SECTION 3

CRASHWORTHINESS SEMINAR

Part 1 – Introduction

Mr. Edward M. Chandler, *Chairman*

Part 2 – Structures

Dr. H. Appel, *Germany, VW*
Mr. William J. Wingenbach, *USA, AMF*
Dr. H. Appel, *Germany, VW*

Part 3 – Restraints and Occupant Simulation

Dr. G. E. Sutherland, *USA, Rocket Research*
Dr. H. P. Willumeit, *Germany, VW*
Dr. Reidelbach, *Germany, Daimler-Benz*
Dr. M. A. Macauley, *United Kingdom, Motor Research*
Mr. J. Leroy, *France, O.N.S.E.R.*
(Presentation not available)

Part 4 – Subsystems, Testing and Other Considerations

Mr. Robert Schwarz, *USA, AMF*
Mr. R. Van Laethem, *Belgium, Glaverbel*
Mr. C. R. Ennos, *United Kingdom, Ford*
Dr. C. Tarriere, *France, Renault*
Mr. W. Rosenau and Mr. U. Seifert, *Germany, VW*

SECTION 3

PART 1 INTRODUCTION

Chairman:
Mr. Edward Chandler
Office of Experimental Safety Vehicle Programs, National Highway Traffic Safety Administration, United States Department of Transportation

Good afternoon. It is a real pleasure to take part in another open forum on crash injury reduction with the participants in the International ESV Program. We were very pleased with the discussion sessions in Paris at the First International ESV Conference. It has been seldom, if ever, that so large a number of people have focused so effectively on the technical aspects involved in crash energy management as was done at this meeting. Also, we felt that the discussions were carried on frankly and with candor. We are sure that the seminar today will proceed in the same manner and again update our understanding of the broad fundamental problems involved in reducing injuries and fatalities from automobile crashes. In addition, we believe that this session will give us a better understanding of some of the special problems encountered in this field.

I will briefly review the discussions in Paris where the United States used its performance specifications for the Family Sedan to structure the proceedings. Our position was then, and is now: 1) that we were presenting a first attempt at formulating such performance specifications, 2) that we recognize the need for further study and possible revision of some of those specifications as feasibility studies proceed, and 3) that we invite your comments and constructive criticism. More important, however, was our desire to make the specifications available to other nations, to be used to the extent they are applicable, as a base from which to formulate your own requirements. At the same time, we certainly have profited from the feed-back we have received from you.

It is our intention that this seminar shall be of the same general character as the Paris discussions, but with a somewhat revised format. By this, we mean that rather than using our specification for format, we have selected certain topics for discussion that we believe present common problems to us all. These suggested topics were sent out in the proposed agenda and, with minor changes in order to group them, appear in the final agenda. If we have inadvertently omitted some topics of broad interest, we invite you to introduce them, to the extent that time permits.

With regard to time, it appears that we may be pressed. We have 13 papers that have been submitted for this seminar and we have divided them into three groups as shown on your agenda for this session. One hour has been allowed for each session with a 15-minute break between sessions. With this many papers, we can only allow ten minutes per paper if we are to provide any time for discussion. Only three papers have been scheduled for the first session, but if we can save some time on this session, we will move onto the next session. By the same token, if we lose time here, we will have to make it up later. Therefore, I may be forced to call for some speakers to conclude their presentations earlier than they wish. All papers that were submitted will be published in the Report of the proceedings of this Conference, even if we are able to spend only a little, or possibly no, time on some of them.

SECTION 3

PART 2 STRUCTURES

BENEFIT/COST ANALYSIS FOR EVALUATION OF ESV IMPACT TESTS PROPOSAL FOR REDUCED REAR END IMPACT SPEEDS

Dr. Hermann Appel, Germany VW

Survey

Benefit/cost considerations show that ESV crash tests are not equivalent in their relation to each other. For the design of the rear structure in ESV terms the benefit/cost factor is considerably smaller than for the front and side structure.

A reduction of the relative impact speed front-to-rear from 75 mph to 50 mph results in a better matching of the benefit/cost factors and thereby in a more significant balancing of the test conditions.

The present derivation has only the character of a model, since the statistical results of the accident research entering the benefits are subject to criticism and the cost estimates are based on coarse assumptions. Absolute values for benefits costs, or benefit/cost factors are of lesser importance than relative relations.

1. Benefit/Cost Studies

The question whether the essential tests for testing the vehicle structure established by the USA Specifications.[1]

50 mph frontal collision against fixed barrier
75 mph front-to-front collision
30 mph front collision against side
75 mph front collision against rear

are meaningful in relation to each other, can apparently be answered only be means of benefit/cost studies. The starting point is the present condition, that is, the present construction of the vehicles to which the results of statistic accident research refer. If the improvements of front, side and rear structures required to meet the specified tests are always resulting in the same benefit/cost factor, the assumption may be made that the tests are meaningful in relation to each other. If not, the use of an additional cost unit in one structural member would bring more effectiveness than the same cost unit when used for another structural member.

