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BASIC RESEARCH IN CRASHWORTHINESS II — STEERING COLUMN INVESTIGATION

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16 Abstract

The role of the steering assembly as an element of a driver restraint system in a crashworthy automobile is studied. "Piggyback" tests were made on instrumented vehicle impacts committed to the structural studies in the program "Basic Research in Crashworthiness". Laboratory tests were made and advantage was taken of data in the literature to define an array of elements that could enhance driver survivability. These include the steering column jacket energy absorber, air cushion, hydraulic strut, knee bar, steering hub pad, and an energy absorbing four bar linkage.

This study is appended to the more basic effort to develop crashworthy structures that assure compartment integrity - a result that must be obtained before driver restraint systems can be of value in high speed impacts.

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FOREWORD

This interim technical report presents the results of a study of automobile steering columns. It is an addendum to the primary structural emphasis of the Basic Research in Crashworthiness II program - the structural crash tests affording an excellent opportunity to obtain precursory information on the steering assembly. This report is submitted in partial fulfillment of a program of research conducted by Calspan Corporation for the National Highway Traffic Safety Administration under Contract No. FH-11-7622. Contract Technical Manager for this project is Mr. Glen Brammeier of NHTSA.

The opinions and findings expressed in this publication are those of the author and not necessarily those of the National Highway Traffic Safety Administration.

This report has been reviewed and is approved by:

Raymond R. McHenry

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Assistant Head
Transportation Safety Department

PREFACE

This report is one of a series of interim reports describing the research efforts being conducted under the "Basic Research in Crashworthiness II" program. These interim reports represent the effort and conclusions in a particular area associated with the overall crashworthiness program. While the conclusions may not be final and the data presented may not be all of the data to be digested in the particular area being reported, it is felt that these reports may be useful in:

(1) providing a focus for the objectives of the overall program, and

(2) providing a convenient means for presentation of the program results in a measured fashion rather than in a voluminous final report.

It is anticipated that the final summary report will fully integrate these interim reports with the overall program objectives and conclusions. In the meantime, a brief discussion of the program objectives and the overall scope of the effort is presented to provide perspective for these interim reports.

The objective of this research program is to obtain experimental and analytical data that will establish the feasibility of designing automobile structures to dissipate energy at a controlled rate during collisions and prevent intrusion into the passenger compartment. The problem has been attacked from three separate and initially distinct aspects, namely,

- to determine structural performance characteristics of current automobiles through a wide range of test conditions, i.e., vehicle size, impact speed, impact conditions, etc. (so-called base line tests).

- to consider structural modifications required of the front of a vehicle to improve its structural performance in frontal impacts.

- to consider structural modifications required of the side elements (i.e., doors, door pillars, roof) of the vehicle to improve its structural performance in side impacts.

Each of these areas has involved a concern for the performance of unmodified and modified vehicles in impact situations with

- fixed pole and flat barriers, and

- in vehicle to vehicle impacts, including impacts between two unmodified vehicles, two fully modified vehicles and one unmodified and one modified vehicle.

The effort also encompasses a concern for the impact behavior between vehicles of

- different weight classifications,

- different structural characteristics, i.e., frame-compartment type vs. unitized type, and

- different impact conditions, namely compatibility in front/side impacts.

The modifications are not restricted to the vehicle exterior. A task concerned with desirable interior modifications, e.g., steering columns, piggyback testing of advanced restraint systems, interior modifications (padding, seats, etc.) is also a part of the program. A natural outgrowth of the overall concern for improving the crashworthiness of the vehicle is the concern for the production feasibility of the recommended structural design modifications made for full size automobiles. A separate task addresses itself to this aspect of the crashworthiness program.

Finally, a substantial analytical treatment of the problem of a frame-compartment type of automobile impacting a rigid obstacle is under development. The objective of the effort is to allow prediction and hence evaluation of the behavior of frame type structures.

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SUMMARY

Results of a study of automobile steering assemblies are reported herein. Primary emphasis is placed on the driver protection aspects - in keeping with the overall objective of increasing car crashworthiness.

Specific design goals inspected for the steering column task were:

a) Load/deflection curve to provide suitable energy absorption up to a barrier impact speed of 25 mph with a torso (body block) force less than 2500 pounds.

b) Investigate desirability and feasibility of elimination of movement of steering assembly into occupant compartment during barrier collisions at speeds greater than 40 mph.

c) Provide vehicle controllability comparable to that of current production assemblies.

Interpretation of studies made were to include:

d) Recommended performance requirements.

e) Recommended demonstration procedures which might be used to determine compliance.

The above task items essentially deal with the steering assembly as a low speed restraint. Another part of the Basic Research in Crashworthiness II study was devoted to advanced restraint systems. This effort also was ancillary and tests were piggybacked on the primary structural tests. Included were tests of driver air cushions - the dominant item of interest being deployment time.

A combination of elements is advanced which addresses the identified driver protection problem areas. These elements are:

1. Steering jacket absorber, steering wheel

2. Air cushion

3. Intruder/absorber

4. Four bar linkage

5. Knee bar

6. Hub pad

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1. INTRODUCTION

Reported herein are the results of a research task conducted by Calspan Corporation (formerly Cornell Aeronautical Laboratory, Inc.) for the National Highway Traffic Safety Administration as part of an effort to improve the crashworthiness of cars - the specific task being associated with the steering assembly as it influences the driver's survivability.

The major thrust of the Basic Research in Crashworthiness II program is the maintenance of occupant compartment integrity (up to speeds of 60 mph) by dissipation of energy in the remainder of the structure. Maintenance of the occupant compartment integrity is the indispensable requirement for occupant survival. There are, however, non-structural aspects of crashworthiness which could be inspected expeditiously and economically in the crash environment necessary for testing the vehicle structural performance. Steering assembly and advanced occupant restraint systems (air cushions) are amongst the auxiliary piggyback tests performed.

The purposes of these appendix efforts were:

- to gauge the problem of driver survivability
- to observe if there are special difficulties with respect to the driver, related to crash energy management structures
- to bring to the surface any incidental items, not present in sled or pendulum tests, that might defeat occupant protection efforts
- to develop data, under known impact conditions, that can be recalled for subsequent inputs to analyses

- to evaluate first generation design elements that would contribute to driver survivability*

- to indicate the performance of air cushion elements that influence deployment

- to provide an indication of performance of air cushions in impacts substantially more severe than the current FMVSS 208 design conditions.

The ubiquitous steering column and driver are elements in all passenger car vehicles and, hence, it is appropriate to emphasize the effect of the assembly on occupant response in a crash.

It is noted here that data obtained from accident investigations are inconclusive with respect to the effect of energy absorbing steering assemblies (as they are installed in production automobiles - Reference 1).

In Reference 1 it is reported that no statistically significant differences can be found between vehicles manufactured between 1960 and 1965 and vehicles manufactured between 1968 and 1971 - the "old" vehicle category having no energy absorbing steering column and the newer category all having steering column or column jacket energy absorbers. Such data suggests that there are factors showing up in vehicle accidents that were not accounted for in FMVSS 203. Similar results had been mentioned by other accident investigators and a basis for this apparent lack of effectiveness was sought. References 2 and 3 suggested sliding friction between telescoping parts as a possible reason for the inability of the energy absorbers to function under oblique impact conditions. Data developed in the current effort support this view.

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* Vehicle occupancy in the U. S. averages less than two persons. Therefore, the driver position represents half the population to be protected.

Initial effort in the steering column task was devoted to a literature search which appears here in Appendix 2 (reported as Attachment 2 to the Eleventh Progress Report on Basic Research in Crashworthiness II). Performance items were identified in that search and augmented during the Crashworthiness II effort. Figure 1 shows the major performance items and the elements postulated to satisfy the performance.

A brief discussion of the effects of the vehicle layout and crash pulse is presented in Section 2. Section 3 contains the results of baseline tests designed to illustrate impact angularity effects and data obtained from the literature search to illustrate column force waveforms and knee bar characteristics. Emphasis is on the low speed characteristics in Section 3. Section 4 is devoted to a discussion of the driver air cushion with emphasis on deployment times observed in the Crashworthiness II piggy-back tests. A hydraulic intruder/absorber is described in Section 5 - a device acting in conjunction with the EA column and air cushion and intended to enhance driver survivability in high speed frontal impacts. Conclusions and recommendations are summarized in Section 6. Appendices 1 and 3 contain further information on the four bar linkage energy absorber and the intruder/absorber.

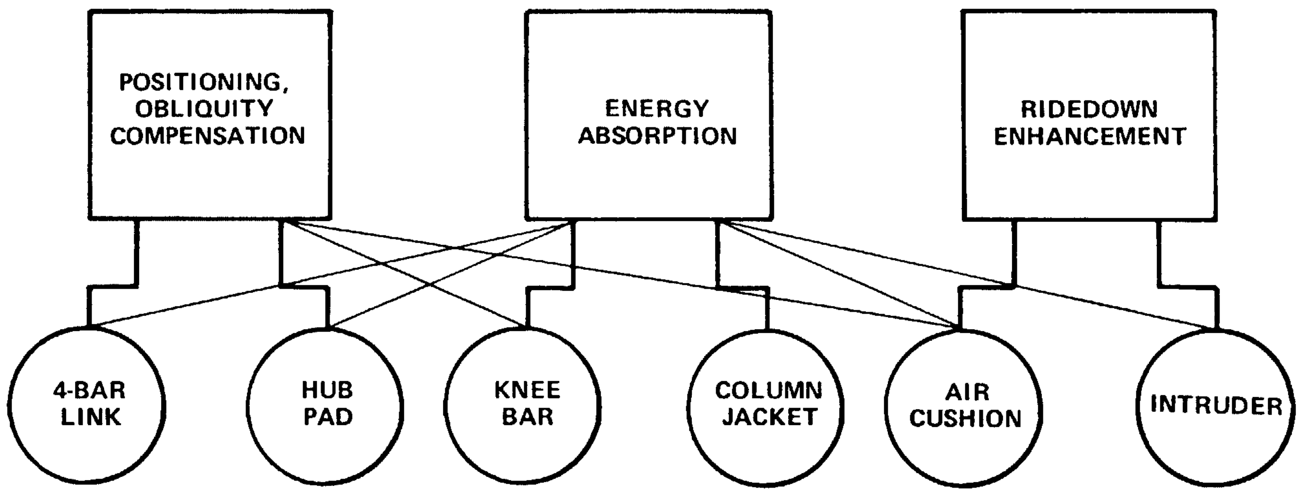


Figure 1 RELATIONSHIP OF DRIVER RESTRAINT ELEMENT TO SYSTEM PERFORMANCE ITEMS

2. EFFECTS OF VEHICLE LAYOUT AND ACCELERATION PULSE

2.1 Peripheral Elements

Elements of vehicle layout that directly influence the ability of the steering assembly to provide driver restraint are the steering gearbox and the driver's seat. The steering gearbox moves aft relative to the compartment as the frame deforms and the following adverse effects can result:

- collapse (before the driver torso impact) of the energy absorbing steering jacket due to crushing against the upper jacket support

- rotation of the steering assembly about the upper support point such that the torso impacts the edge of the steering wheel and axial motion of the energy absorbing jacket is prevented.

A possible cure for the above adverse effects is provision for a disengagement of the steering shaft from the gearbox when the structure moves the gearbox aft. One technique for accomplishing this by offsetting the steering shaft and providing a breakaway gear support is illustrated in Figure 2. Use of a flexible shaft (Figure 3) to permit relative motion between the gearbox and steering column jacket is another possibility. In both of these cases, the offset was introduced to reduce the longitudinal steering column angle for purposes to be discussed in Section 3.3. Other solutions - such as use of a rack and pinion - would be expected in production vehicles.

Seats and seatbacks constitute a second class of items that could defeat occupant restraint systems. The energy of a seat that tears loose at the track or anchors is usually transmitted to the structure through the occupants. Breaking of the latch on a bucket seat or the

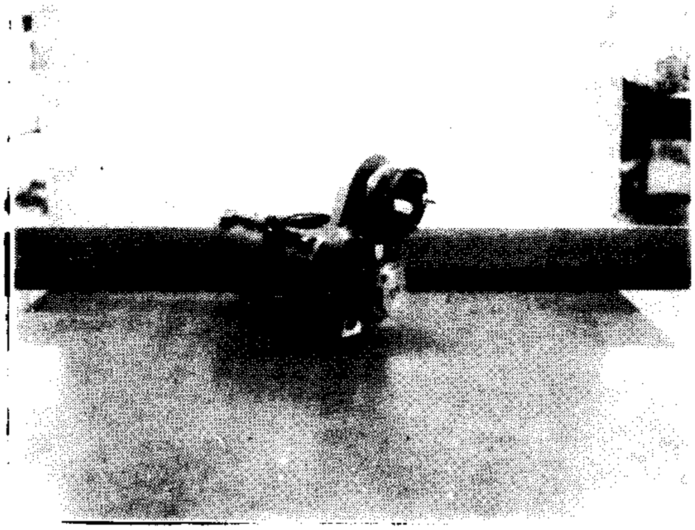


Figure 2 GEAR TRAIN (MOUNTED ON STEERING GEAR BOX) TO ACCOMMODATE STEERING SHAFT OFFSET

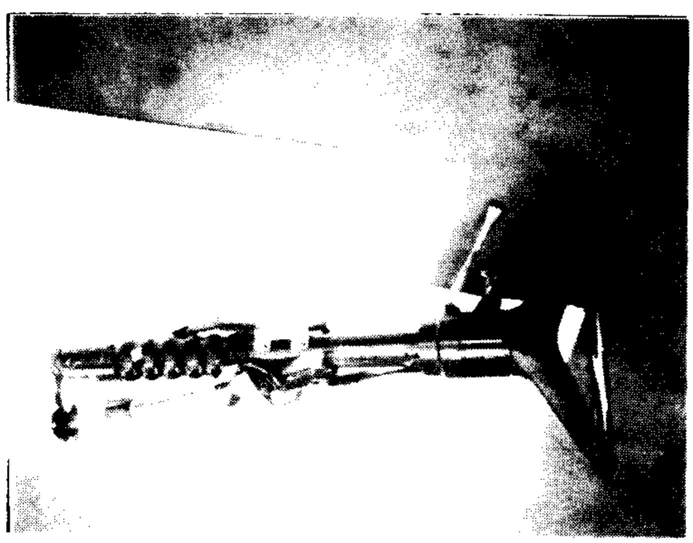


Figure 3 FLEXIBLE SHAFT TO ACCOMMODATE STEERING SHAFT OFFSET

seatback on a bench seat provides an increment of inertia to the occupant torso and/or head that must be decelerated.

Alleviation of the above peripheral problems could be accomplished, e.g., by

- providing for disengagement of the steering shaft from the steering gearbox

- providing strengthened seats and seat latches

2.2 Effect of Vehicle Pulse Characteristics

Questions about the effect of vehicle crash pulse characteristics on occupant response have been treated in the literature, e.g., in References 4-7. One of the main results derived from the analyses is the possible reduction of occupant loads by "ride down" wherein he is in contact with the vehicle interior before its velocity reaches zero.

For a vehicle having a constant acceleration crush characteristic, A_v, it can be shown (Reference 4) that a ride down benefit can be obtained only when the occupant separation distance is less than the vehicle crush distance. Driver separation distance, b, is the distance from his chest to the steering wheel and Figure 4 shows how it is related to the minimum speed at which a benefit is obtained. The case indicated is one in which the driver chest is 10-1/2" from the steering hub, and three vehicle acceleration levels are shown. No ride down is obtained for the following conditions:

$A_v = 20$ g, $V < 33$ ft/sec.
$A_v = 40$ g, $V < 47$ ft/sec.
$A_v = 60$ g, $V < 58$ ft/sec.

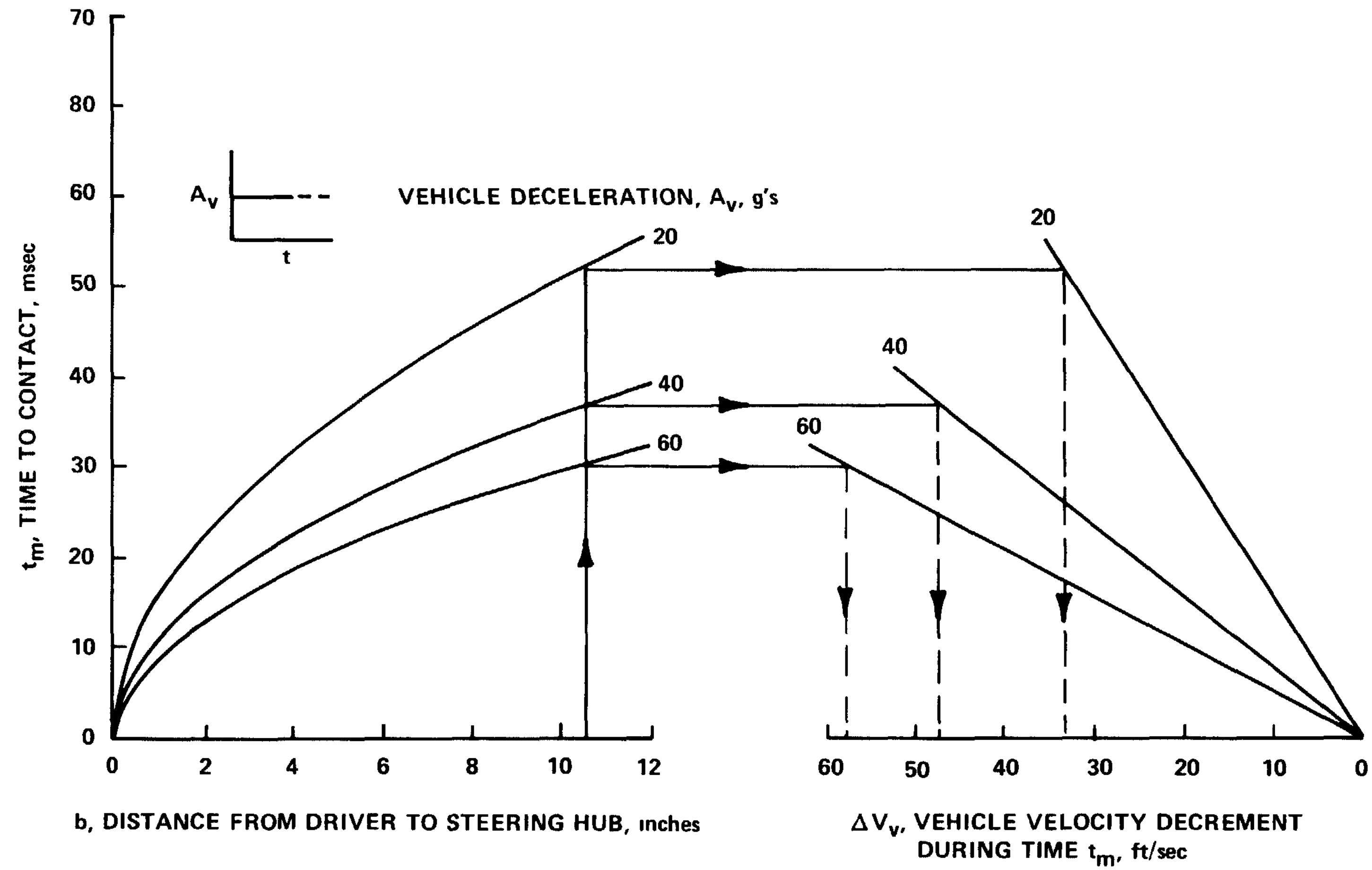


Figure 4 TIME OF CONTACT, t_m, AND VELOCITY DIFFERENTIAL AT TIME t_m BETWEEN UNRESTRAINED DRIVER AND VEHICLE INTERIOR

A finite onset rate preceding the constant vehicle acceleration increases the velocity range in which no benefit is available. Increasing car crush acceleration (i.e., higher vehicle stiffness) also increases the velocity range in which no ride down is available.

In summary, vehicles having nearly constant acceleration crash responses, finite rise times and acceleration levels greater than 20 g's offer little or no ride down for drivers at velocities less than about 20 mph. The vehicle interior - i.e., the steering assembly - is stationary or rebounding and the occupant impacts the assembly with at least the initial velocity of the vehicle. All the occupant energy must be dissipated by the interior of the vehicle (principally the steering assembly) and/or the occupant in the absence of a restraint system. Energy absorbing (EA) columns are discussed in Section 3 and Appendix 2.

3. ENERGY ABSORBING (EA) STEERING ASSEMBLIES AND KNEE BARS

3.1 Background

The steering column was identified as a major contributor to driver injury - in the sense of being the object struck by the injured driver - many years ago (e.g., Reference 8). Introduction of the energy absorbing column jacket and performance requirements on intrusion appeared to moderate this problem - see Figure 5 taken from Reference 9. Apparently the authors of Reference 9 and Reference 1 (alluded to in the Introduction, Section 1) are not in agreement on this point. Data gathered by GM on their 1968 columns suggest that the potential performance of the EA columns was not being realized - Figure 6 taken from Reference 10.

Figure 6 shows clearly that the EA columns were being influenced by factors besides speed of impact. Static column crush results are presented in Section 3.2 and body block impact tests to illustrate angularity effects are presented in Section 3.3 and the data show a strong dependence on angle of contact.

Section 3.4 is a discussion of a four bar linkage mechanism intended to reduce angularity effects by permitting lateral motion of the steering wheel; and Section 3.5 deals with extrusion as an EA process with (possibly) reduced angularity effects. Knee bar, as energy absorbers, are discussed in Section 3.6.

3.2 Static Crush Test - 1968 Column

A static crush test was performed to determine the characteristics of the force/deflection curve of a 1968 Ford EA column.

