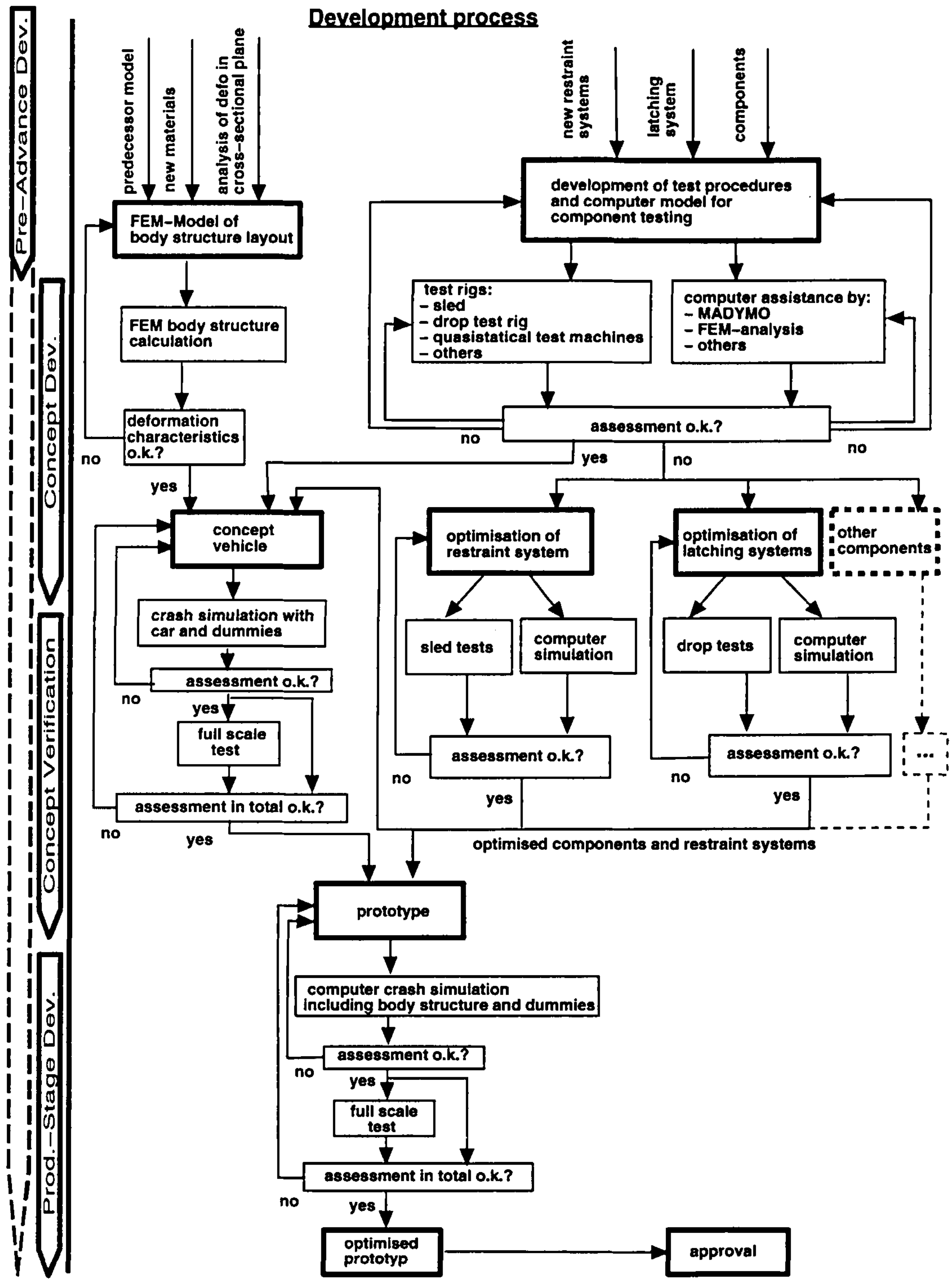

Figure 26 b): Side-impact relevant activities in the development process

SUMMARY AND OUTLOOK

In this paper, a short discussion of the significance of the side impact in real accidents is followed by a description of the requirements to be met by a vehicle designed to withstand a side impact. It becomes apparent that the requirements of the US side impact test procedure and those of the European side impact test procedure do not necessarily supplement each other. Whilst measures for increasing lateral stiffness and thus also for reducing the penetration velocity mean that the requirements of FMVSS 214 are met, the side impact carried out in accordance with ECE-R 95 requires that the occupant impact kinematics follow a predetermined scheme. Whilst harmonised padding measures at the thorax height lead to a significant reduction in thoracic loading in US-SID, they are not taken into account with EuroSID-1. Compliance with a maximum survival space which, in a real accident, reduces crushing injuries and facilitates removal of passengers is not taken into account by EuroSID or US-SID.

In the further course of the paper, development tools and methods are described which Volkswagen AG uses to achieve its objective of ensuring comprehensive side protection in its vehicles, taking additional requirements into account. These methods include use of FEM models with integral FEM occupants, as well as classical test methods.

The conflicting objectives observed in day-to-day work concerning vehicle design carried out in accordance with FMVSS 214 and ECE-R 95 must not lead to reduced protection afforded to occupants in real accidents. For this reason, more extensive biomechanical research and incorporation of the results in the further development of side impact dummies is necessary. It is equally important that shape and stiffness of the deformation elements are continuously matched to real-life accidents.

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