While the expenses (cost) for a given improvement of the structure can be calculated, the determination of the respective effectiveness must be based on statistical results of accident research.

This procedure is connected with some uncertainty for the following reasons:

a. the discrepancy of tests and real accidents;
b. the inaccurately estimated impact velocities;
c. the assumption that the future traffic-system will be similar to that today.

A few significant results are shown below:

1. More than 50% of accidents are vehicle-vehicle collisions[2,3] refer to Figure 1.

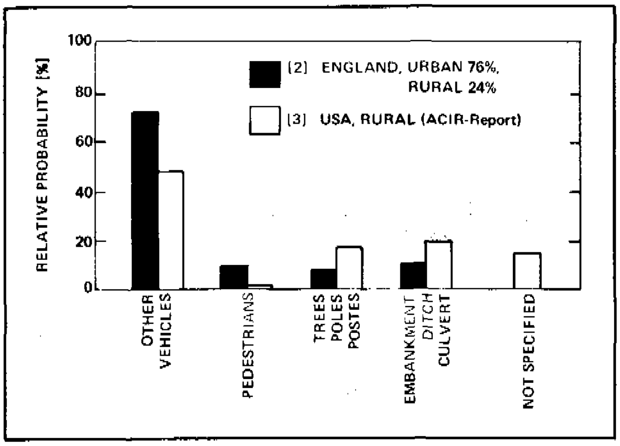

Figure 1

2. Approximately 50% of all serious accidents are front impacts, 25% are side impacts, 15% are rollovers and 2% are rear impacts[4], refer to Figure 2. In cities, the share of front impacts drops to 35%, the share of side impacts increase to 35%[5].

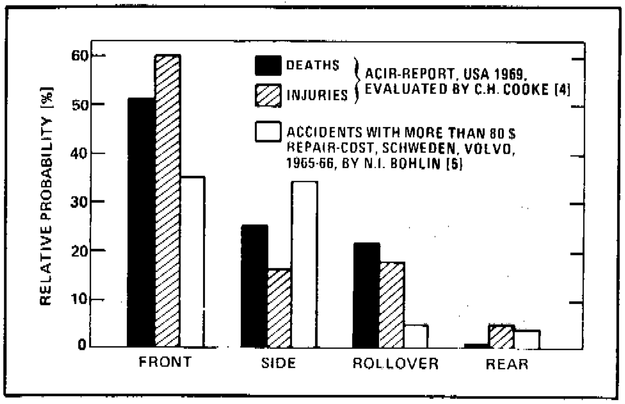

Figure 2

3. In 50% of all accidents the impact speeds are lower than 25 mph (city traffic[2,5]) or 45 mph (rural traffic[2]), refer to Figure 3.

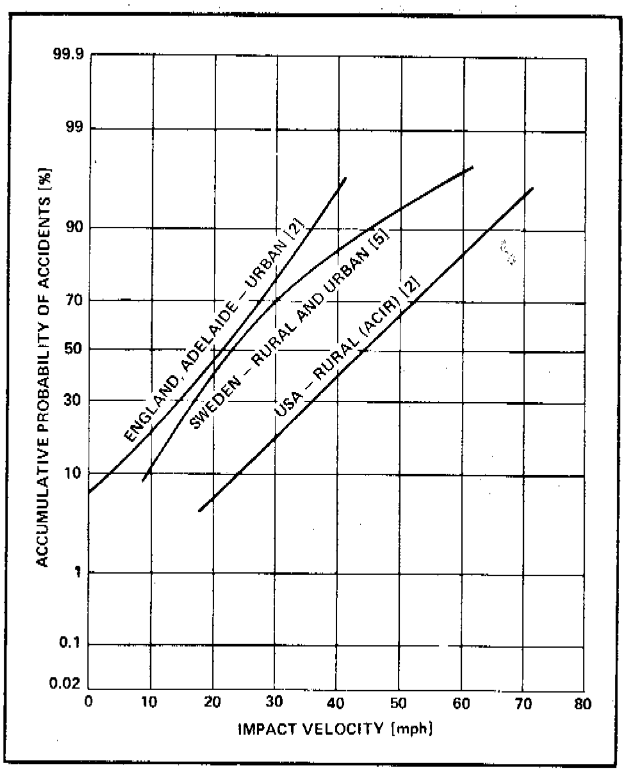

Figure 3

4. At a given impact speed the risk of injuries and death increases from the single-vehicle-side-impact via the single-vehicle-front-impact and the vehicle-to-vehicle-front-impact to the most dangerous type, the vehicle-to-vehicle-side-impact[4], refer to Figures 4 and 5.