Results of the static load/deflection test are shown in Figure 7. The major energy absorbing portion of the assembly is the slotted,

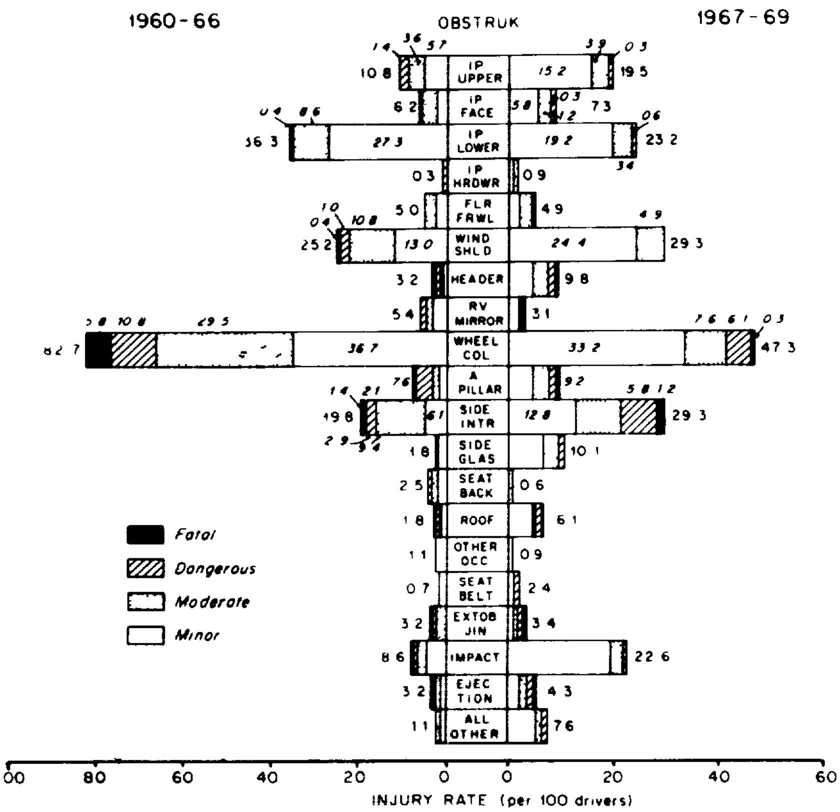


Figure 5 DRIVER INJURY RATES FOR OBJECT STRUCK AND INJURY SEVERITY (FROM REF. 9)

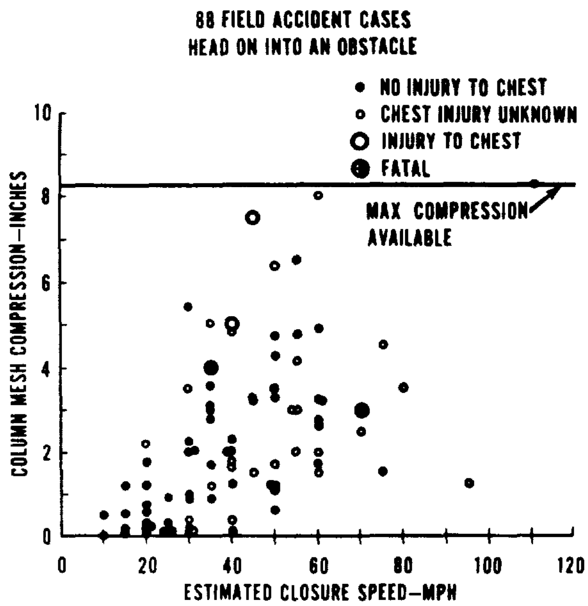
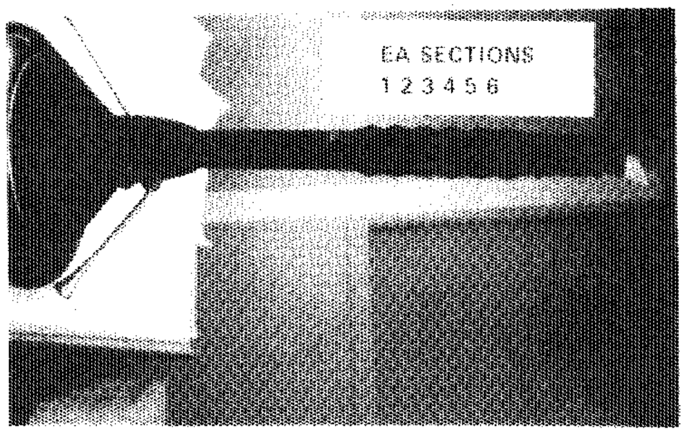


Figure 6 ENERGY ABSORBING STEERING COLUMN COMPRESSION [From Reference 10]



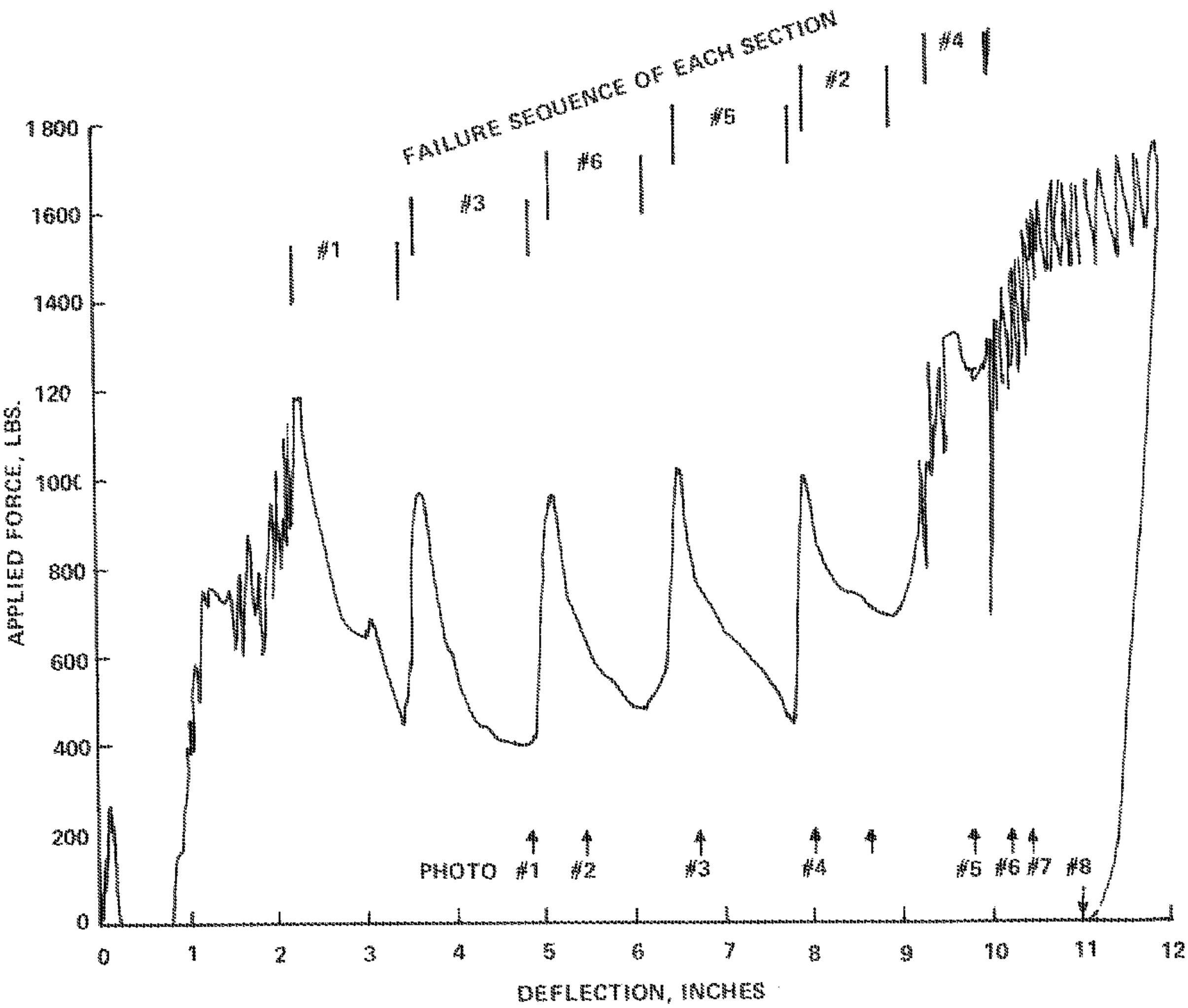


Figure 7 STATIC TEST RESULTS FOR 1968 FORD ENERGY ABSORBING STEERING COLUMN

convoluted section of the jacket. Energy absorbing sections are identified in the photograph at the top of the Figure 7. Section 1 is closest to the steering wheel. The energy absorbed in the test shown was about 10,000 inch-lbs. Development of the collapse is shown in Figures 8 and 9 and the photo numbers correspond to the loading conditions indicated along the abscissa of Figure 7. Failure sequence of the EA sections is noted along the top of the data curve in Figure 7, the section sequence being 1, 3, 6, 5, 2, 4. The small region of the curve at small deflections (less than 3/4") in Figure 7 appears to correspond to the shearing* of the plastic buttons in the steering column and the subsequent telescoping under a load which is nearly zero.

The measured static deformation curve agrees qualitatively with convoluted mesh characteristics (e.g., see Figure 17 in Reference 2).

3.3 Effect of Angular Impacts on EA Steering Assembly Performance (Body Block Tests)

It has been suggested (References 2 and 3) that imposition of bending moments on the steering shaft caused an increased friction between telescoping elements during collapse and, hence, higher force levels and reduced stroke. A steering column support mechanism was built and tests performed to check the postulate.

The type of column was the same as that used in the static tests. Dimensions characteristic of 1968 Fords were maintained by using Ford upper attachment points (shear capsules) and lower support points. A standard body block on a trapeze pendulum furnished the impacting force.

The range of conditions and the jacket collapse distances are

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* These buttons were probably partly sheared before the start of static tests.

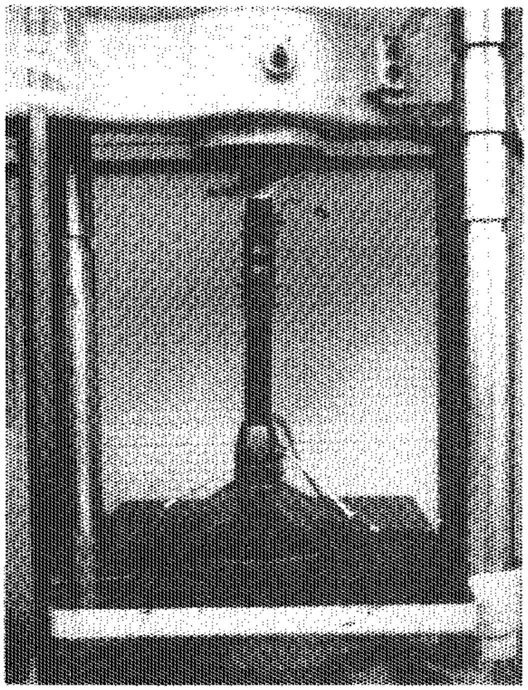


PHOTO 1

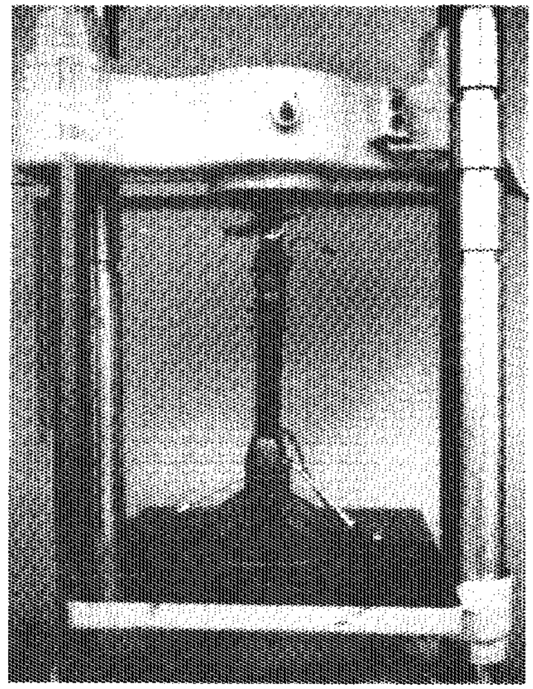


PHOTO 2

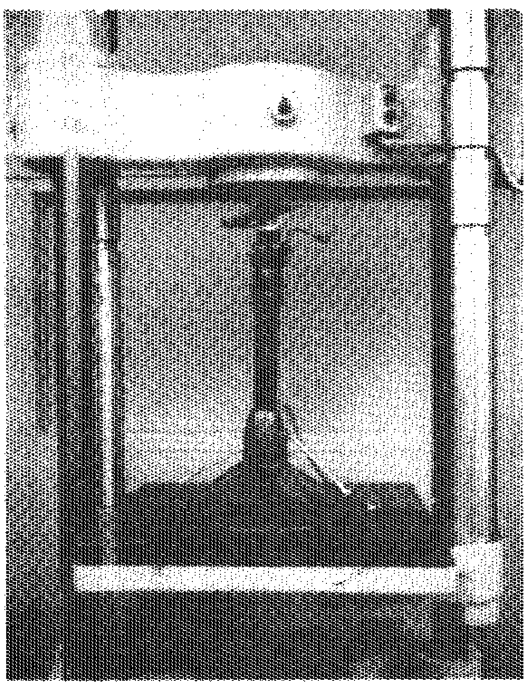


PHOTO 3

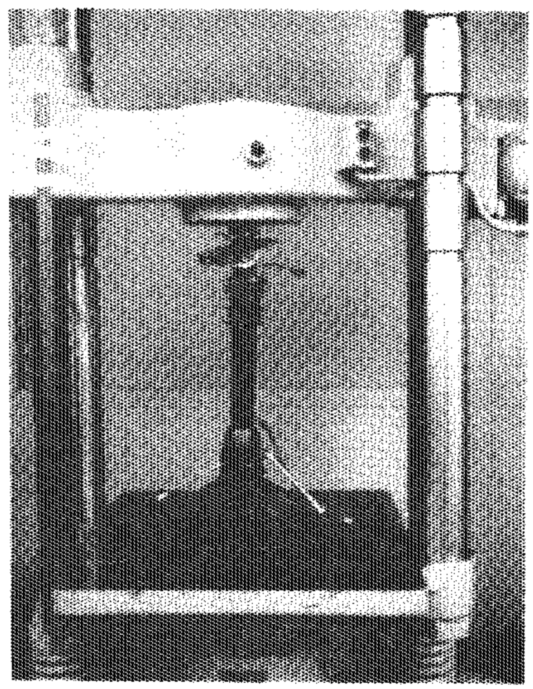


PHOTO 4

Figure 8 DEVELOPMENT OF COLLAPSE OF EA STEERING COLUMN

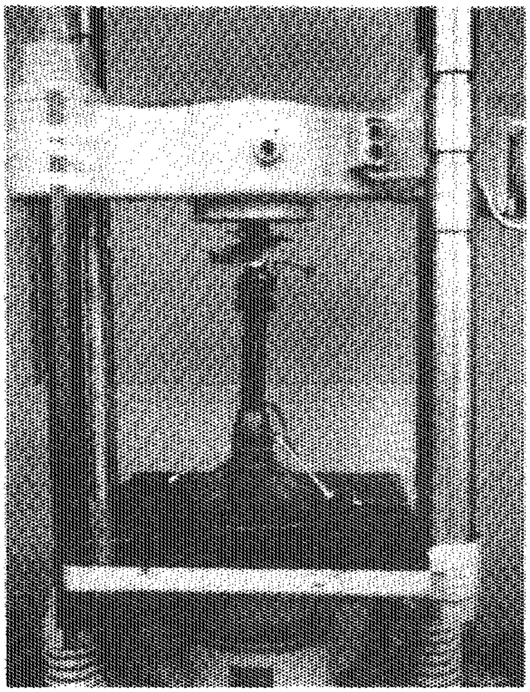


PHOTO 5

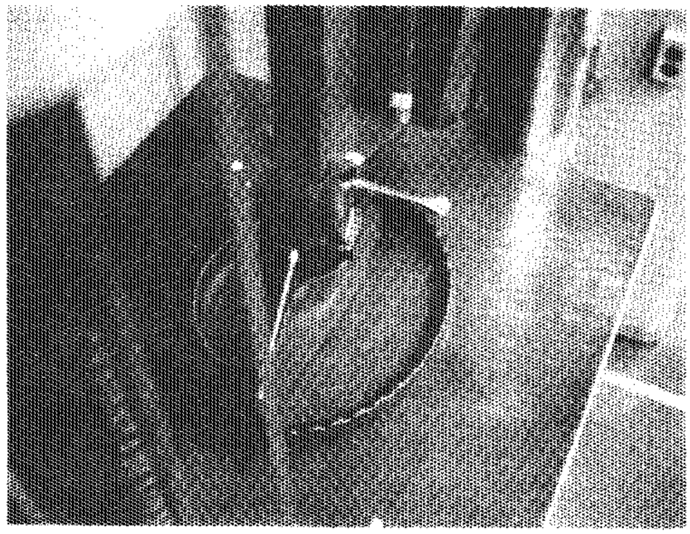


PHOTO 6

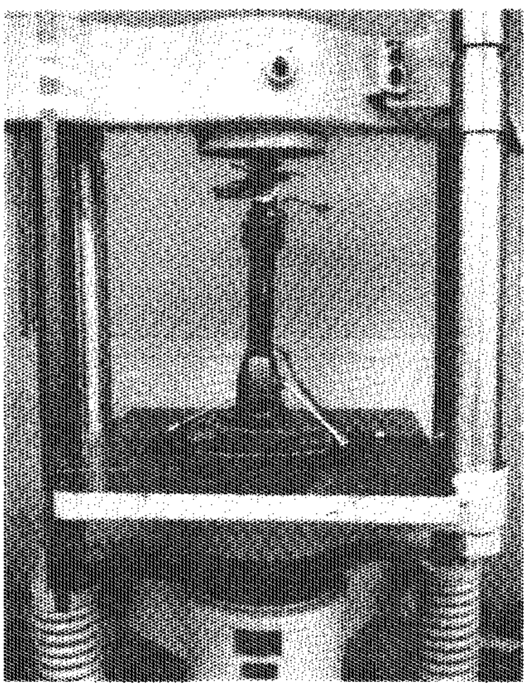


PHOTO 7

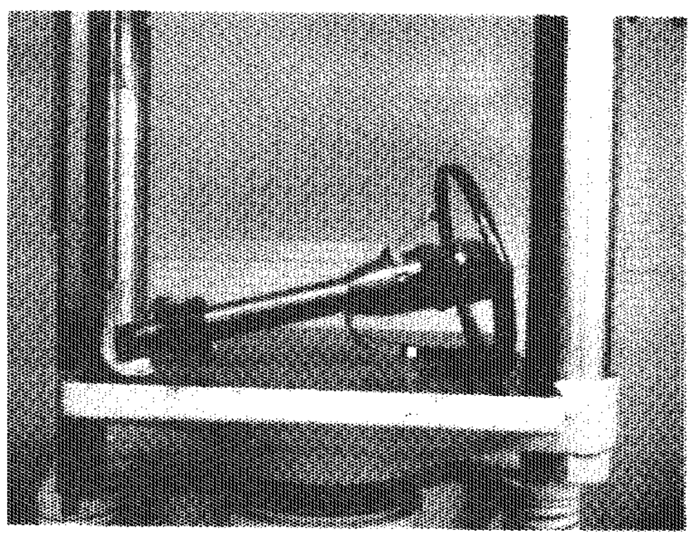


PHOTO 8

Figure 9 DEVELOPMENT OF COLLAPSE OF EA STEERING COLUMN given in Figure 10. Also shown in Figure 10 is a line sketch of the arrangement. Nominal speed of impact was 18 MPH.

The data table in Figure 10 shows that for small angularity of impact (0°, 0° and 0°, 15°), a substantial stroke, 5", was obtained. In all cases for which the longitudinal angle was 26° the collapse stroke was substantially reduced. Further, there is a dependence on steering wheel chord spoke location (horizontal or vertical).

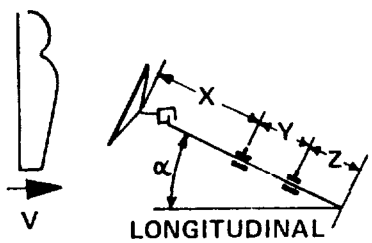
The simple bench tests described above confirm the hypothesis that at least a portion of the scatter of column collapse distances observed in Figure 6 is due to obliquity of impact. Similar observations with respect to the G.M. ball-type EA column are reported in Reference 11. It will be seen in Section 4 that column angularity is also a consideration with respect to air cushion deployment.

A graph presented in Reference 12 is shown as Figure 11. The original graphical presentation showed only the driver air bag data and the data on the unrestrained anthropomorphic devices (solid symbols) appeared only in a tabulation. Acceptable values of head severity index (SI = $\int_0^T a^{2.5}\, dt$) based on the resultant acceleration, a, are indicated for the simulated drivers for speeds below about 30 MPH. However, it is reported in Reference 12 that special pains were taken to reduce column frictional effects and the data shown were obtained at zero offset and zero angularity. Further, a substantial part of the total load was taken out through the knees into an EA dash panel.

3.4 Four Bar Linkage

Angularity effects were indicated in Section 3.3 to be important with respect to occupant arrest. Bending moments applied to the steering assembly apparently are reacted as couples, and the accompanying normal forces generate friction forces in telescoping elements. These

SUPPORT AND OVERHANG DIMENSIONS



X INCHES	Y INCHES	Z INCHES	FIXITY	
			FREE	*
**	**	*		✓

* RESTRAINED BY LOAD CELL AT STEERING SHAFT COUPLING

** STANDARD 1968 FORD DIMENSIONS

RUN IDENTIFICATION	LONGITUDINAL DEGREES	LATERAL DEGREES	ESTIMATED VELOCITY FT/SEC	STEERING WHEEL YOKE ALIGNMENT	JACKET COLLAPSE DISTANCE INCHES
BLT 1	0	0	26	HORIZONTAL	5.0
BLT-2	0	15	26	HORIZONTAL	5.5
BLT 3	26	15	25	HORIZONTAL	0 6
BLT 4	26	0	27	HORIZONTAL	3.2
BLT-5	26	0	26	VERTICAL	1.3

Figure 10 BENCH TEST OF THE INFLUENCE OF IMPACT ANGULARITY ON STEERING COLUMN EA EFFECTIVENESS

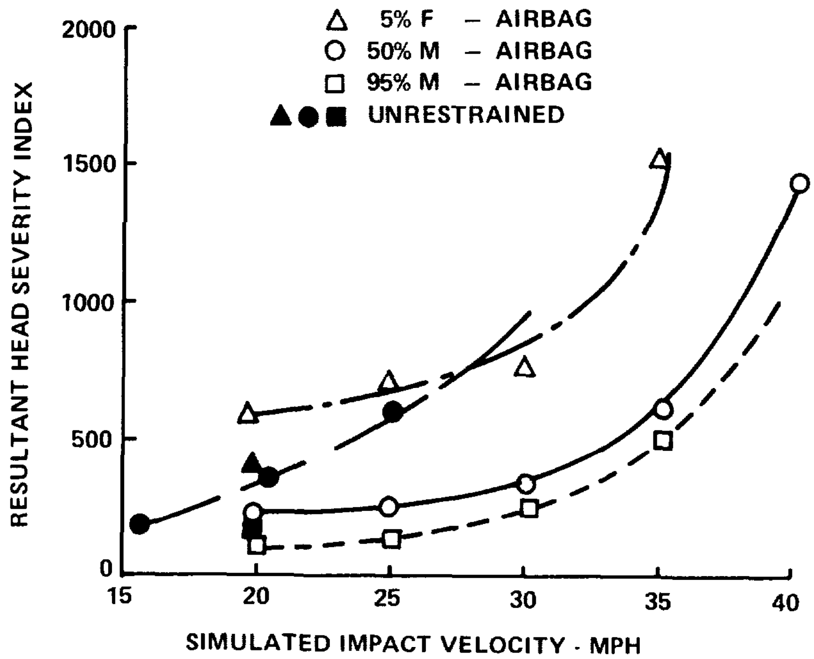


Figure 11 SI vs. VEHICLE VELOCITY FOR 50th PERCENTILE AND 95th PERCENTILE MALE AND 5th PERCENTILE FEMALE DUMMIES (REFERENCE 12)

frictional forces add to the EA axial loads and, hence, change the force level required to crush the column. In some cases oblique impacts probably cause breakdown of the steering wheel rim, deflection of the torso away from the steering axis, and consequently defeat of the EA jacket. Left front impacts, in particular, could produce driver contact with the A pillar.

Means for accommodating oblique driver impacts were inspected and it was concluded that a four bar linkage pinned at the firewall and at the instrument panel location could perform the desired functions:

- dissipation of energy corresponding to lateral velocity component.

- maintenance of steering wheel plane position parallel to original plane.

A theoretical analysis was made, single hinge tests were conducted to obtain design data, and a laboratory mock-up was built and tested. Energy dissipation was accomplished by metal cutting washers at the hinge points. Appendix 1 contains the analysis and component test results.

Promise is shown by the linkage but it must be integrated with and tuned to all other elements of the restraint system. Reduced sensitivity of the axial energy absorber to angular impacts would permit more latitude in the selection of four bar linkage characteristics. A possible axial absorber is discussed in Section 3.5.

3.5 Axial Energy Absorber - Extrusion Process

Another possible approach to alleviating the effects of moments applied to the steering assembly EA unit is to reduce the sliding friction by material selection. This would have the effect of keeping the axial collapse force more nearly independent of angularity of impact. Lubrication was the solution pursued in Reference 3. Changing the material and the process was the line of attack in the current effort.

Teflon rod was selected as the material and extrusion was the EA process chosen in the current study. It was reasoned that the low coefficient of friction would reduce the axial force component arising from sliding contacts. (Use of a Teflon coating on a metal tube was considered but was not tried.) A laboratory experiment was set-up to inspect performance.

Energy absorption of a Teflon rod during impact extrusion was measured on the drop test facility. Figure 12 shows the apparatus. The drop weight is raised to the height necessary to obtain the desired impact velocity. A plunger with rounded top drives the specimen through an area-reducing orifice. Plunger, guide tube, and orifice block are mounted on a plate which is supported, in turn, by three load cells.

Exploratory tests were made to determine effects of orifice area ratio, die chamfer and preseating of the specimen in the die, and elimination of the guide tube. Chamfer and preseating reduced the initial load peak and, as expected, the die area ratio controlled the steady-state force level. Removal of the guide tube and plunger (i.e., direct impact of the drop weight on the specimen) had relatively little effect. It is believed that the small effect of the guide tube friction is due to the low frictional coefficient of Teflon.

Load cell readings and the constructed sum of these readings for an impact velocity of about 20 MPH is shown in Figure 13. The peak

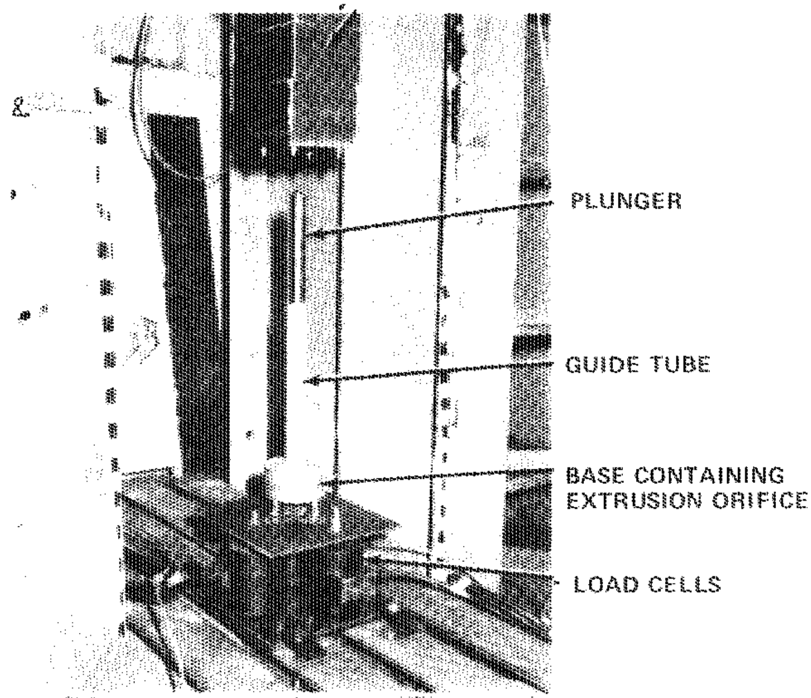


Figure 12 APPARATUS FOR IMPACT EXTRUSION OF RODS

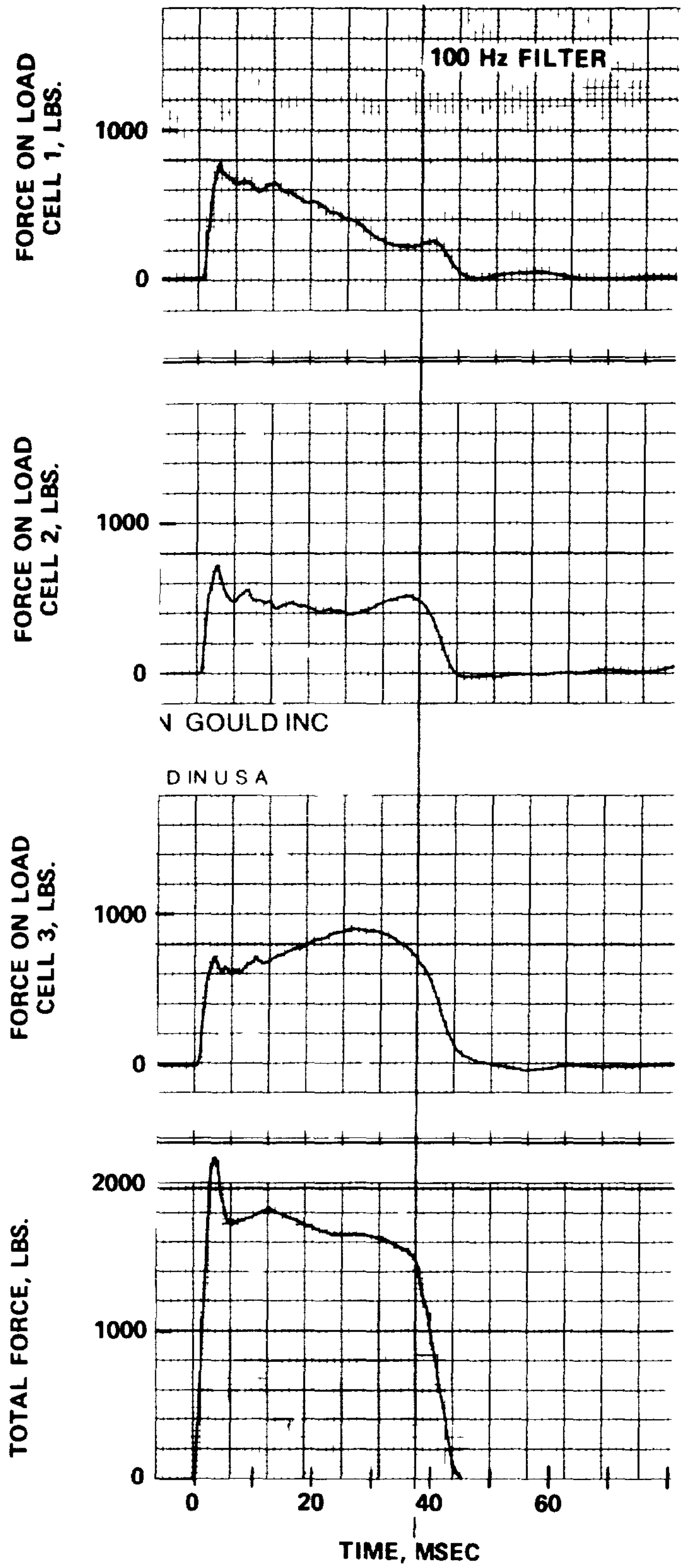


Figure 13 FORCE TIME HISTORY OF TEFLON ENERGY ABSORBER. EXTRUSION DIAMETER RATIO = 0.93, DROP HEIGHT = 13', DROP WEIGHT = 70#.

force is about 2100 lbs. and the steady-state force is around 1850 lbs. Waveform is satisfactory except for the initial peak. A change in the chamfer would probably reduce this peak. However, a more satisfactory change would be to make the rod stationary and use a moving die. Moving die extruders are less likely to have high starting force peaks.

Adaptation of the extrusion process to the EA steering column was not carried beyond the above demonstration. Strength and load path considerations would strongly influence the interface between the column and the four bar linkage.

3.6 Knee Bars

Simple energy calculations demonstrate that deceleration of the whole occupant mass with the steering column alone would require unrealistic stroke lengths because the allowable force is limited by tolerable chest loads. The other available load path (for unrestrained occupants) is the knee-femur-pelvis complex. Permissible femur loads are estimated to be at least 1400# - so a 2800# deceleration force could be applied in a symmetric load situation. Possible auxiliary advantages of knee bars are the following:

- small separation distance between knee bar and knees leading to substantial ride down

- penetration of the knee target would tend to bring lateral support loads into play

- reduced submarining tendencies

Knee restraints were under investigation at Calspan as part of a rear seat inflatable occupant restraint system in 1971 and 1972 (Reference 13). Good performance was being achieved and, therefore, the design was adopted for the driver. Figures 14 and 15 (Figure 15 taken from Reference 13), show the design and the performance, respectively.

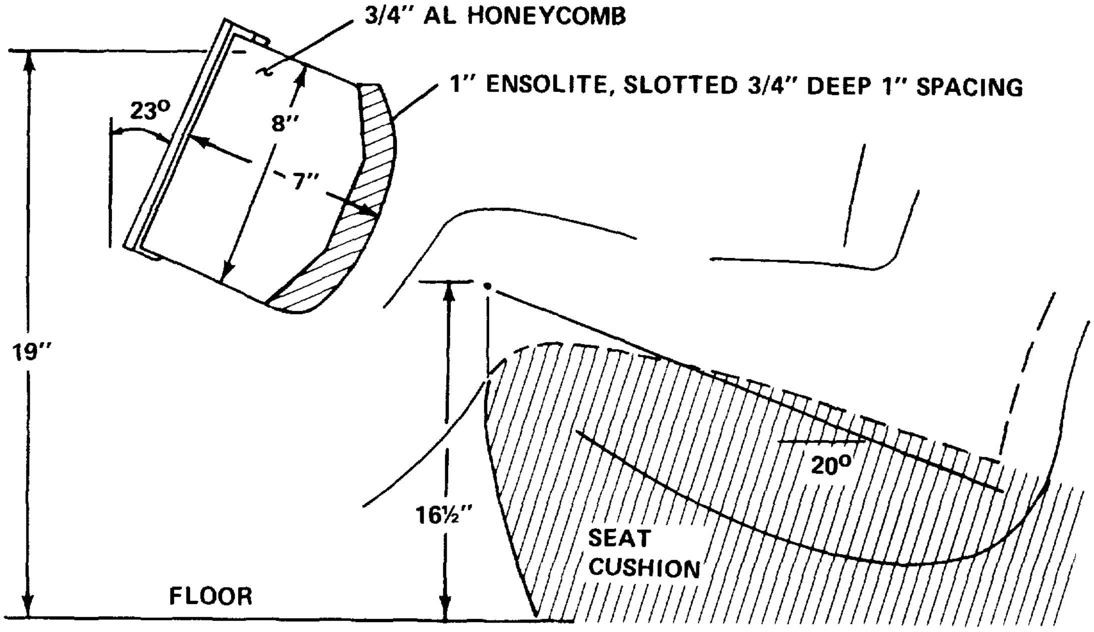


Figure 14 KNEE ENERGY ABSORBER INSTALLATION

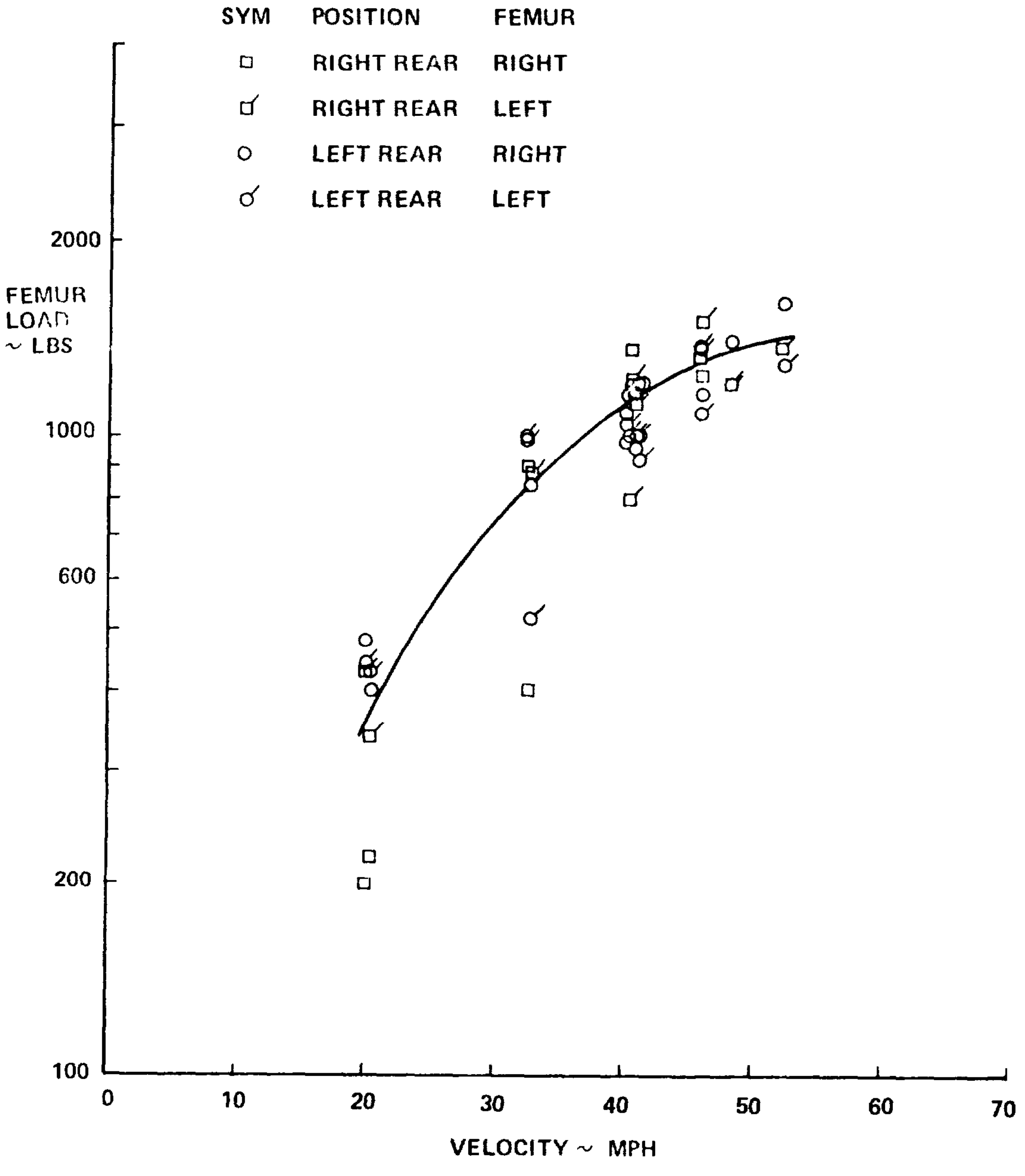


Figure 15 FEMUR LOAD AS A FUNCTION OF SLED VELOCITY, 50th PERCENTILE DUMMIES, NORMAL SEATED POSITION (From Reference 13)

4. DRIVER AIR CUSHION

4.1 Background

Limited piggyback tests were made with advanced occupant restraint systems - air cushions. Observation of deployment times of "fast" state-of-the-art air bags was the main concern. The events were substantially more severe than the design conditions for the restraint - and the compartment dimensions and vehicle acceleration pulses were not necessarily consistent with the restraint system design.

It was pointed out in Section 2.2 that there is a limit velocity below which the unrestrained occupant, for a fixed initial separation distance, b, cannot benefit from ridedown. This limit velocity corresponds to the condition of occupant separation distance being equal to vehicle crush distance (Section 2.2). Deployment of an inflatable occupant restraint could have the effect of increasing ridedown by reducing the effective separation distance between driver and "structure". It is, however, to the higher input speeds that attention will be directed here.

4.2 Simple Analysis of Ridedown Potential at High Speeds

Reference 14 contains a brief analysis of occupant/sled motion for square wave accelerations of the sled. Sled velocity at time of impact is shown to be:

$$V_r = V_1 \sqrt{\frac{b}{D_v}}$$

where

V_r = sled velocity at time of occupant impact
V_1 = ultimate velocity
b = initial distance from occupant to structure
D_v = vehicle acceleration distance

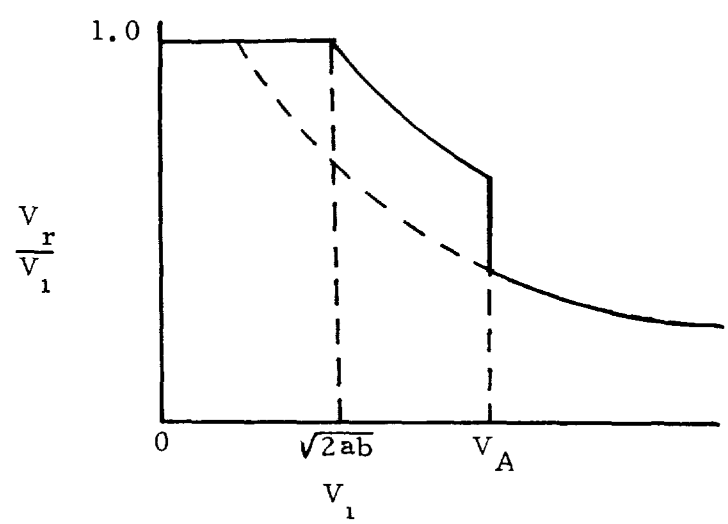
In a uniform acceleration vehicle crash event, then, the velocity ratio between occupant and frame at the time of occupant impact would be given by:

$$\frac{V_r}{V_i} = \sqrt{\frac{b}{D_V}}$$

$$= \sqrt{\frac{2\ b\ a}{V_i^{\ 2}}}$$

where a = vehicle acceleration.

It will be recalled that for a given value of b there is a $\bar{V}_i = 2\ ab$ below which $\frac{V_r}{V_i} = 1.0$ (Section 2.2). Use of an air cushion changes the distance from driver to structure to a new effective valve b' which depends on deployment time and threshold velocity, V_A. The resulting situation is sketched below for the case where the air cushion has an assumed square wave force characteristic.



4.3 Driver Air Cushion Deployment

Information on time required for deployment of driver air cushion restraints (advanced restraint system) was sought using "fast" available 30 mph equipment supplied by Olin Corporation. Sensing time was also measured on the impact detection unit (10g/10 msec impulse for closing) so that actuation time - sensing interval plus deployment interval - was obtained. Performance data, in terms of loads on simulated occupants, was not of great interest because current state-of-the-art hardware was employed - i.e., 30 mph systems consistent with FMVSS 208 requirements. It is believed that a careful review of the performance data, however, would show that the driver cushion was applying chest loads that would enhance high speed survivability if head accelerations were brought under control.

Two aspects of bag deployment were considered:

- angle
- time

The first piggyback test (Run #33) films showed the cushion to be deploying high (on the chest and face of the simulated occupant). This observation reinforced the opinion that steering column angularity should be reduced (for column stroke reasons described previously) and, subsequently, column angles slightly less than 20° were used on the modified standard size vehicles (Ford Custom sedans).

System actuation times demonstrated were good. Sensing time, for example, was as low as 7 msec on modified Vega vehicles. These cars (Reference 15) were characterized by energy management structures that provided direct strut paths from the bumper to the firewall (where the sensors were mounted). Firewall-mounted sensors on unmodified vehicles, on the other hand, showed relatively longer sensor times. These results are tabulated below:

Test No.	Vehicle	Target	Impact Velocity, mph	Sensor Pulse Time, msec.
51	1971 Vega Mod 6	Pole	58	7
51				7
47	1969 Ford	1971 Vega	80 (closing)	30
47	1971 Vega	1969 Ford	80 (closing)	17
78B	1971 Vega Mod 8	1969 Ford	80 (closing)	7-1/2

*Two separate sensors

Similar early detection (7 msec) was achieved on modified standard size cars by mounting the sensor on the frame at the first frame cross member (Reference 16).

Deployment times for the augmented air driver air cushions were short. For example, in Test No. 42, driver bag deployment times of 7 and 8 msec were deduced from high speed film of the Ford/Buick baseline head-on impact at 80 mph closing speed (Reference 17).

Forward sensor location combined with the rapid inflation demonstrated should yield actuation times (sensing plus deployment) of about 20 msec. This would be a substantial improvement over current performance. Additional attention to possible threshold problems is still required, however.

5. INTRUDER/ABSORBER

Driver survival in high speed impacts hinges on a combination of compartment integrity and tolerable loads on the decelerating occupant. A simple estimate of the geometry implied for occupant survival was presented in Reference 18 in terms of the following "whole body" severity index, S.I., :

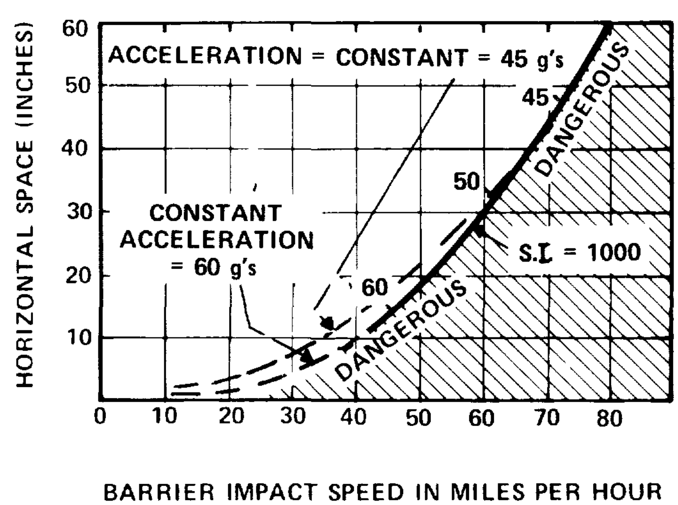
$$S.I. = \int_0^T a^{2.5}\, dt \quad 1000$$

where

a = acceleration, g's

T = event duration, seconds

Assumption of constant force deceleration permitted the plotting of deceleration distance vs barrier impact speed (Reference 18) for the bounding value S.I. = 1000 as shown below. Added to the plot are the constant acceleration lines for 45 and 60 g's and tick marks where constant acceleration lines (45, 50, 60 g's) intersect the S.I. = 1000 curve.



Inspection of the above sketch shows that a 45 g acceleration over a distance of about two feet would allow arrest from a speed of 50 mph without exceeding the S.I. limit. However, the corresponding force level required on a

50% male would be about 7200 lb/ft. which is about 1500 lb/ft. greater than the sum of allowable chest load (say, 2500 lb/ft.) and the femur loads (say, 1600 lb/ft. each). When, in addition, imperfections of the energy absorbers are accounted for, the stroke required increases further and the desirability of a driver restraint beyond the air cushion becomes evident. An intruder/absorber was selected for this purpose.

Briefly, the intruder/absorber is a passive restraint device which deploys* from the steering wheel axis and acts as a hydraulic shock absorber when impacted by the driver. Figure 16 shows the elements disassembled in photographs (a) and (b), absorber extended in (c) and retracted in (d). Performance data from an early trial are shown in Figure 17. To be noted in the force/displacement record is the programmed force level rise from 1900 lb/ft. to about 4000 lb/ft. Subsequent adjustments in fluid properties produced a more rapid initial force rise and, hence, a greater stroke efficiency.

All the energy for the deployment is supplied by a squib similar to the type used on air cushion systems, energized by a 12 volt source (battery). Deployment times as low as 7 msec. have been measured - but the addition of a magnesium cap (to obtain lower pressures on the chest) and a crushable pad to provide protection for the out-of-position driver would probably increase that time to about 15 msec. Sensor closing times would be the same as those associated with inflatable occupant restraint systems - about 7 msec.

Additional data on the experimental intruder/absorber are contained in Appendix 3.

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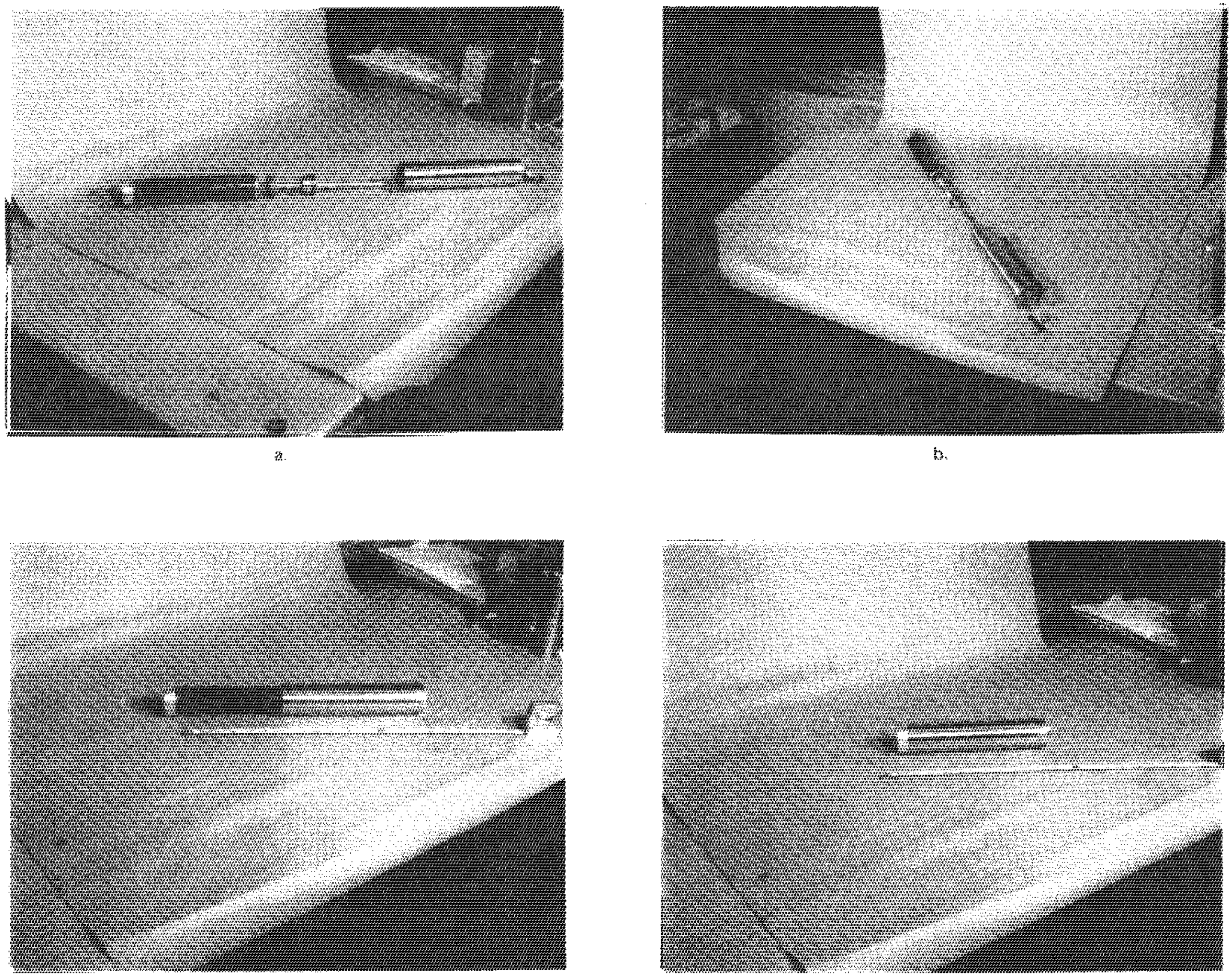
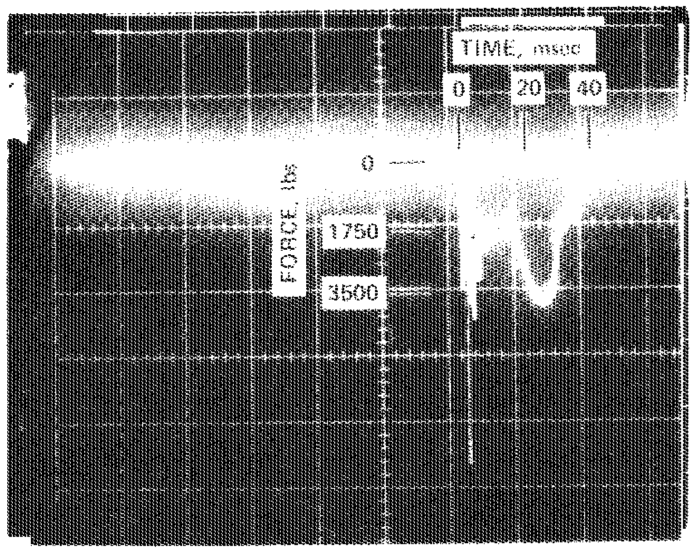
* Deployment was limited to 5" in keeping with FMVSS 204 limits.



a. b. c. d.

Figure 16 INTRUDER/ABSORBER (a), (b) DISASSEMBLED, (c) EXTENDED, (d) RETRACTED



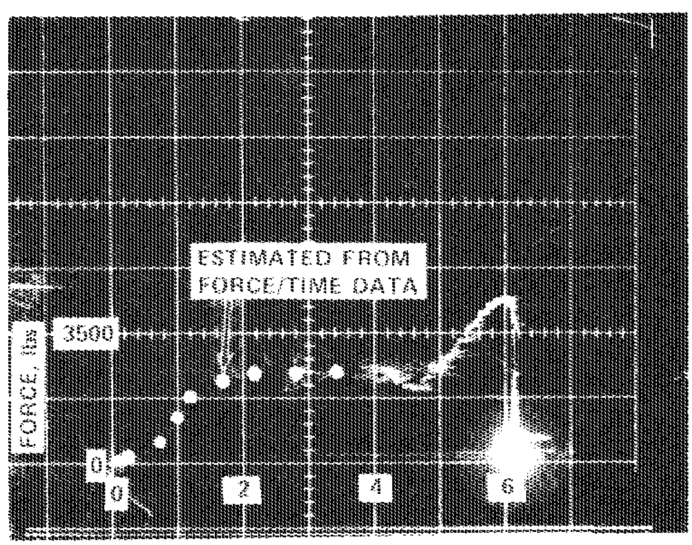


Figure 17 FORCE-TIME AND FORCE-DISPLACEMENT PERFORMANCE OF ABSORBER SECTION OF INTRUDER/ABSORBER (20 MPH, TAYLOR DROP TOWER)

6. SUMMARY AND RECOMMENDATIONS

Previous discussion sections and results presented in the literature show that vehicles with energy management structures will influence the driver in the following ways:

- compartment integrity will be maintained to barrier speeds above 50 mph
- mean accelerations will be higher than those of current vehicles
- onset rates will be higher than those of current vehicles

The higher onset rates and higher mean accelerations have compensating effects with respect to driver ridedown but, on balance, there probably will be a net loss compared to current conventional automobiles. Loss of driver ridedown at the lower velocity limit (Section 2.2) should not introduce a significant new problem because current steering columns have the EA capacity to satisfy injury criteria at speeds up to at least 25 mph (Figure 11) *.

Emphasis is required on means for avoiding defeat of the driver restraint system by peripheral effects. The following items are particularly noted regardless of structural characteristics:

- disengagement of the steering gearbox, to avoid intrusion and/or rotation of steering assembly
- seat and seatback structural integrity
- maintenance of relatively small longitudinal angle

* For symmetrical (barrier) impacts.

Moderate steering column angle changes have no effect on vehicle handling as long as the frictional torque is not increased.

Three aspects of driver restraint have been identified:

- positioning, obliquity compensation
- energy absorption
- ridedown enhancement

Elements that contribute to the above functions included the following (see Figure 1):

- intruder/absorber
- air cushion
- column jacket
- knee bar
- steering hub pad
- four-bar linkage

The driver position is the major occupancy position and deserves emphasis. Accident data (Figure 6) suggest that current EA column test procedures do not reproduce the frontal accident situation in enough detail.

The following recommendations are made with respect to EA column demonstration:

1. Increase the allowable load in the body block test to 2500 lb.

2. Increase the demonstration speed of the body block to 25 mph.

3. Demonstrate survivable loads (in the FMVSS 208 sense) on anthropomorphic devices in a body buck (including lower dash or knee bar) at $\pm 20°$ obliquity at 20 mph.

The following recommendations are made with respect to driver survivability enhancement in high speed impacts:

1. Determine EA element characteristics (e.g., corresponding to elements in Figure 1) which, in combination, will provide:

   - protection in frontal ($\pm 15°$) collisions for the range of driver masses representative of 5% female to 95% male

   - protection up to a barrier velocity of 50 mph (and $\pm 15°$) for the above range of driver masses

2. Determine the effect of vehicle crash pulse on the characteristics satisfying item 1.

3. Determine the incremental restraint performance that would be required to obtain "survivable" loads on a 50% male anthropomorphic device at the limit of performance of the current generation of energy management vehicles.

4. Determine the effect of vehicle size on required driver restraint characteristics (the vehicle having an energy management structure).

5. Pursue the implications of an intruder/absorber which

   - intrudes further than the currently allowed 5"
   - has a stroke greater than the current model (about 7")
   - has a reduced deploying mass
   - employs a second squib to provide redundancy and shorter deployment time

6. Perform 3-D occupant simulation studies to inspect possible trade-offs between vehicle interior padding and driver restraint elements in frontal oblique impacts.

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APPENDIX 1

FOUR BAR LINKAGE - STEERING COLUMN SUPPORT PROVIDING ENERGY ABSORPTION FOR LATERAL AND OFFSET IMPACTS

Four-Bar Linkage - Steering Column Support
Providing Energy Absorption for Lateral and Offset Impacts

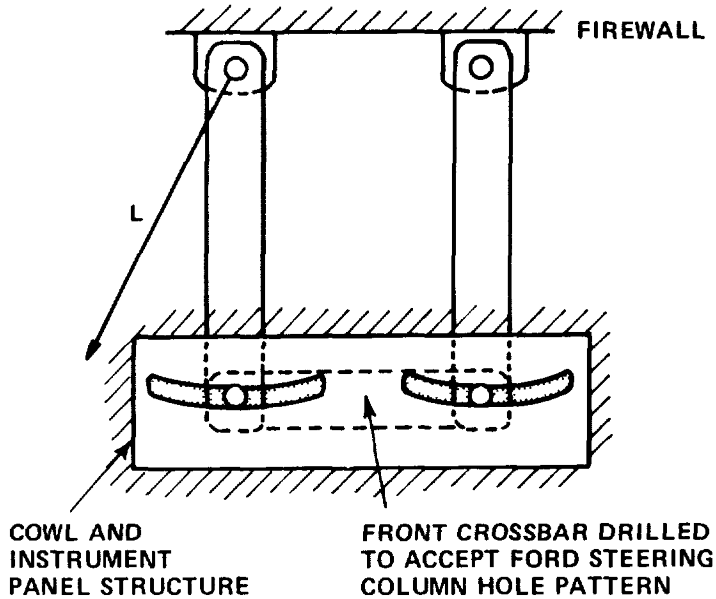
A possible means for accommodating lateral and/or offset impacts of drivers on steering wheels is to provide a structure that imposes a lateral load within human tolerances. Such a structure should have the following desirable characteristics:

1. Tolerable force level.
2. Energy absorption capacity sufficient to handle a 30° oblique impact (in conjunction with the axial E. A. Element).
3. Promote retention of the driver body, for example, by encouraging the steering wheel to remain parallel to its initial plane.
4. Be compatible with the steering task.
5. Possess travel limits (not necessarily symmetric) to preclude carrying the driver into the door.

One possible mechanism satisfying most of these requirements is described below - a four-bar linkage that provides energy absorption by metal cutting at the four hinge locations and a combination support and guide system that limits lateral travel.

Configuration

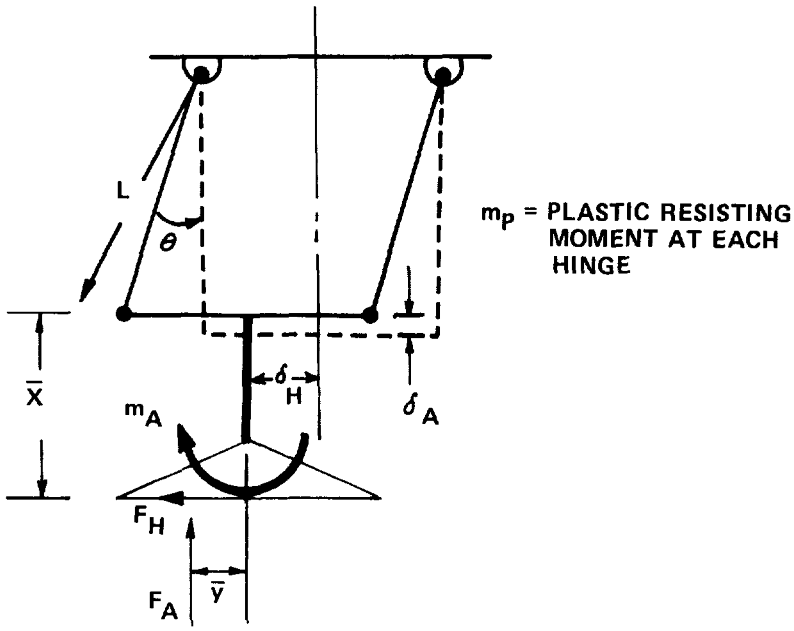
The following sketch shows a postulated top plan view for a standard size Ford:



In the above sketch, the firewall acts as one arm of the four bar linkage. The opposite arm would accept the bolt hole pattern of a standard Ford steering assembly attachment plate and it is anticipated that the steering shaft, from that position to the firewall, would be a flexible shaft. The hinge pins indicated also furnish the compression needed to force toothed lock washers into the arms and support points of the four bar linkage. The cutting of the overlapping metal arms of the linkage by the toothed lock washers furnishes the plastic deformation necessary for energy dissipation. The slots indicated at the instrument panel hinge pins permit movement of the linkage laterally on circles of radius L. As shown on the sketch, the slots would be tapered in order to provide a hardening limit for large travels and the slot length toward the centerline of the vehicle should be greater than the slot length toward the door. (It is considered at this time that it is probably undesirable to have the driver contact the door and/or A pillar in a substantially frontal collision.) A brief derivation of the performance of such a device is presented in the next section on the basis of rigid plastic deformation theory.

Theory

Geometry and loading conditions for the four-bar E.A. column support are shown on the following sketch:



Rigid plastic analysis is based on the assumption that all energy absorption occurs at discrete elements and that the remainder of the structure is rigid. In the present case, all energy absorption takes place at the hinge locations - and the links between are assumed to be rigid. Inertial effects are neglected and the rate of work of external forces is assumed to be equal to the rate of energy dissipation at the hinges:

$$\text{Rate of work of external forces} = F_H \bar{X} \dot{\theta} + M_A \dot{\theta} + F_A \bar{y} \dot{\theta} + F_H \dot{\delta}_H + F_A \dot{\delta}_A \quad (1)$$

$$\text{Energy dissipation at each hinge} = M_p \dot{\theta} \quad (2)$$

Since the links are inextensible,

$$\delta_H = L \sin\theta \; ; \quad \dot{\delta}_H = (L \cos\theta)\, \dot{\theta}$$

$$\delta_A = L(1-\cos\theta) \; ; \quad \dot{\delta}_A = (L \sin\theta)\, \dot{\theta}$$

Equations (1) and (2) produce

$$(F_H L \cos\theta)\dot{\theta} + F_A (L \sin\theta)\dot{\theta} = 4M_p\dot{\theta} - M_A\dot{\theta} - F_H\bar{x}\dot{\theta} - F_A\bar{y}\dot{\theta} \tag{3}$$

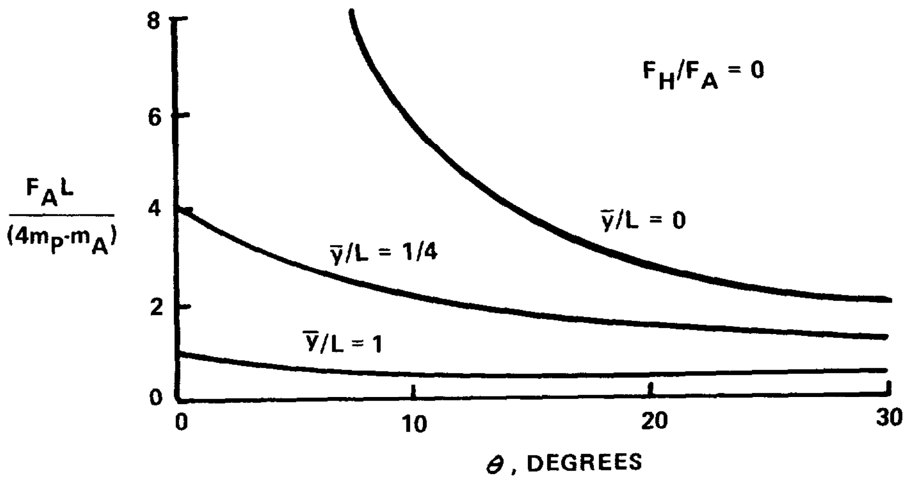
or

$$\frac{4M_p}{L} = F_A\left[\frac{F_H}{F_A}\left(\cos\theta + \frac{\bar{x}}{L}\right) + \left(\sin\theta + \frac{\bar{y}}{L}\right) + \frac{M_A}{F_A L}\right] \tag{4}$$

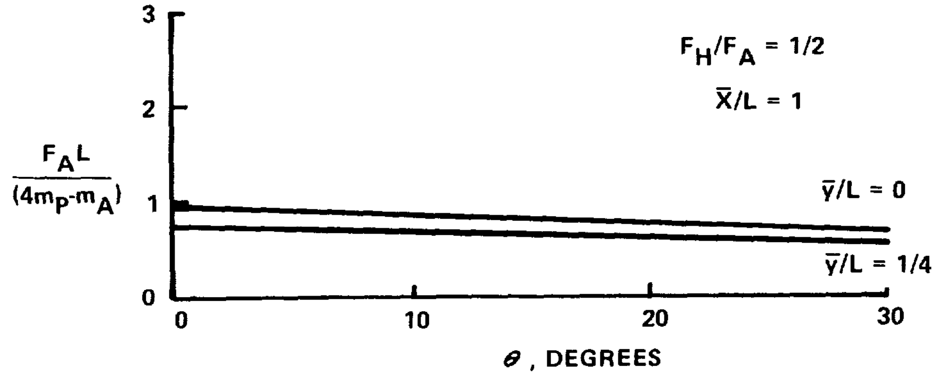
Pure torque component of loading, M_A, is judged to be relatively small. In any case, M_A can be lumped with the dissipation moment:

$$\frac{F_A L}{4M_p - M_A} = \frac{1}{\left(\sin\theta + \frac{\bar{y}}{L}\right) + \frac{F_H}{F_A}\left(\cos\theta + \frac{\bar{x}}{L}\right)} \tag{5}$$

Results computed from the above equation are shown in the following plots.



8
6
4
2
0
$\frac{F_A L}{(4m_P \cdot m_A)}$
$F_H/F_A = 0$
$\bar{y}/L = 0$
$\bar{y}/L = 1/4$
$\bar{y}/L = 1$
0
10
20
30
θ, DEGREES



3
2
1
0
$\frac{F_A L}{(4m_P \cdot m_A)}$
$F_H/F_A = 1/2$
$\bar{X}/L = 1$
$\bar{y}/L = 0$
$\bar{y}/L = 1/4$
0
10
20
30
θ, DEGREES

Equation (5) exhibits limit cases, within the operating range, at which $F_A \rightarrow \infty$ (that is, at which no rotation of the hinges will occur). This is seen to be (for small θ)

$$\left.\frac{F_H}{F_A}\right|_{F_A \rightarrow \infty} = - \frac{\theta + \frac{\bar{y}}{L}}{1 + \frac{\bar{x}}{L}}$$

and it corresponds to conditions where, roughly, the moment due to the axial load offset is balanced by the moment due to the horizontal force component.

The performance sketches show the scheme has certain desirable features:

1. When the axial force offset and the horizontal force components are small, the axial force component required to collapse the four-bar linkage is large.
2. When the horizontal force component is large, the axial collapse loads are nearly independent of angle and axial load offset.

Sample calculations are made in the next section.

Sample Calculation and Trial Design

The preceding rigid plastic theory applied to the generic configuration previously postulated permits the calculation of arm lengths and resisting forces generated for a given hinge energy dissipation. Design data for estimating the dissipation moments at the hinges were developed by means of a few simple tests. These are discussed and presented in the Appended section, "Energy Dissipation in Hinges of Four-Bar Linkage-Metal Cutting Serrated Washer Tests". The dimensions are defined on the following sketch and are compatible with the compartment dimensions of standard size Ford vehicles.

Estimate L.

From the data in the Appendix, choose a configuration giving $M_p = 400\ FT^{\#}$

and choose to use four such hinges to provide

$$4 M_p = 1600\ FT^{\#}$$

Take $F_A \approx 1600^{\#}$, $F_H \approx 0$, equation (4) gives

$$\frac{1600}{L} = 1600 \left(\sin \theta + \frac{\bar{y}}{L} \right)$$

$$\text{if } \theta \approx \frac{1}{4} \text{ radian}, \quad \frac{\bar{y}}{L} = \theta\left(\frac{1}{2}\right)$$

$$\boxed{L \approx 1.3\ FT.}$$

Now choose $F_A = 0$, $F_H \approx 800^{\#}$,

$$\frac{1600}{L} \approx 800 \quad 1 + \frac{\bar{x}}{L}$$

$$\text{FOR } \frac{\bar{x}}{L} = \theta(1),$$

$$\boxed{L \approx 1\ FT.}$$

Therefore, $L \approx 1\ FT$ is consistent with desired hinge performance.

Geometric Constraints

Maintaining current Ford attachment points and steering wheel hub locations gives

$$\bar{x} \approx 1\ FT$$

$$\frac{\bar{x}}{L} \approx 1$$

Performance

Assume no pure torques ($M_A = 0$). Then equation 4 becomes

$$\frac{1600}{1} = F_A \frac{F_H}{F_A}\left[(\cos\theta + 1) + \left(\sin\theta + \frac{\bar{y}}{L}\right)\right]$$

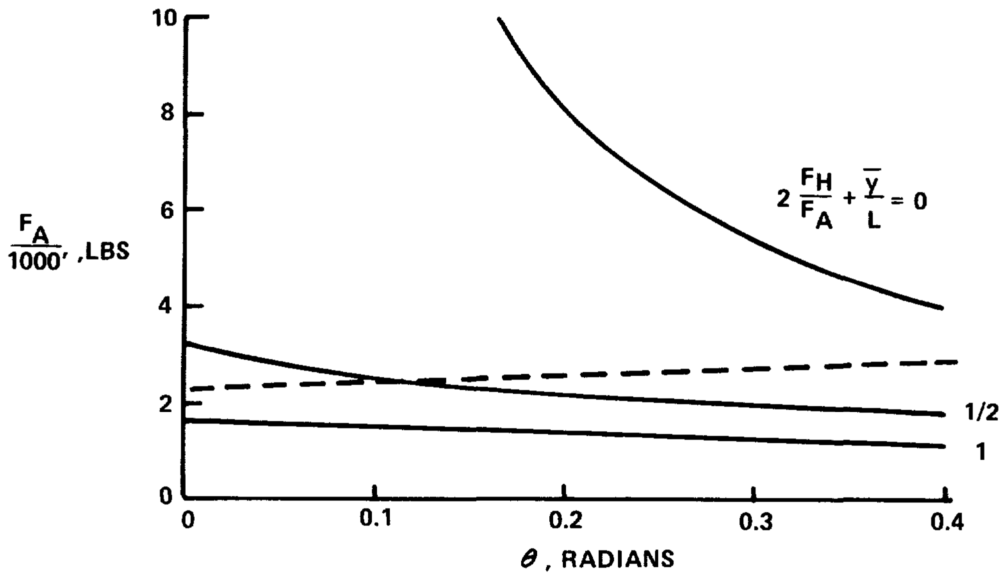
For θ small, $\cos\theta \approx 1$, $\sin\theta \approx \theta$ radians

$$1600 = F_A\left[2\frac{F_H}{F_A} + \theta + \frac{\bar{y}}{L}\right]$$

or

$$F_A = \frac{1600}{2\frac{F_H}{F_A} + \theta + \frac{\dot{y}}{L}}$$

Calculations produce the following results:



The dashed line on the above sketch corresponds, roughly, to the axial load under which axial collapse of the extrusion absorber and/or the intruder/absorber would occur. When either or both of these EA devices are activated, L changes and the four-bar linkage responses may change.

Energy Dissipation Tests of Metal Cutting Serrated Washers

Internally serrated and externally serated washers were tested to develop data on their ability to supply energy dissipation moments. In particular, effects of washer configuration and preload (torque) were investigated. Bars of cold rolled steel (1/4" x 2" cross section) were used for the cut members. Results are presented in the following table.

Polaroid pictures of specimen parts after test are shown in the accompanying Figures 1-1 and 1-2. Figure 1-3 shows the simple loading arrangement for measuring moment required for rotation of the arm. These are essentially static moments required for hinge rotation. Rate effects cannot be evaluated until drop tests are run. Performance plots are shown in Figure 1-4.

TEST TABLE - SINGLE HINGE CUTTING TORQUES

Configuration	Load Direction		Bolt Initial Joining Torque	Load at 2'	Moment
	Clockwise	Counter Clockwise	ft. lbs.	lbs.	ft. lbs.
1. Internally Serrated	X		115	107	214
2. Internally Serrated (Hardware same as 1)	X		130	107	214
3. Internally Serrated		X	115	167	334
4. Internally Serrated (Continuation of 3)		X	130	207	414
5. Repeat 3		X	115	142	284
6. Repeat 4		X	130	192	384
7. Internally Serrated		X	115	142	284
8. Internally Serrated		X	130	157	314
9. Externally Serrated		X	115	157	314
10. Externally Serrated		X	130	182	364
11. Domed Washer				Small	N. G.
12. Sand blasted straps Externally Serrated		X	115	157	314
13. Sand blasted straps Externally Serrated		X	130	207	414

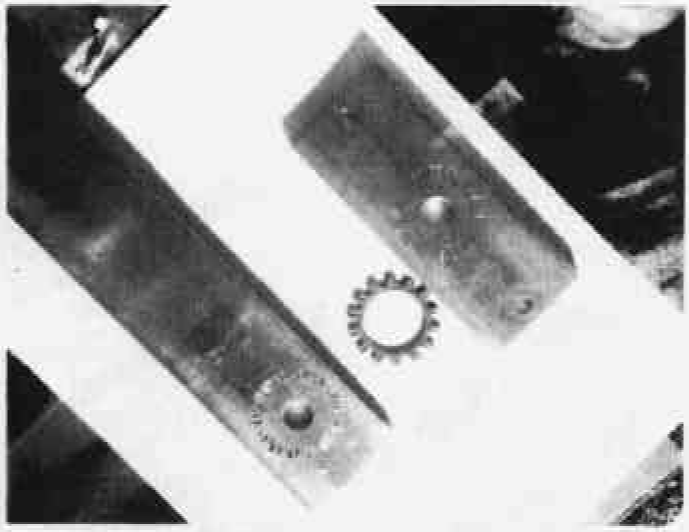
'Changed nut and bolt used to hold parts together.

Allowable travel for the washers tested is about 15°. Corresponding energy from the data in Figure 1-4.

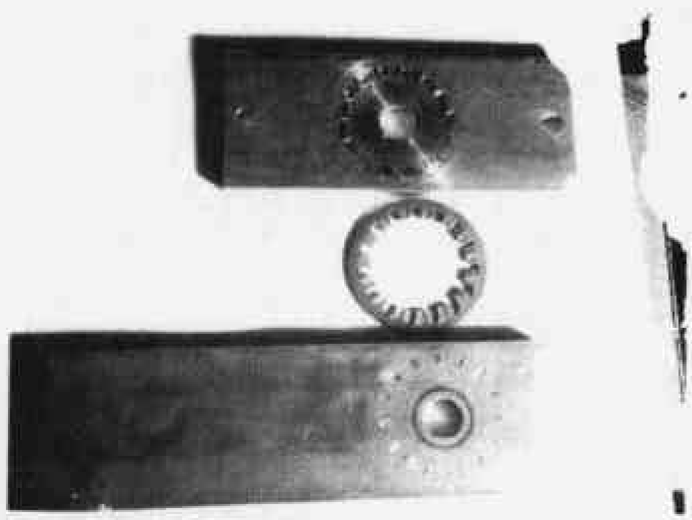
$$E \approx \frac{15}{57} \cdot 4.8 \cdot (Q_o - 50) \text{ ft.\#}$$

$$\approx 1.2 \, (Q_o - 50) \text{ Ft.\#}$$

in the range around $Q_o = 120$ ft.# where Q_o is the torque applied to the nut of the cap screw used as the hinge pin.



EXTERNALLY SERRATED WASHER
AND CRS HINGE ARMS



INTERNALLY SERRATED WASHER
AND CRS HINGE ARMS

Figure 1-1 SINGLE HINGE TEST SPECIMEN

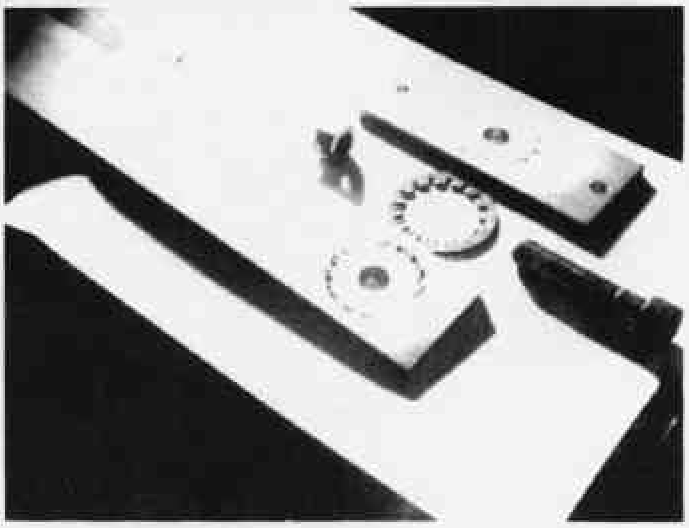


Figure 1-2 INTERNALLY SERRATED WASHER, CRS HINGE ARMS AND JOINING BOLT

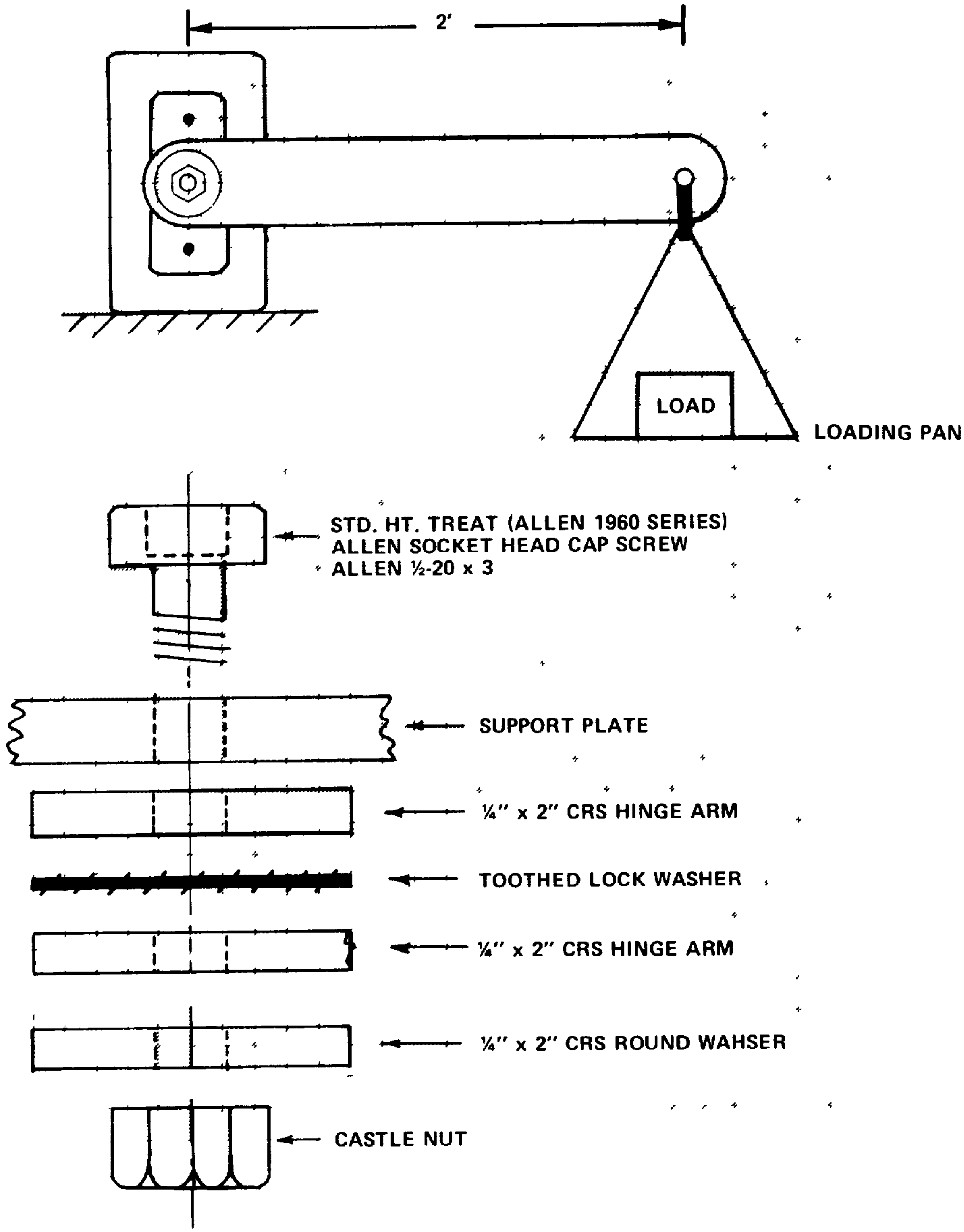
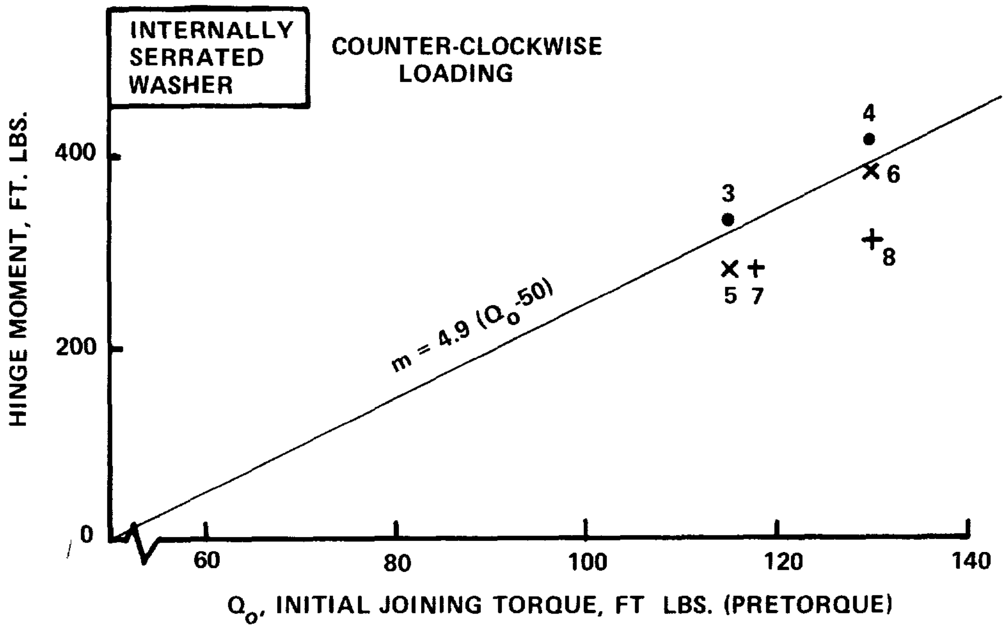


Figure 1-3 TEST SETUP FOR MEASURING MOMENT RESISTANCE ACHIEVABLE WITH METAL CUTTING TOOTHED WASHERS. (FOUR-BAR LINKAGE APPLICATION).



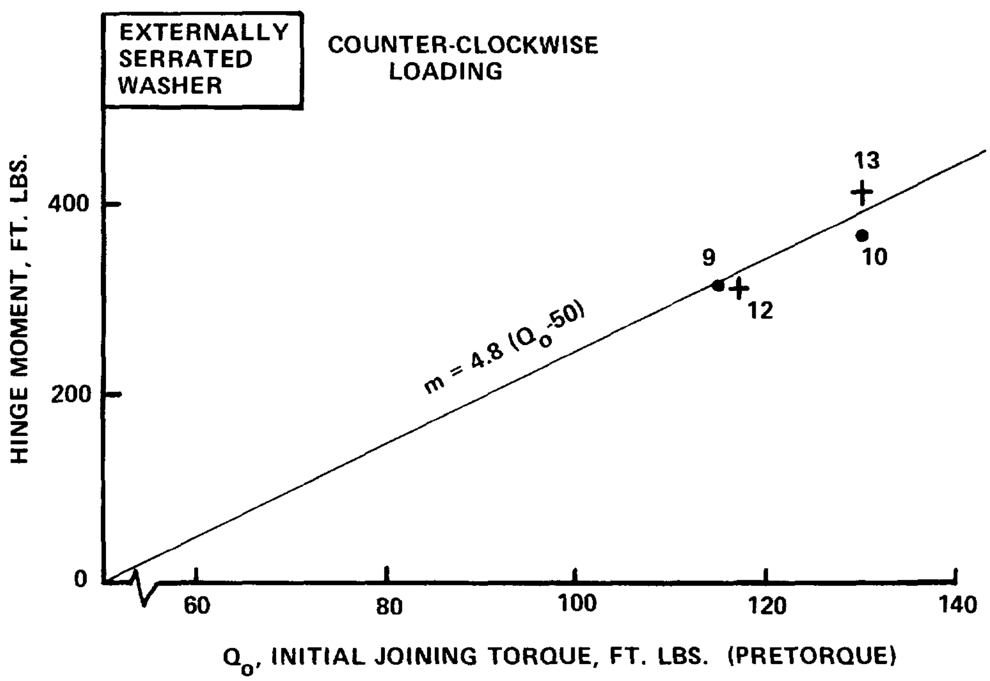


Figure 1-4 PERFORMANCE PLOTS, SINGLE HINGES

APPENDIX 2

LITERATURE SEARCH - ENERGY ABSORBING (EA) STEERING ASSEMBLIES

BASIC RESEARCH IN CRASHWORTHINESS II

Literature Search
Energy Absorbing (EA) Steering Assemblies

ATTACHMENT NO. 2
to
ELEVENTH PROGRESS REPORT
PERIOD 1 May to 31 May 1971

PROJECT NO. YB-2987-V
CONTRACT NO. FH-11-7622

For
U. S. DEPARTMENT OF TRANSPORTATION
National Highway Traffic Safety Administration
Washington, D. C. 20591

Literature Search - Energy Absorbing (EA) Steering Assemblies

Introduction

Available literature and communications were reviewed to obtain existing data on

- configurations,
- performance,

with respect to energy absorbing steering assemblies. This information is being used as a guide to additional required tests and analyses to define steering assemblies compatible with crashworthy structures.

No consideration is given in this review to "radical" steering systems that eliminate the steering wheel. Such surrogates should be emphasized when a mechanism is found that increases the survival probability of an unrestrained right front passenger above that of the driver.

All the reported EA devices seem to work effectively (that is, required energy absorption at allowable force levels) in purely head-on crash simulations under laboratory conditions. Accident investigation results and simple analyses on the other hand, suggest that initial conditions of a collision also strongly influence performance. All production EA columns in the United States are designed to limit column intrusion and available compliance tests show these have been successful.

Primary compendia of source listings used in this review were taken from References 2-1 and 2-2.

Conclusions

Basic considerations for passive restraints were documented some years ago, (e.g., Reference 2-3).

All production EA steering assembly configurations appear to satisfy the old FMVSS 203 demonstration requirements (15 MPH impact of SAE body block with peak load less than 2500 lbs.) and the 204 maximum intrusion specification.

Little information was found on the effects of oblique lateral angular impacts or offset lateral impacts on steering assembly EA performance. Some data on longitudinal column angularity has been published and this data suggests collapse load dependency on applied moments as well as applied force for current assemblies. Such dependencies require detailed examination of the load paths through the column supports and examination of possible frictional loads introduced by relative motion between telescoping elements.

Revision of steering column EA performance requirements to provide driver protection at higher speeds (greater than 15 MPH) implies:

1. Increased stroke for the energy absorber (e.g., by introducing controlled intrusion).

2. A column force-deflection profile that gives protection in "low" speed accidents without bottoming-out at high speeds.

3. Some mechanism for automatic adjustment of absorber force level to initial impact severity.

4. Reduction of column force sensitivity to longitudinal angularity.

5. Provision for energy absorption through plastic deformation of knee supports to provide body deceleration forces other than chest loads.

6. Re-examination of available tolerance data in conjunction with accident data to determine if there is a weight/strength relationship that would permit a rational compromise for EA allowable loads and/or force-deflection profiles.

7. Application of a Mercedes-like absorber at the steering wheel to provide some measure of steering wheel realignment with the chest during crush development.

8. Elimination of reliance on partial EA deformation to compensate for intrusion induced by compartment deformation since this leads to loss of useful stroke.

9. Introduction of EA mechanisms that can accommodate lateral oblique impacts of drivers on steering wheels.

Introduction of stiff crashworthy forestructures increases the difficulty of driver deceleration at a given impact speed because the time available for occupant deceleration may decrease. Additional steering column collapse distance is needed in such structures to compensate for potential loss of ride-down.* Hence, the instrument panel and windshield characteristics become involved in the setting of collapse distance limits.

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* A controlled intrusion limited by contact with the driver would alleviate this problem.

The multiplicity of EA configurations already in use augers difficulty in establishing demonstration procedures that encompass all devices.

Selected supporting data for most of the above items is contained in the following sections on configurations and performance.

Configurations

Figures 2-1 and 2-2 -- taken from Reference 2-4 -- illustrate steering wheel types and column geometries. It is evident that the multiplicity of these element arrangements and geometric layouts of steering assemblies used in automotive vehicles is huge. Therefore, it is necessary to restrict attention to a relatively few geometries and to trust that the performance characteristics can be duplicated with a number of possible combinations.

Energy absorbing steering assemblies are intended to decelerate the driver relative to the vehicle. Driver deceleration forces are introduced generally by the steering wheel and hub bearing on the chest. Irreversible features are incorporated (in the column support mechanisms and/or the shaft from the steering rod to the gearbox) to prevent intrusion of the column into the occupant compartment while permitting subsequent axial collapse motion of the steering wheel when the driver impacts the assembly.

U. S. Assemblies

All the production EA steering assemblies in the U. S. dissipate energy through plastic deformation of metal and/or sliding friction. Some differences arise in the methods of avoiding column intrusion. A convenient listing of the major assembly parameters is given in Reference 2-5 for 1970 and prior vehicles. Figure 2-3 identifies the major elements of the U. S. assemblies. Identification of the EA elements installed in various cars is given in Figure 2-4 and the corresponding absorbers are shown in Figure 2-5.

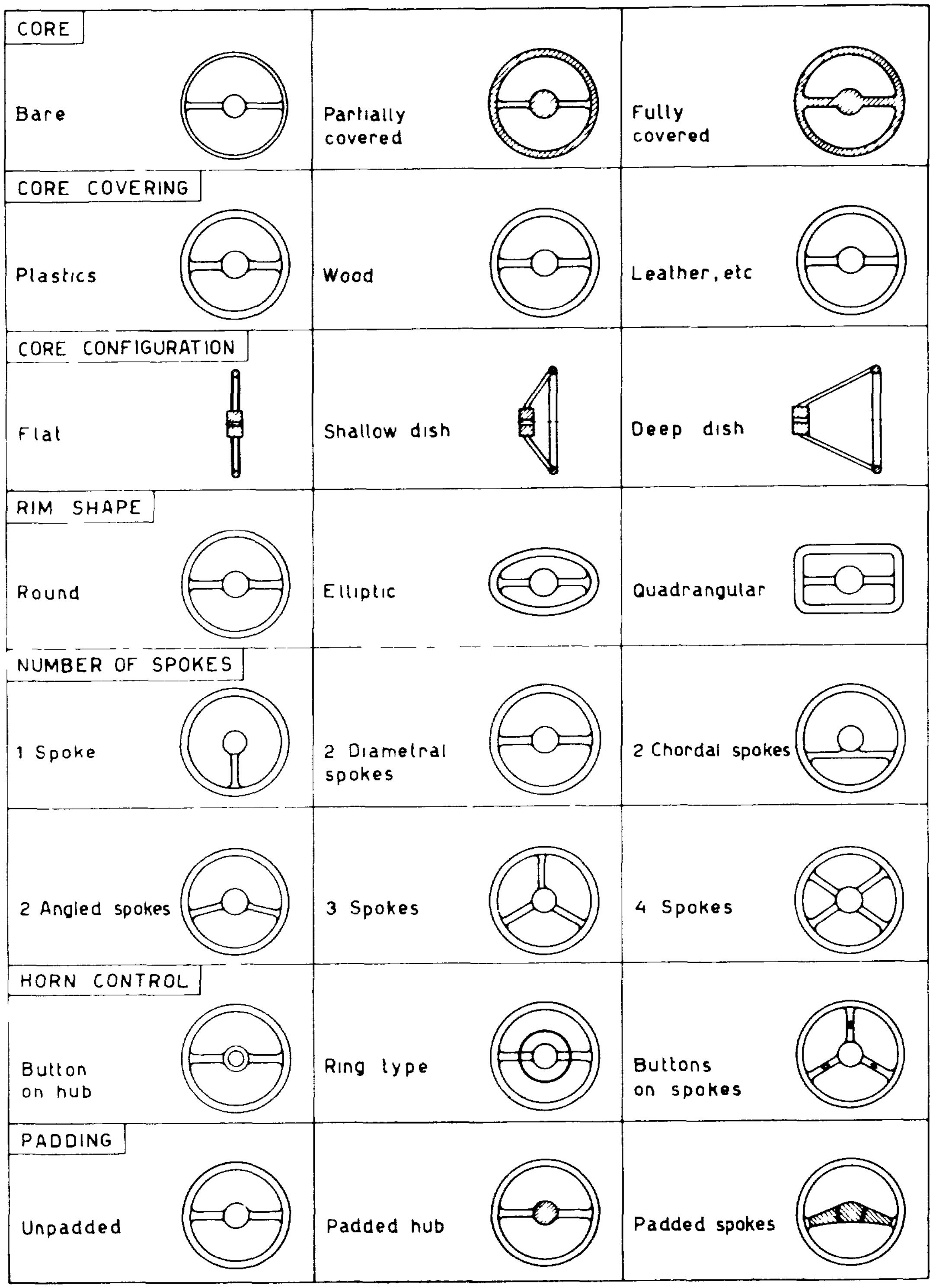


Figure 2-1 STEERING WHEEL TYPES (FROM REFERENCE 2-4)

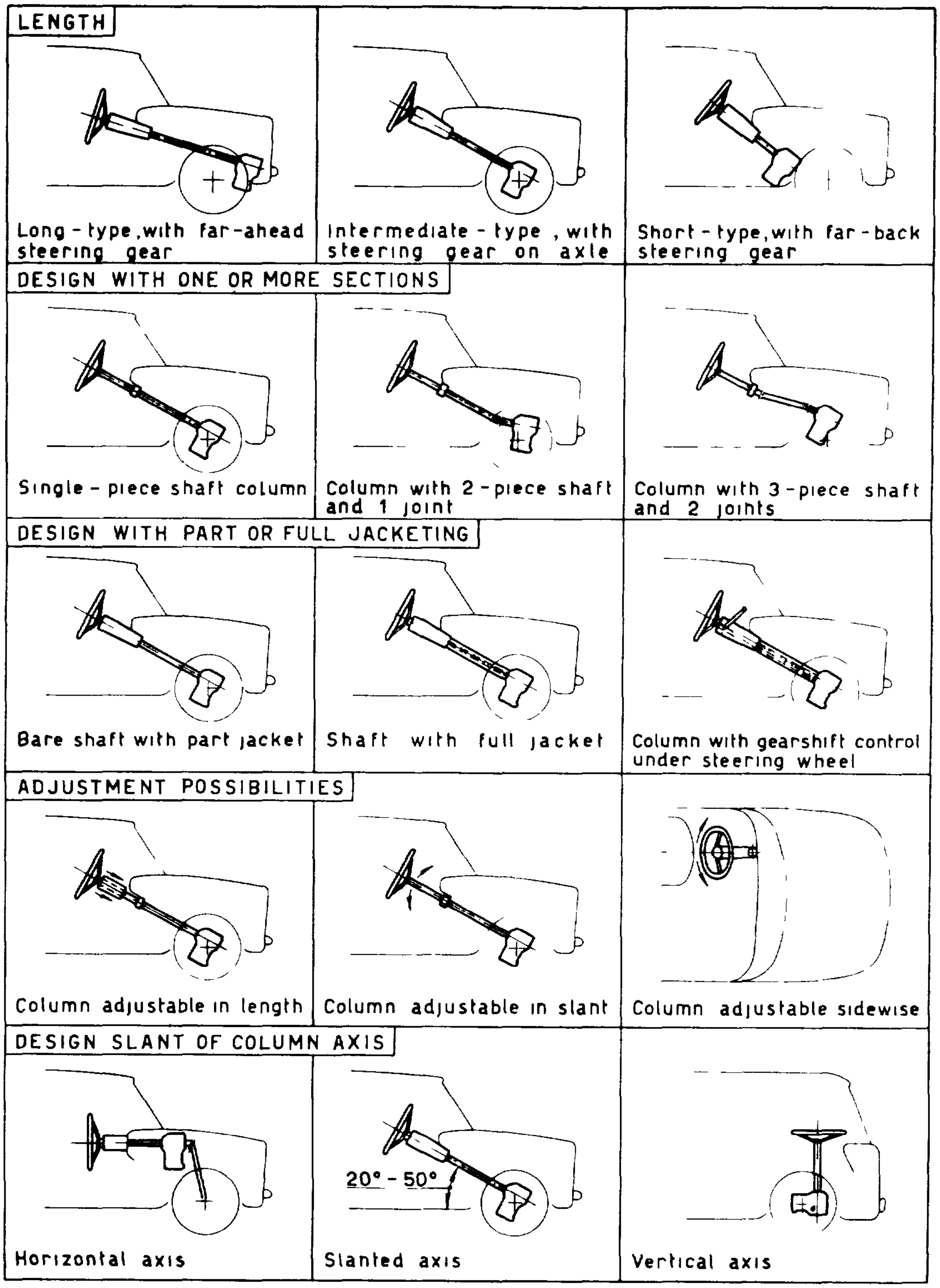


Figure 2-2 STEERING COLUMN GEOMETRIES (FROM REFERENCE 2-4)

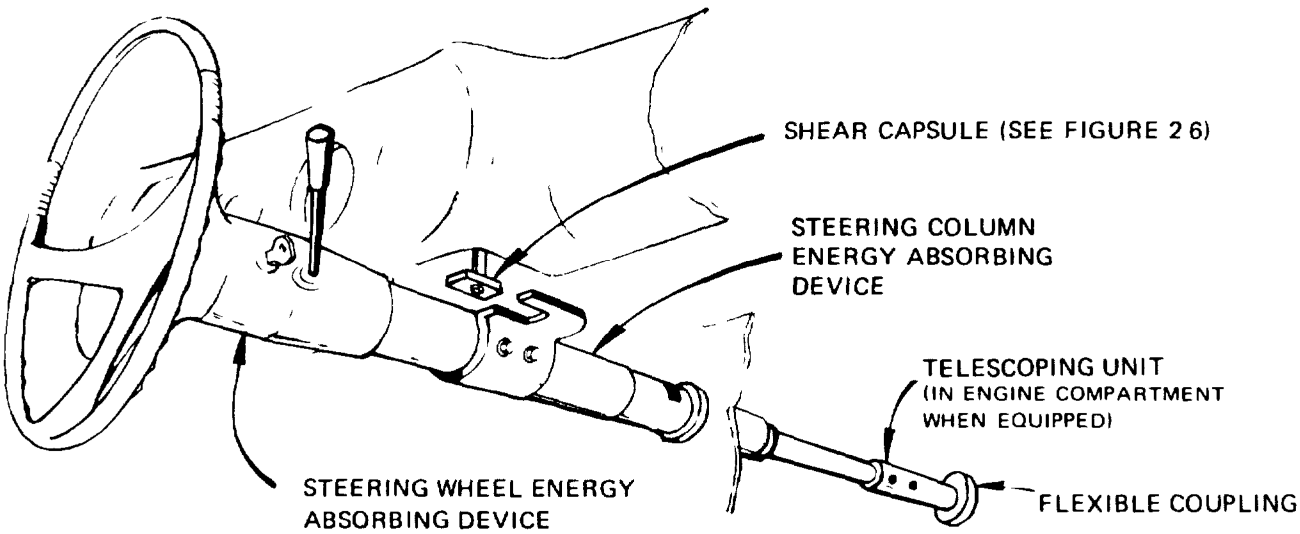


Figure 2-3 GENERIC STEERING COLUMN (ADAPTED FROM REFERENCE 2-5)

CORPORATION	MODEL YEAR	MAKE	STEERING COLUMN TYPE	E A * DEVICE TYPE	ORIGINAL LENGTH C
GENERAL MOTORS	69 70	ALL GM MAKE COLUMN COMBINATIONS OTHER THAN THOSE LISTED BELOW		2	8 25
		CORVETTE	ALL TYPES	3	8 25
		CAMARO CHEVELLE FIREBIRD TEMPEST GTO F 85, CUTLASS, VISTA-CRUISER SPECIAL, SKYLARK, SPORTWAGON GS 400	STANDARD	2	8 25
			TILT	2	7 8
		CADILLAC (EXCEPT ELDORADO)	TILT TELESCOPE	2	7 8
	70	GM SMALL CAR (XP887)	STANDARD	4	8 1
	67 70	OPEL	ALL TYPES	1	10 3
	67-69	CORVAIR	ALL TYPES	1	10 3
	67-68	ALL (EXCEPT CORVAIR & OPEL)	ALL TYPES	1	9 7
FORD	68 70	ALL MAKE COLUMN COMBINATIONS		5	9 5
	71	PINTO		6	
	71	CORTINA		5	
CHRYSLER	67 69	ALL MAKE COLUMN COMBINATIONS		1	9 7
	70	ALL (EXCEPT BARRACUDA & CHALLENGER)		1	9 7
		BARRACUDA & CHALLENGER		NONE	
AMC	67-69	ALL MAKE COLUMN COMBINATIONS		1	9 7
	70	ALL MAKE COLUMN COMBINATIONS		2	8 25

*SEE FIGURE 2 3 FOR SKETCHES

Figure 2-4 STEERING COLUMN EA DEVICE TABLE

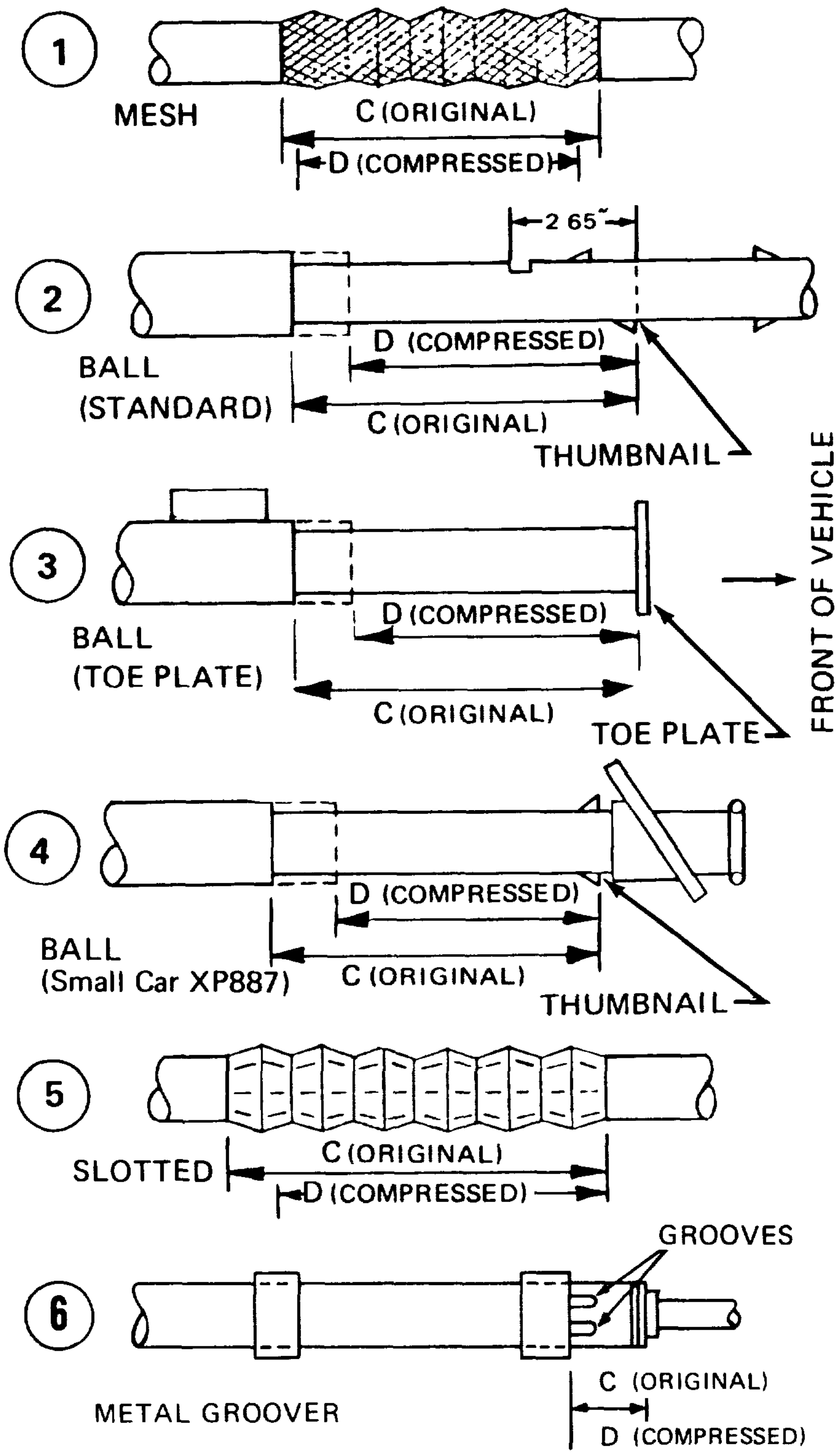


Figure 2-5 SKETCHES OF STEERING COLUMN ENERGY ABSORBING DEVICES
(SEE FIGURE 2-4 FOR LENGTH DIMENSIONS)

A shear capsule sketch is given in Figure 2-6. The capsule is intended to prevent movement of the assembly into the passenger compartment but to allow movement toward the engine compartment when impacted by the driver with a sufficient force to overcome the breakout force. Some vehicles have telescoping shaft units connecting the steering rod to the steering gearbox (Figure 2-7). If the front end crush causes the gearbox to move aft, some of the travel can be accommodated in this link rather than in aft motion of the steering column.

The Chrysler Corporation energy absorbing steering wheel is sketched on Figure 2-8.

Foreign Developments

Wilfert presented a paper in 1966 (Reference 2-6) that contained a description of the steering wheel energy absorber and the aft location of the steering gearbox on the Mercedes. A drawing of the current steering assembly (Reference 2-7) is reproduced as Figure 2-9. The telescoping steering and shift shafts are not coaxial. The energy absorber is just under the flat steering wheel. Note that the telescoping shafts are intended to limit intrusion and the action of the energy absorber is purposely made to be independent of these sliding sections.

Accles and Pollock in Britian developed a "shaftless" steering column. A sketch of the design (which appeared in Reference 2-8) is presented in Figure 2-10. The term "shaftless" designates a steering assembly in which the jacket transmits the steer torque and, of course, the gear shift mechanism is separated physically from the steering column. The shear capsules are identical with those used on U S. cars. Finally, the number of convolutions is large and the depth of the convolution is small. This column will be referred to again in the section on performance.

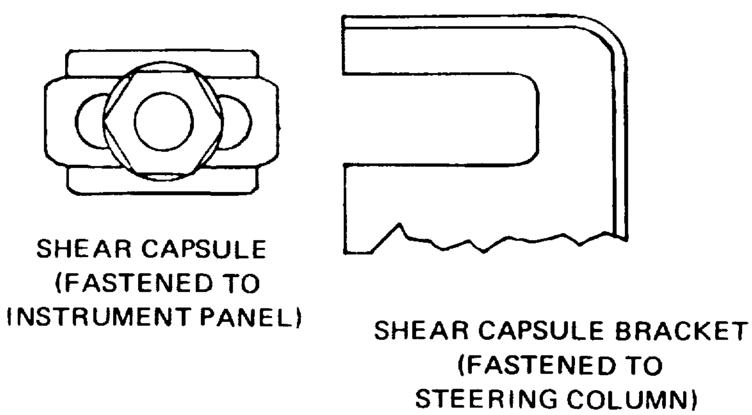
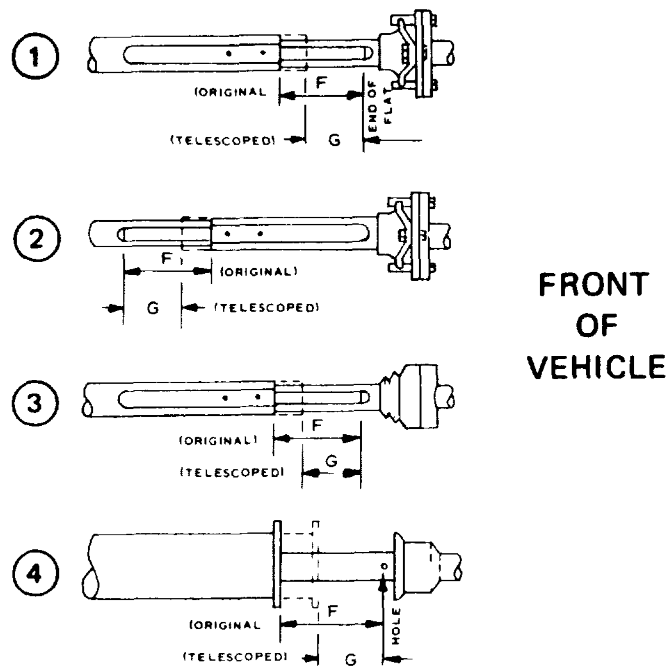
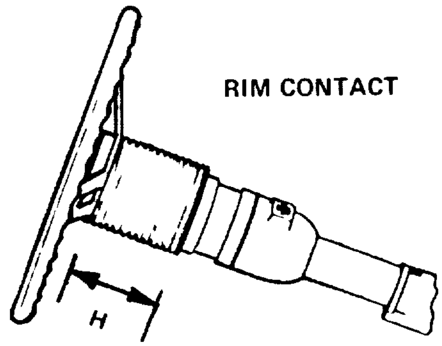


Figure 2-6 SHEAR CAPSULE ELEMENTS (ANTI-INTRUSION ELEMENT) (REFERENCE 2-5)



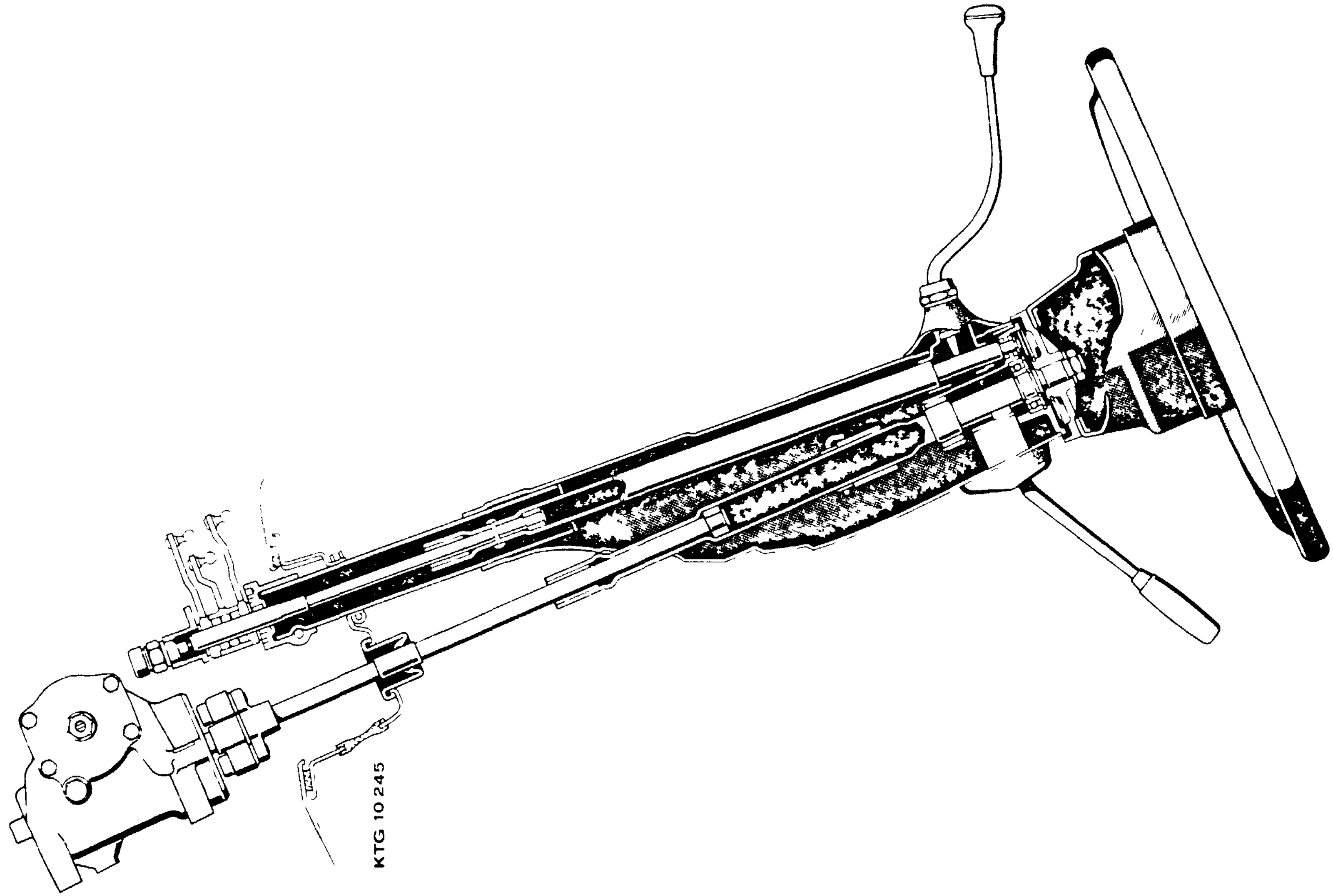
CORPORATION	MAKE	MODEL YEAR	TYPE OF UNIT	ORIG. LENGTH (F)
GM	CHEVELLE, EL CAMINO, TEMPEST, F-85, VISTA CRUISER, SPECIAL, SPORT WAGON	67	1	8.4
		68-70	2	5.74
	CAMARO FIREBIRD	70	2	5.74
	GM SMALL CAR (XP887)	70	3	6.2
AMC	AMBASSADOR	67	1	8.4
		68-69	2	6.1
		70	2	7.7
CHRYSLER	BARRACUDA CHALLENGER	70	4	10.8 (Manual) 7.6 (Power)

Figure 2-7 ENGINE COMPARTMENT TELESCOPING UNITS (FROM REFERENCE 2-5)



STEERING WHEEL ENERGY ABSORBING DEVICE TABLE			
CORPORATION	MODEL YEAR	MAKE	ORIG LENGTH (H)
CHRYSLER	70	BARRACUDA CHALLENGER	4.9

Figure 2-8 STEERING WHEEL ENERGY ABSORPTION



Sicherheitslenkung der Typen 200 D/220 D/200/220/230/250

Lenkgetriebe hinter Vorderachse angeordnet Lenksäule und Schaltrohr teleskopartig ineinander schiebbar Lenkrad mit Pralltopf und Polsterplatte

Figure 2-9 MERCEDES STEERING ASSEMBLY (FROM REFERENCE 2-7)

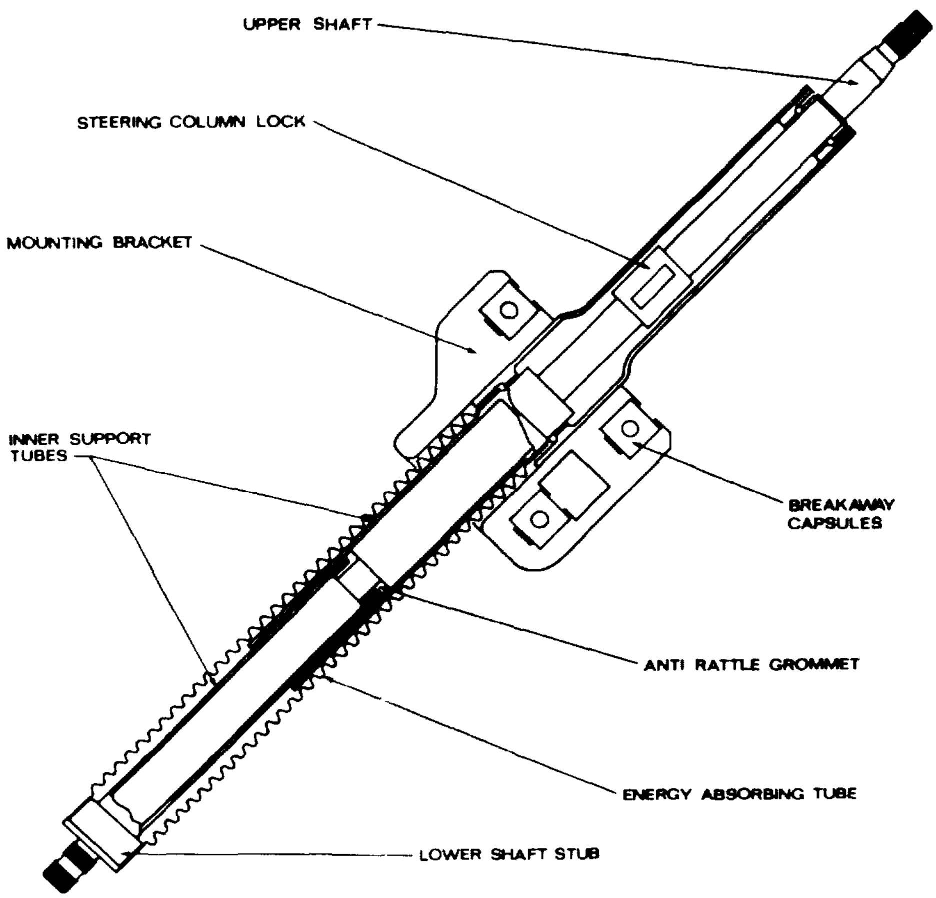


Figure 2-10 "SHAFTLESS" ENERGY ABSORBING STEERING COLUMN (FROM REFERENCE 2-8)

British studies (for example, Reference 2-9) emphasized the role of the longitudinal angularity of the steering column. In fact, in Reference 2-9, it was concluded that an optimum steering column angle existed and was greater than 40°. The reason for this conclusion was that the energy absorption was actually accomplished through a large bending deformation of the steering column as the driver forced the wheel over the dashboard. Use of this failure mode requires the imposition of large bending moments. These columns might be relatively ineffective if axial loading occurs.

Mr. Franchini reported on a steering column proposal by Fiat (Reference 2-10). Use was made of universal joints and plastic failure of supports and instrument panel during a deformation similar to that proposed by the British (see Figure 2-11). It should be noted, however, that the Italian proposal was aimed at the difficult problem of restraint for truck and van drivers.

A Japanese supplier (Reference 2-11) has developed an energy absorber based on frictional forces. In essence, the energy dissipation is accomplished by frictional forces brought into play when a section of the telescoping steering shaft is forced through tight collars. A sketch of this system is shown in Figure 2-12. Further information will be given in the section on performance.

Performance

Performance -- in the sense that FMVSS 203 is satisfied -- must be demonstrated with all U. S. current steering assemblies. Furthermore, FMVSS 204 intrusion compliance must be demonstrated also. It is to the details of EA performance in the laboratory that the rest of this discussion is devoted. Available field observations at accident scenes will be discussed briefly in a later section.

Force Characteristics

The peak forces and the energy absorbed by the various assemblies are roughly the same since the manufacturers are attempting to satisfy

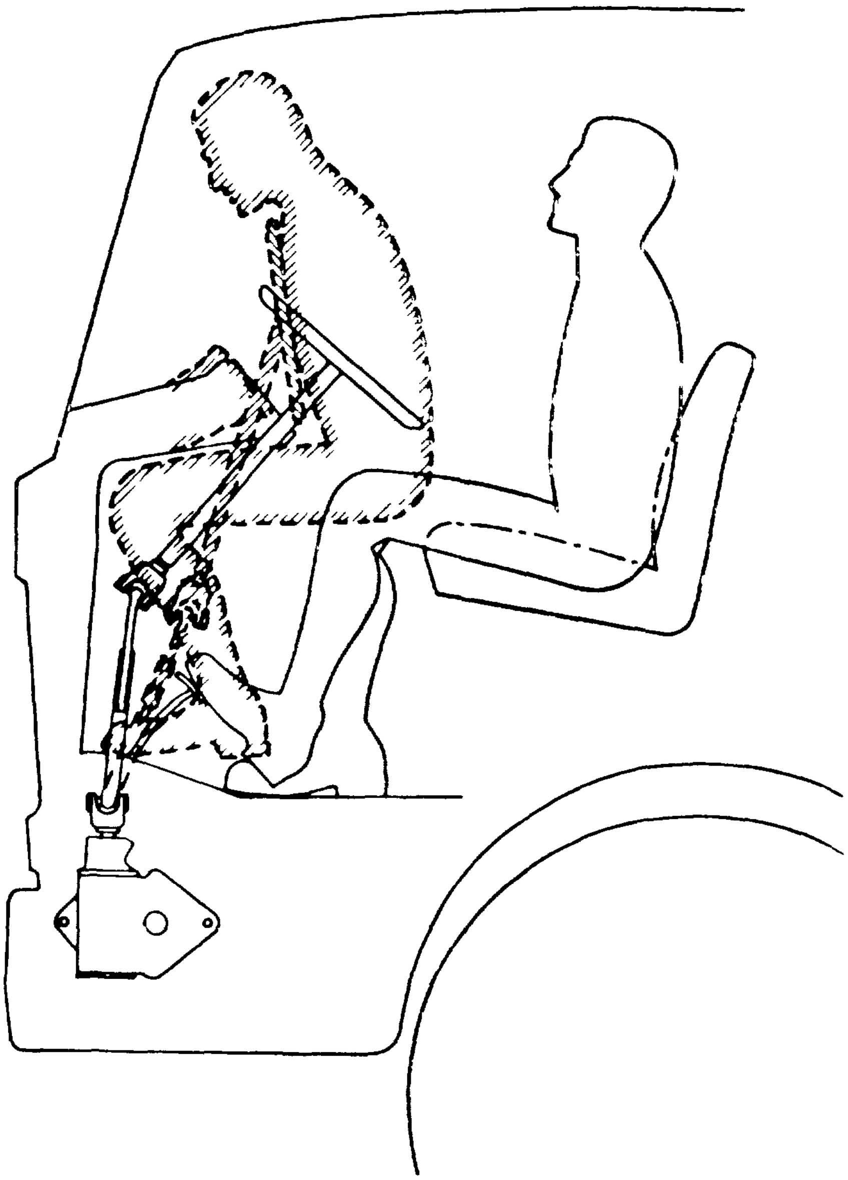
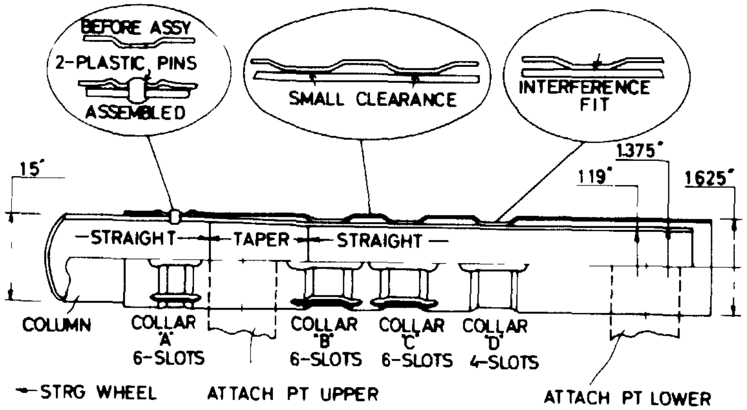
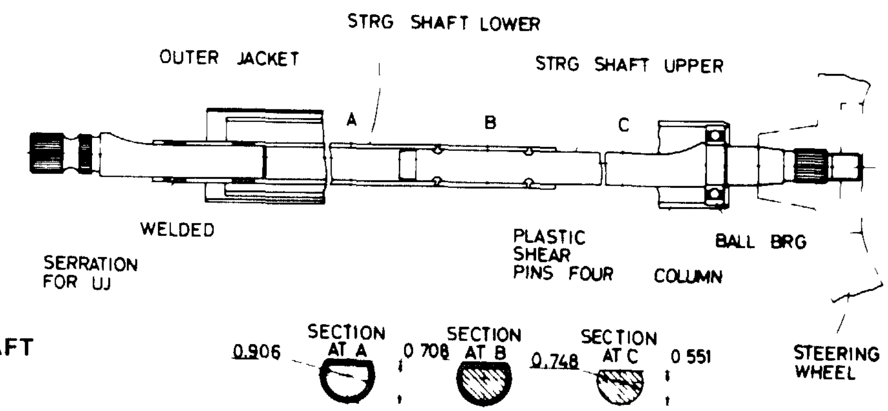


Figure 2-11 STEERING WHEEL WITH FORWARD-TILTING COLLAPSIBLE COLUMN (IN DIFFERENT SECTIONS, WITH TWO ARTICULATED AND ONE SLIP JOINT INTERPOSED) (FROM REFERENCE 2-10)



OUTER JACKET
AND COLUMN



STEERING SHAFT
ASSEMBLY

Figure 2-12 FUNCTIONAL COLLAR DESIGN – ASSEMBLY OF COLLARS, JACKET, COLUMN, AND STEERING SHAFT (TAKEN FROM REFERENCE 2-11)

FMVSS 203. Differences arise in the waveform. U. S. columns tend to have peak force values early in the collapse history; a recent British column shows an increasing force with increasing deflection and the Japanese have presented data on an EA assembly that shows two force peaks.

The static force-deflection curve for the early mesh or Saginaw EA jacket, presented in Reference 2-12, is shown in Figure 2-13. Successive shears of brackets and the plastic pins holding the telescoping sections together is indicated along with the buckling of the convoluted mesh sections. Under dynamic loading, the collapse force is increased and the individual events are less identifiable. This can be seen in the sled test result shown on Figure 2-14.

General Motors' second generation configuration was the ball EA column in which ball bearings groove the jacket metal shafts as the shafts telescope (Reference 2-13). Detailed studies of the collapse sequence were made and some of the results are reproduced in Figures 2-15, 2-16, and 2-17.

The Ford slotted convoluted EA assembly has characteristics similar to those of the G. M. assembly.

The Cortina column, developed in Britian, has an interesting force-deflection curve in that the force is supposed to increase almost linearly with deflection beyond an initial deflection of about 3-1/2". The reported performance curve (Reference 2-8) is shown in Figure 2-18.

Japanese data presented in Reference 2-11 illustrates the force-deflection and force time performance of the slotted collar configuration. A typical load deflection curve is shown in Figure 2-19. Their reported results also indicate that manufacturing imperfections can induce performance variability (see Figure 2-20).

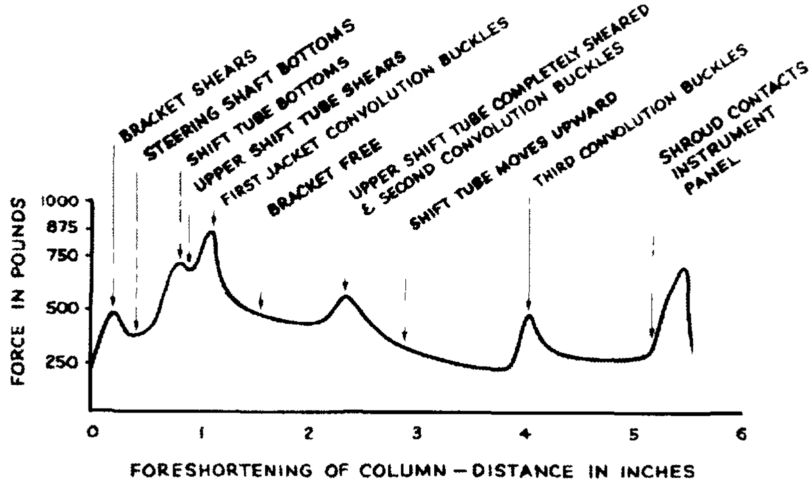


Figure 2-13 SEQUENCE OF LOADING AS COLUMN FORESHORTENS IN TENSILE MACHINE – G.M. MESH COLUMN, (REFERENCE 2-12)

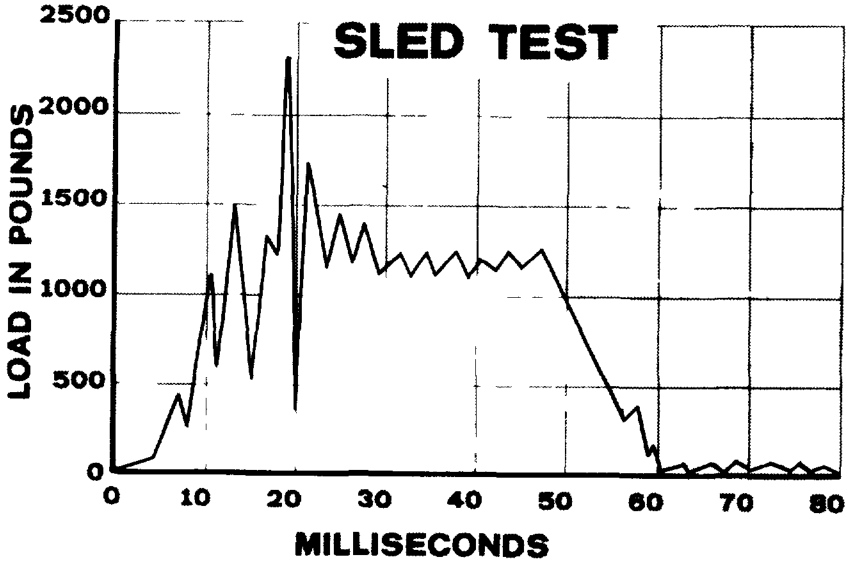


Figure 2-14 DYNAMICS CHARACTERISTICS (SLED TEST) OF G.M. MESH COLUMN (REFERENCE 2-12)

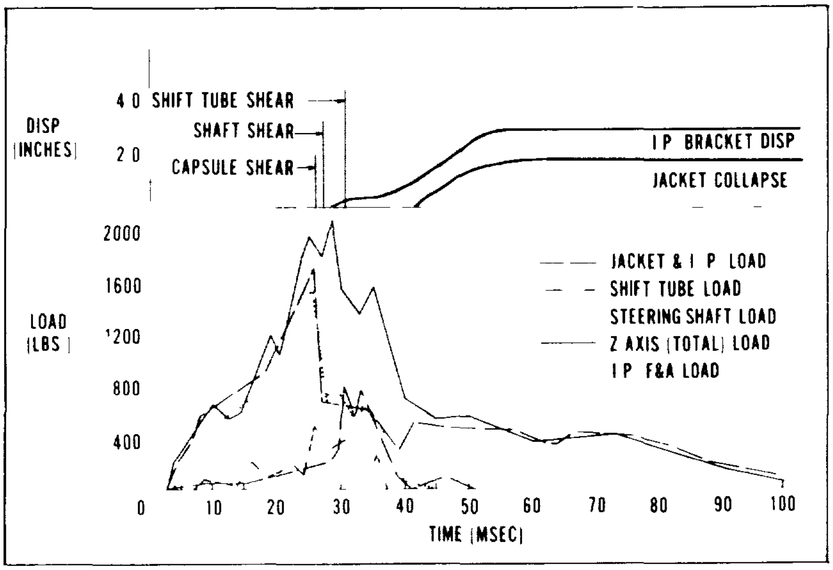


Figure 2-15 LOADING SEQUENCE – G.M. BALL ENERGY ABSORBING COLUMN (REFERENCE 2-13)

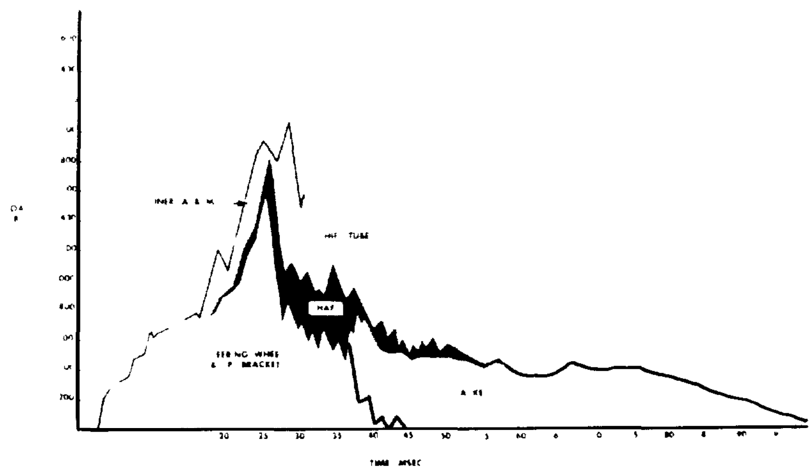


Figure 2-16 LOADING COMPONENTS – G.M. BALL ENERGY ABSORBING COLUMN (REFERENCE 2-13)

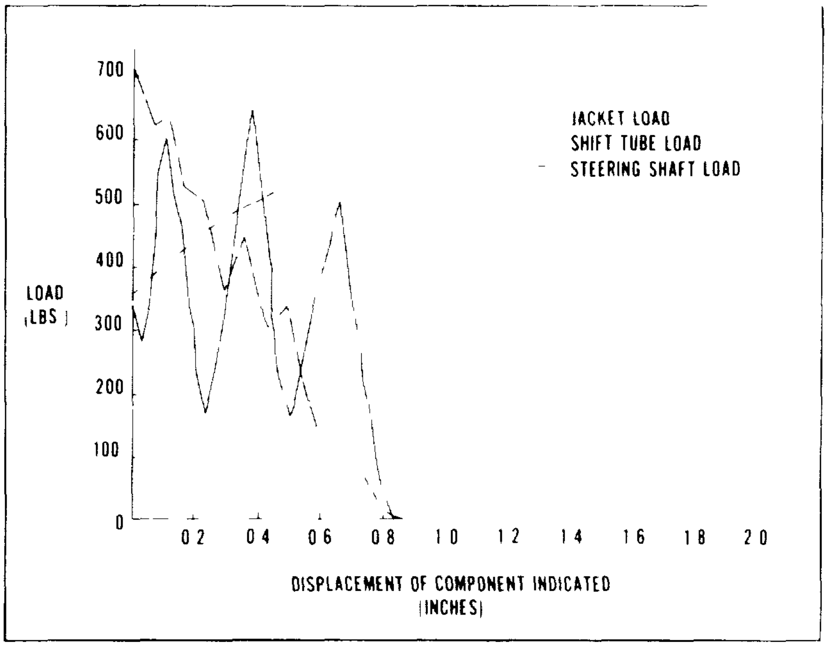


Figure 2-17 LOAD-DISPLACEMENT CHARACTERISTICS OF BALL ASSEMBLY COMPONENTS DURING COLLAPSE (REFERENCE 2-13)

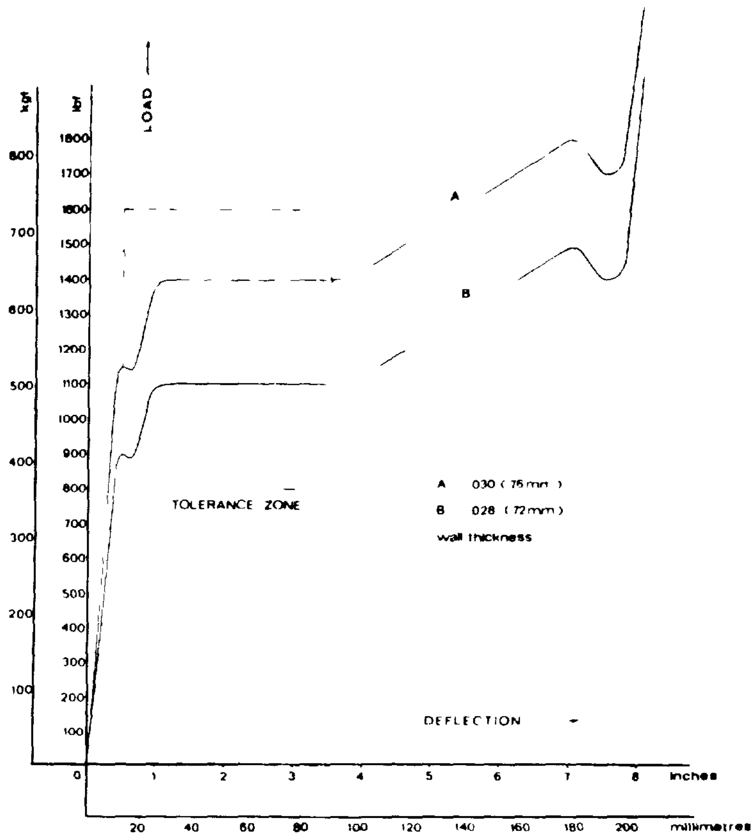
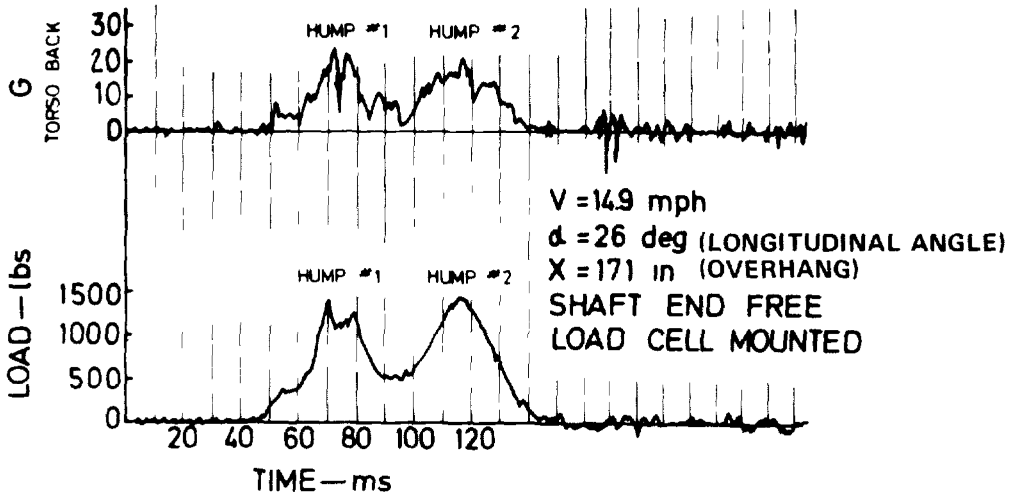


Figure 2-18 LOAD-DEFLECTION CURVES FOR "SHAFTLESS" CORTINA EA ASSEMBLY (TAKEN FROM REFERENCE 2-8)



EXAMPLE OF SLED TEST RESULT

Figure 2-19 FORCE TIME HISTORY FOR JAPANESE COLLAR STEERING ASSEMBLY (REFERENCE 2-11)

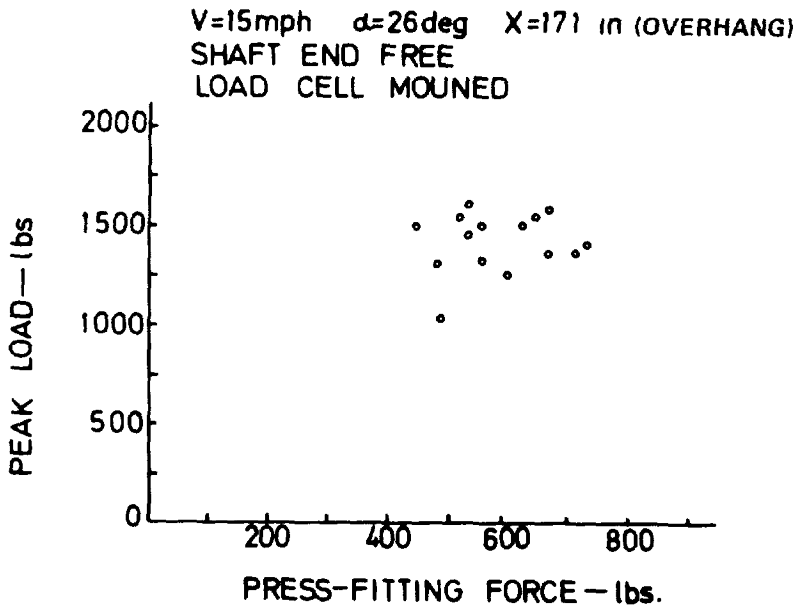


Figure 2-20 EFFECT ON PEAK LOAD OF PRESS-FITTING FORCE TO ASSEMBLE COLUMN INTO JACKET (REFERENCE 2-11)

Angularity and Overhang

A systematic study of the effects of longitudinal angularity was reported by the Japanese (Reference 2-11). Their results showed that longitudinal angularity, overhang, friction coefficient of sliding parts, and stiffness of the steering wheel rim changed the force levels at which the deceleration of a body block was accomplished. These physical parameters in combination determine, apparently, the contribution of the friction of sliding parts to the total force level. Angularity, overhang, and steering rim stiffness govern the applied moments which, in part, are reacted as normal forces at the sliding contact surfaces. Figure 2-21 shows the effects of lubrication and longitudinal angularity on the Japanese assembly performance. Figure 2-22 shows the effect of overhang. Figure 2-23 illustrates the effect of mounting angle on peak load and collapse distance.

The data on longitudinal angularity and overhang (moment) effects suggests that load control problems would exist also for lateral offset and lateral angularity of applied loads. This is substantiated by the Cornell Aeronautical Laboratory sled tests reported in Reference 2-14. Dummies impacting a 1968 Chevrolet EA column during 10 g and 20 g sled pulses did not deform the EA mesh when the impact direction was $\pm$ 30° from head-on.

Steering Wheel

The geometric layout of the steering wheel, as well as its construction influences the energy absorption and effectiveness of the steering assembly. This was illustrated by the British, who present acceleration time histories for two body block tests in which only steering wheel orientation was changed.

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*Lap belted, 50% male dummy.

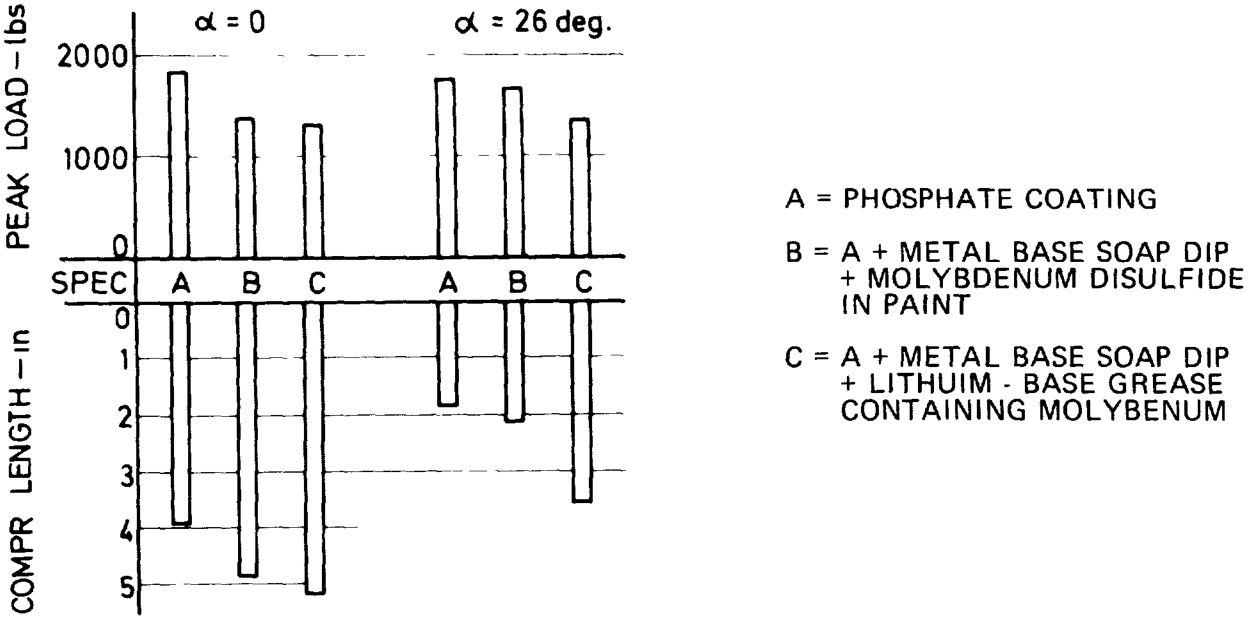


Figure 2-21 EFFECTS OF LUBRICATION AND LONGITUDINAL ANGULARITY ON JAPANESE STEERING COLUMN IMPACT FORCE CHARACTERISTICS (REFERENCE 2-11)

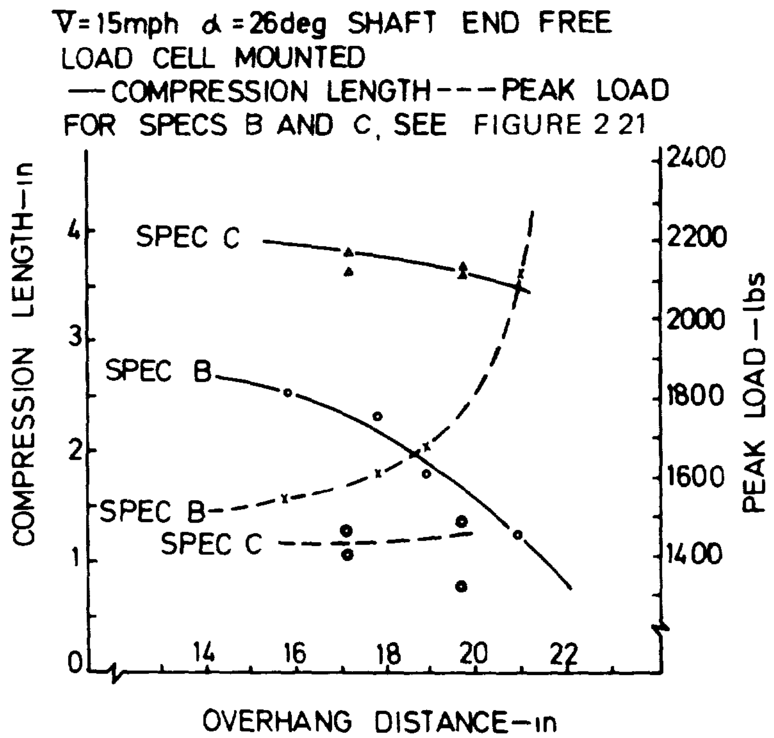


Figure 2-22 EFFECT OF OVERHANG LENGTH ON COMPRESSION LENGTH AND PEAK LOAD FOR TWO SURFACE TREATMENT SPECIFICATIONS (REFERENCE 2-11)

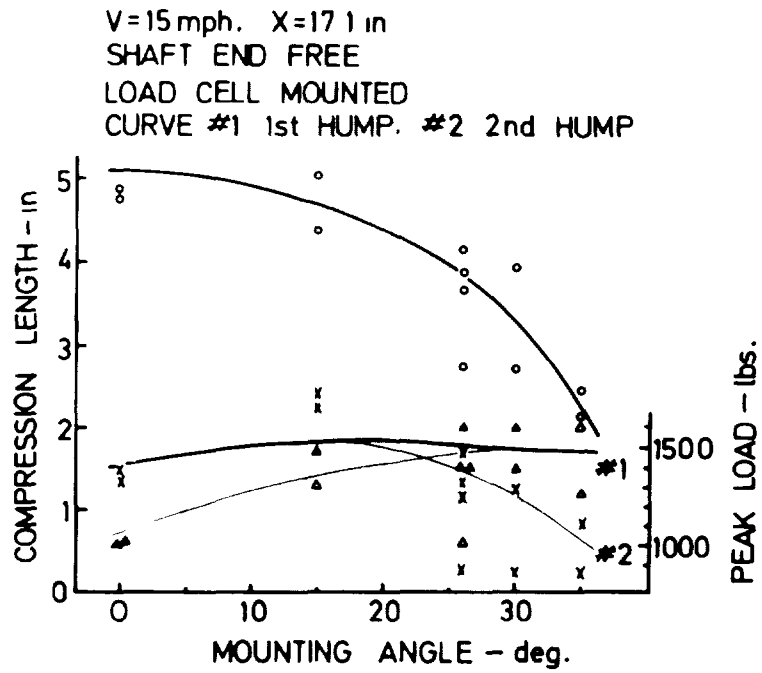


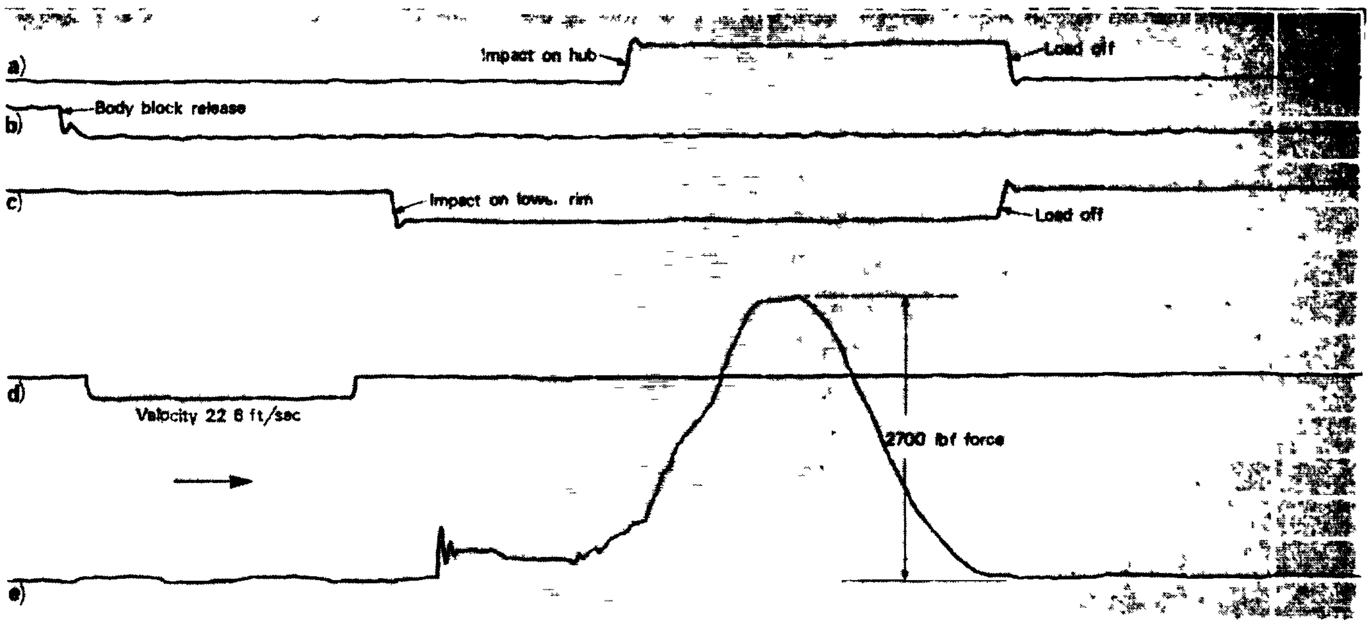
Figure 2-23 EFFECT OF MOUNTING ANGLE ON COMPRESSION LENGTH AND PEAK LOAD (FROM REFERENCE 2-11)

Their force time histories are shown in Figure 2-24. When the two spokes were aligned vertically, the initial force peak occurred earlier in the collapse and the peak load was half that observed when the spokes were horizontal (Reference 2-15).

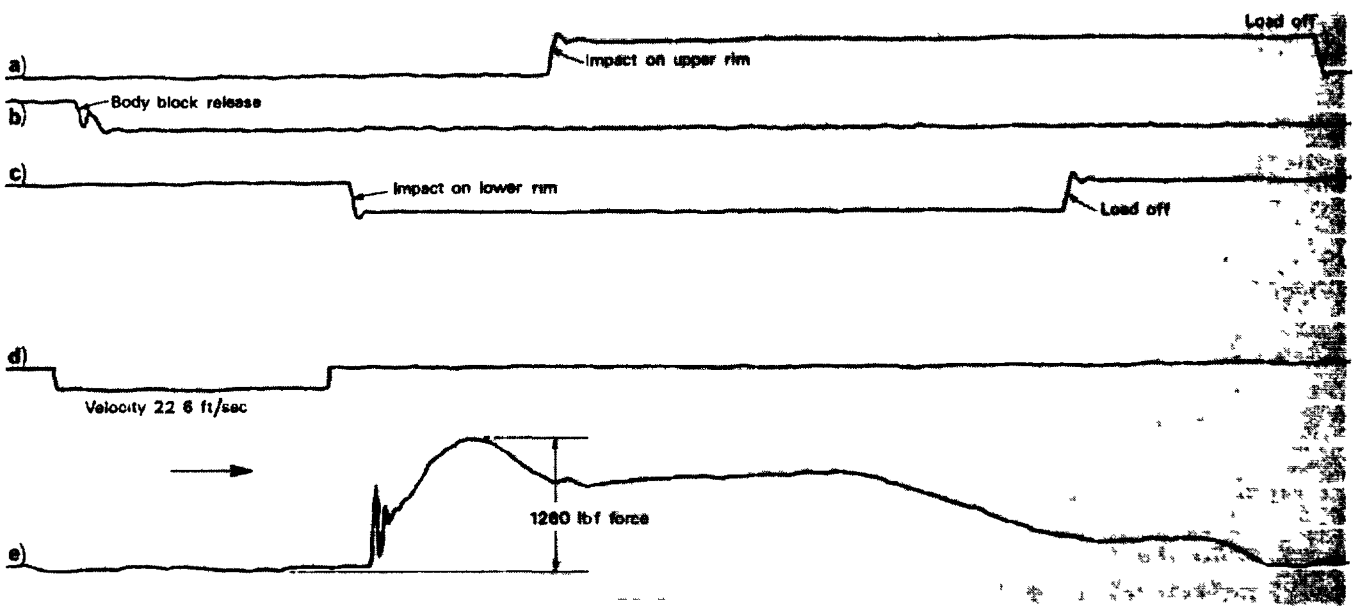
A rigid steering wheel was used in some of the Japanese tests at a longitudinal mounting angle of 26° (Reference 2-11). Performance comparisons in compression length and peak body block torso accelerations are shown in Figure 2-25. The indicated plus symbols represent the rigid wheel performance and the circles, triangles, and x symbols represent the corresponding standard wheel performance at various mounting angles.

Field Observations

Both statistical analysis of accident reports and field investigations of individual accidents indicate that the EA steering column has reduced driver fatalities (e.g., References 2-16, 2-17 and 2-18). Performance measured in the field in terms of collapse distance vs. estimated impact speed, seems to vary widely -- see Figure 2-26. Obviously there are effects of differences in driver restraint use and driver mass. In addition, however, there are factors such as the initial separation distance between driver and wheel and the offset or obliqueness of torso-wheel impact.



WHEEL SPOKES HORIZONTAL



WHEEL SPOKES VERTICAL

Figure 2-24 EFFECT OF STEERING WHEEL SPOKE ORIENTATION ON BODY BLOCK RESPONSES (TAKEN FROM REVERENCE 2-13) - 2 SPOKES

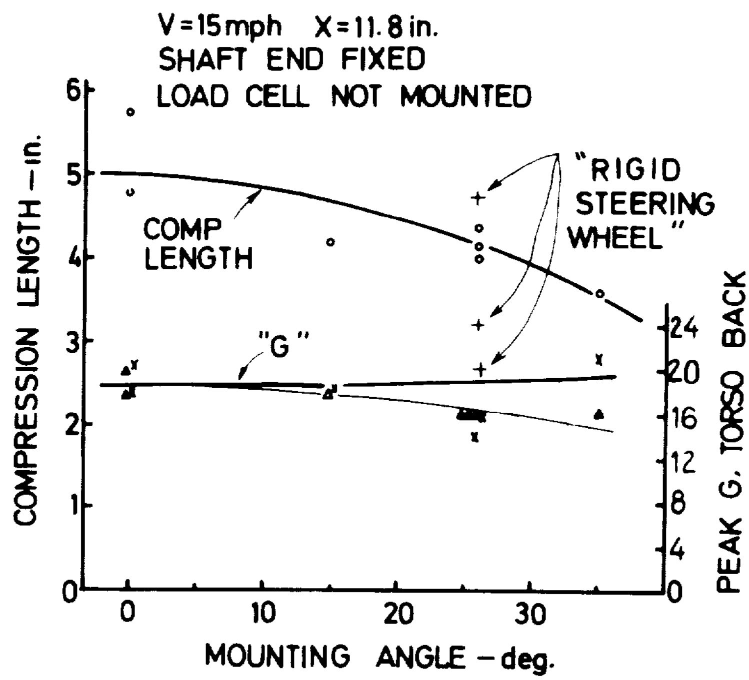


Figure 2-25 EFFECT OF RIGIDITY OF STEERING WHEEL ON PERFORMANCE

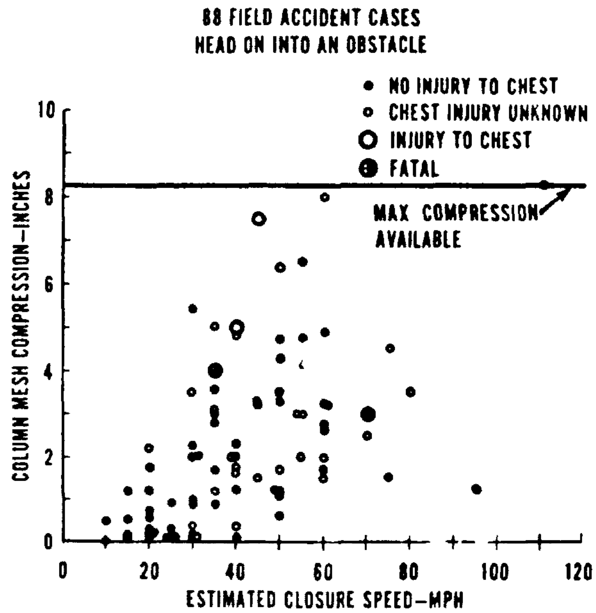


Figure 2-26 EA STEERING COLUMN PERFORMANCE MEASURED IN ACCIDENTS (REFERENCE 2-19)

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APPENDIX 3

INTRUDER/ABSORBER

Maintenance of passenger compartment integrity in high speed impacts (say, greater than 45 mph) implies energy absorption primarily in the frontal structure. Stroke limitations, on the other hand, dictate high stroke efficiency (rapid onset) and high force levels. The onset increase enhances the ridedown of the occupant but the high deceleration level causes a loss of ridedown compared to conventional structures. Use of an air cushion passive restraint system for the driver decreases occupant separation from the "structure" and, hence, offers some ridedown advantages in addition to its load distribution and damping gains. A balance between cushion size and actuation time must be made, however, and current air bags exhibit a relatively slow build-up of forces and, hence, poor stroke efficiency. The intruder/absorber is intended to assist the air bag by furnishing an improved stroke efficiency and an incremental force level that may be required at high speeds.

The device is a cylindrical hydraulic strut. The moving part is deployed by the burning of an 8 grain squib very much like an air cushion squib. Activation would be initiated by a crash sensor in the same way an air cushion is activated. It is hypothesized that the the device would be used in conjunction with a toroidal air cushion - the cushion providing load distribution on the driver thorax and head and the intruder applying loads to the chest.

The deployable part of the intruder weighs about one-half pound and, hence, inertial forces during extension are relatively low. The absorber stroke provides a force of about 1800 lbf for the first 5" and around 4000 lbf peak for the final 2" (see Figure 17 in the body of the report).

The plots on the following pages contain data on tests made by the manufacturer. They are:

Plot	Title
3-1	Peak Absorber Force During First Five Inches of Stroke.
3-2	Peak Absorber Force During Final Two Inches of Stroke.
3-3	Change of Stroke with Angularity.
3-4	Deployment Time as a Function of Squib Size.

The scatter of deployment times with squib size can probably be reduced by requiring closer tolerances on performance or by using two squibs with a total change of about 10 grains. (The slower deployment at larger charges is believed to be the result of "quenching" - i.e., the breaking off and extinguishing of a part of the explosive.)

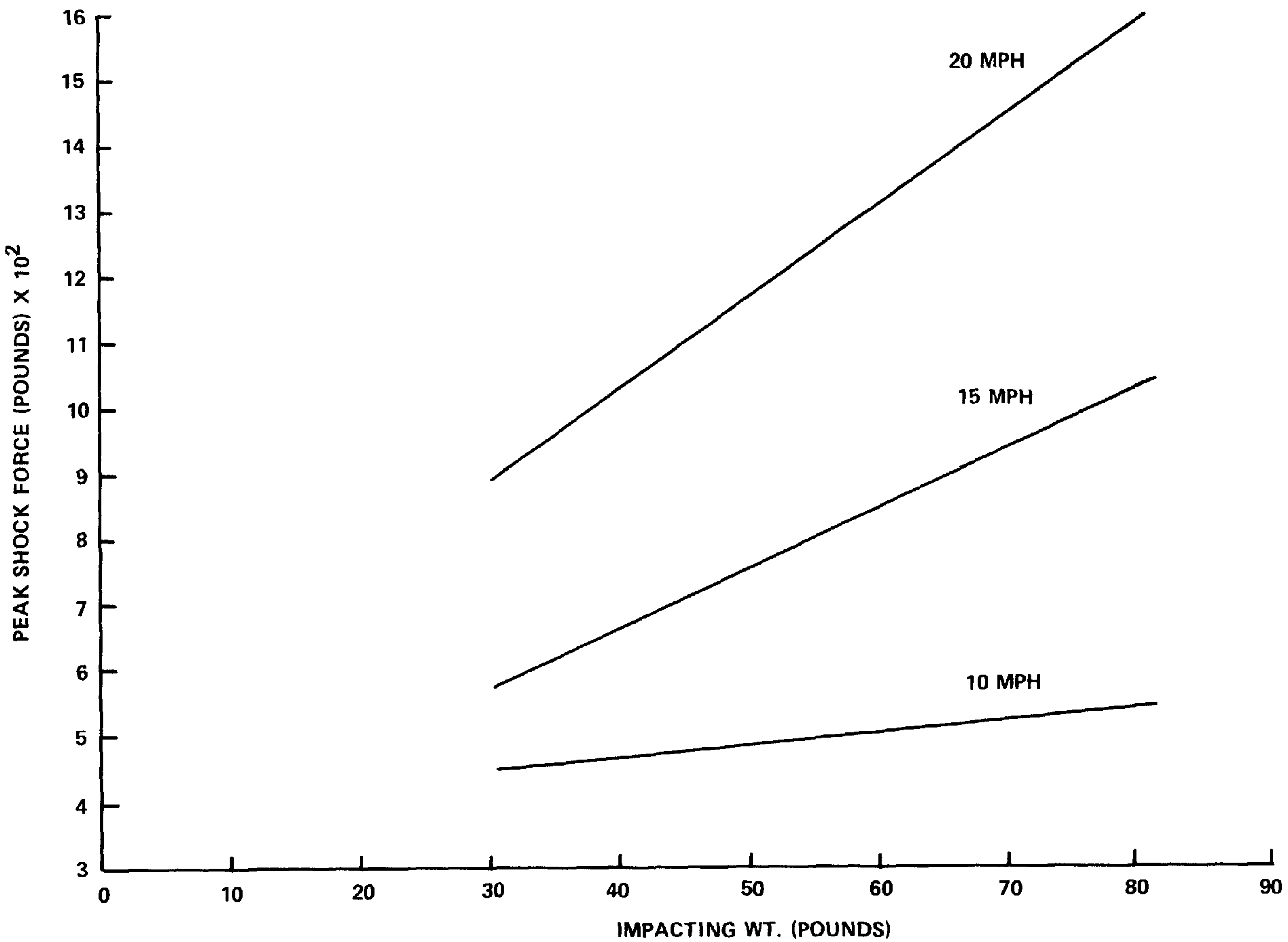

Figure 3-1 PEAK ABSORBER FORCE DURING FIRST FIVE INCHES OF STROBE

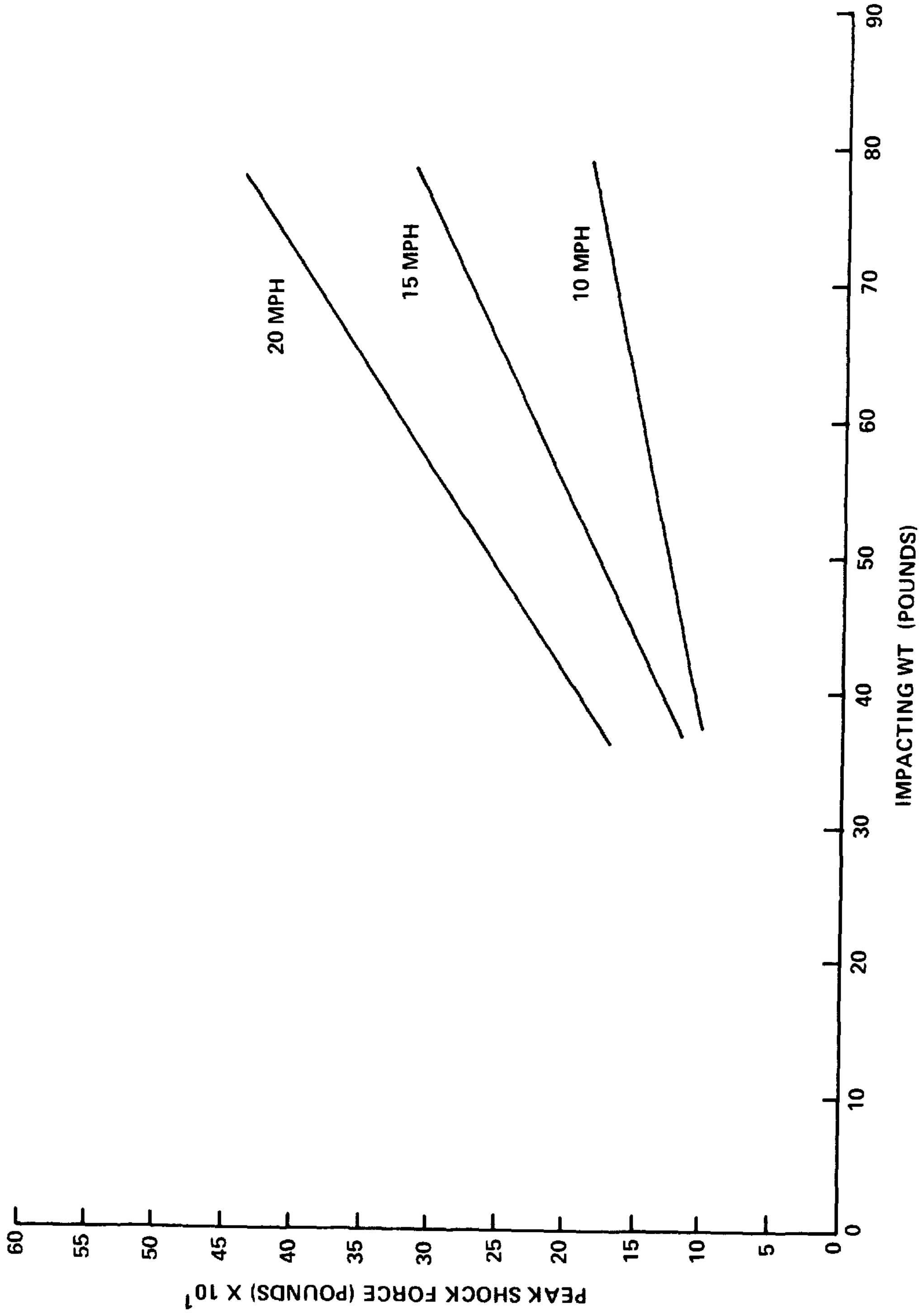

Figure 3-2 PEAK ABSORBER FORCE DURING FINAL TWO INCHES OF STROBE

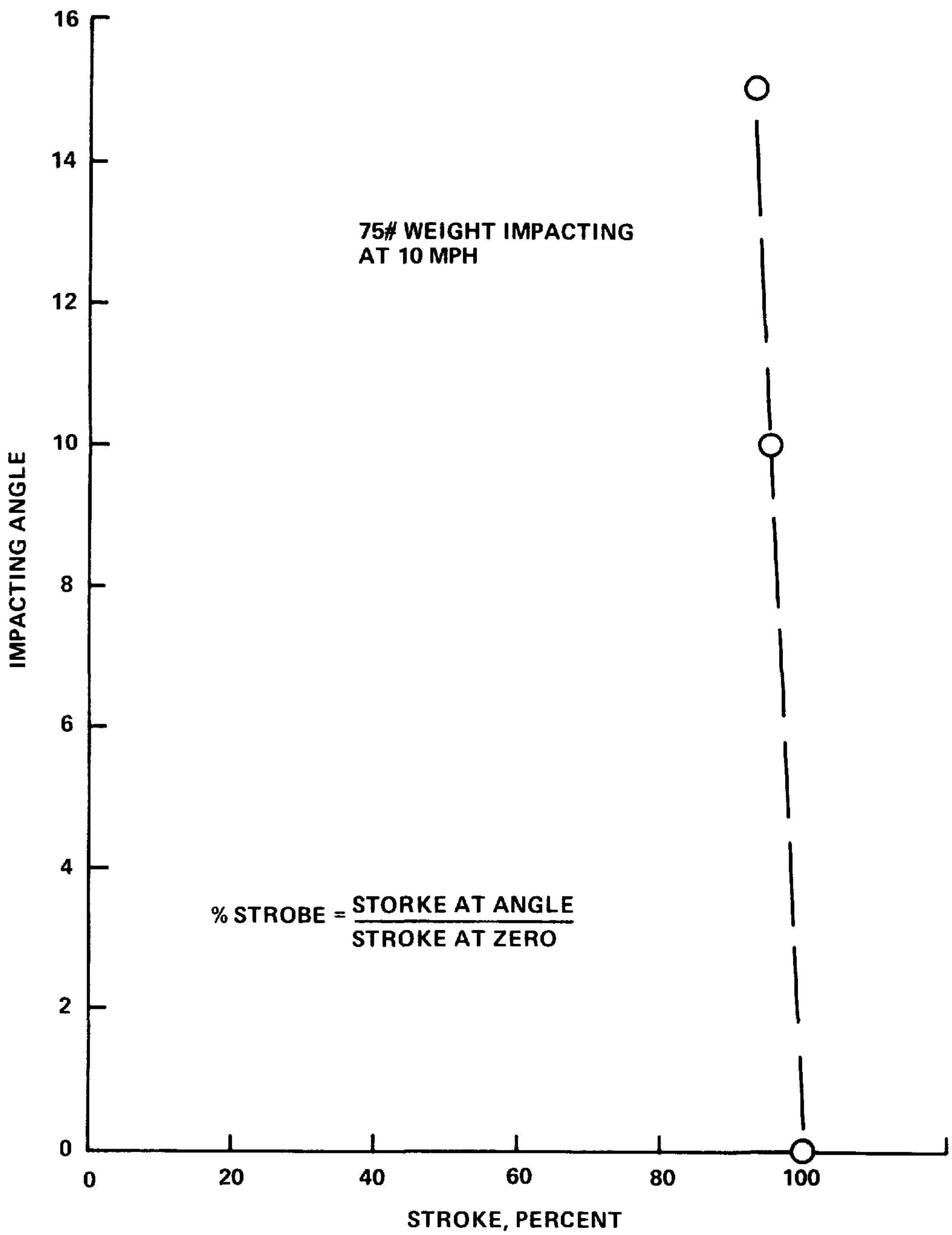

Figure 3-3 CHANGE OF STROKE WITH ANGULARITY

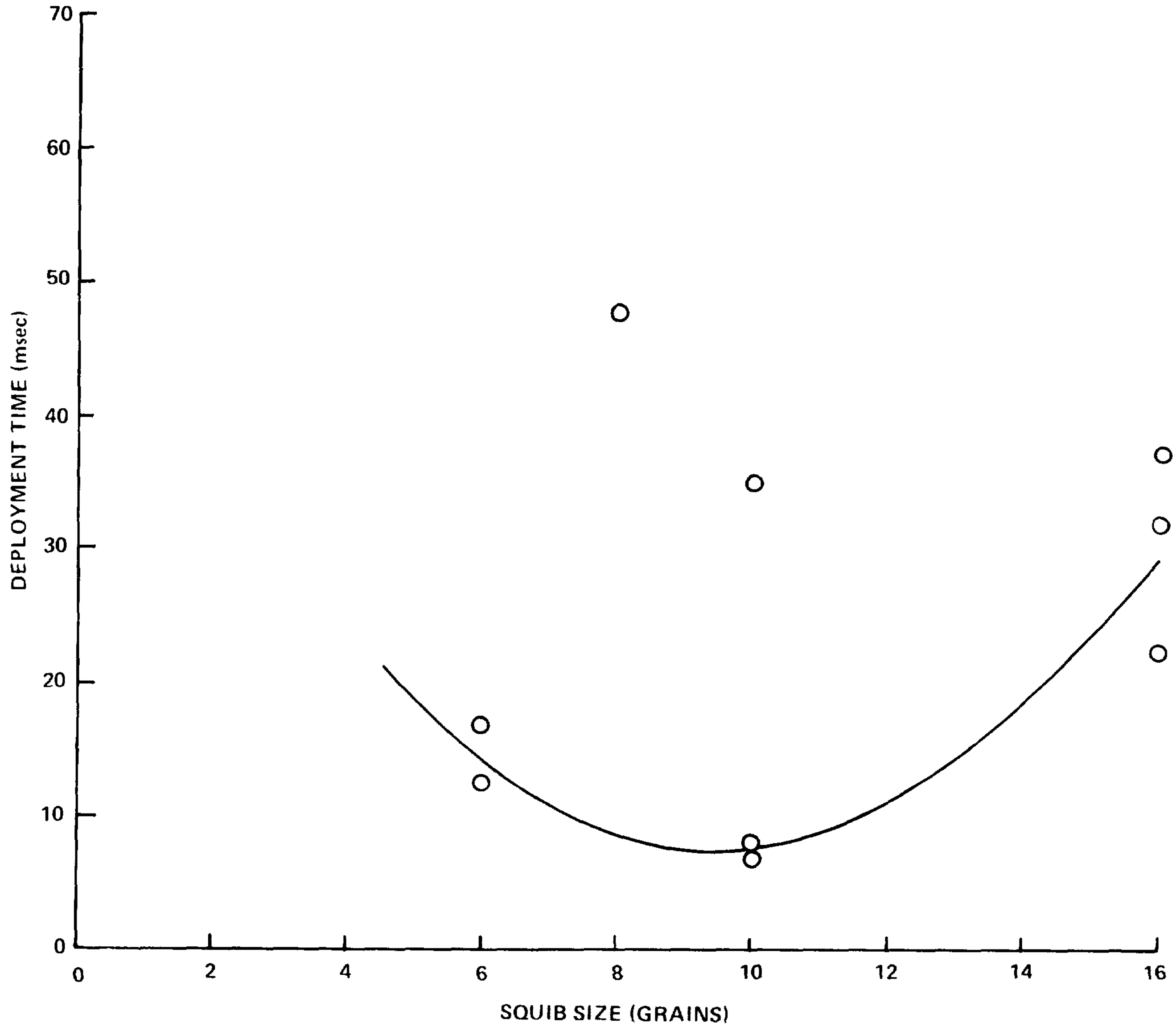

Figure 3 4 DEPLOYMENT TIME AS A FUNCTION OF SQUIB SIZE