In view of the ESV tests Figures 4 and 5 provide the following information:

1. Protection up to 30 mph against vehicle-to-vehicle-side-impacts would save 55% of (for this type of accident) the injured and 38% of the dead.

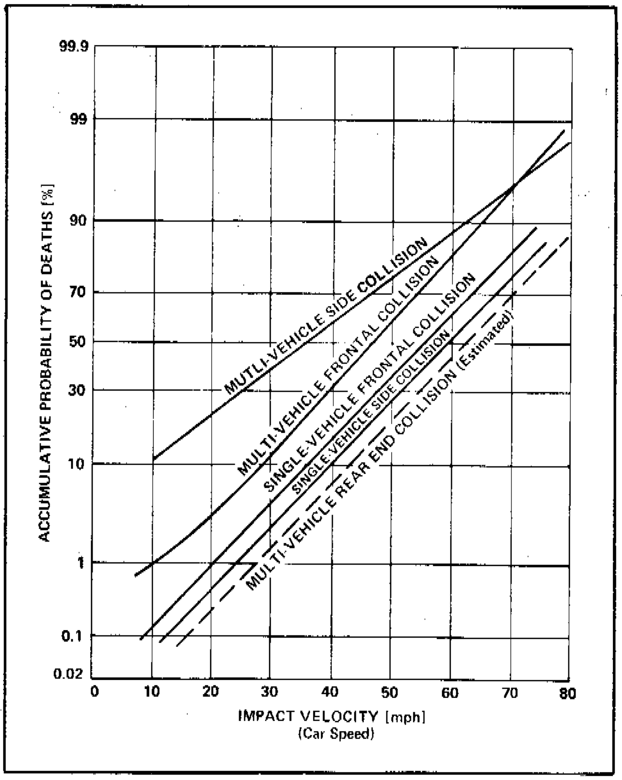

Figure 4

2. Protection up to 75 mph relative speed (that is 37.5 mph vehicle speed) at vehicle-to-vehicle-frontal-collisions would save 50% of the injured and 25% of the dead.
3. Protection up to 50 mph for single-vehicle-frontal-collisions would save 66% of the injured and 35% of the dead.

The surprising part of these results is the low percentage of effectiveness expressed in terms of persons injured and dead in view of the relatively high impact speeds.

As an example, Bohlin[5] says that the use of safety belts alone in the Volvo P 11 and P 12 reduced the fatality rate by 82% and at speeds of 60 mph even by 100%!

The statistical data[4] for the above figures appear in fact not too plausible and in need for a thorough checkup. An essential reason for the discrepancy may be the fact that the statistics[4] include a considerable share of eccentric, angular impacts, while the estimates of the effectiveness are referring to a straight, central impact, a much sharper test condition. This discrepancy is assumed to have no influence on the following information, since only relative relations are of importance.

Statistical results concerning the interrelation of probable injuries and fatal accidents in dependence of the impact speed during rear collisions are not given in[4]. On the basis of the results shown by Cooke[4] the upper limit of the estimates is that up to a rear impact speed of 75 mph or 50 mph relative speed, 80% (95%) or 20% (50%) of the dead (injured) can be saved (refer to Fig. 4 and 5).

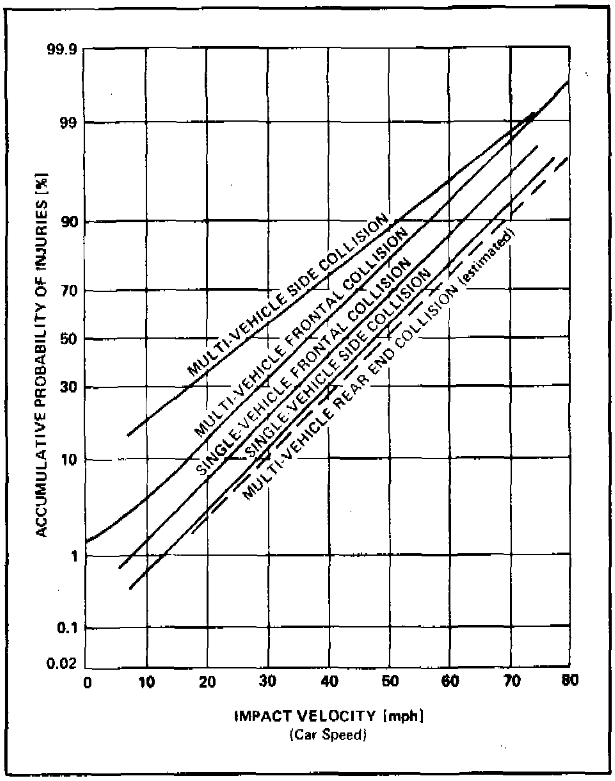

Figure 5

Table I shows that the frequency of serious frontal and rear accidents is at a ratio of about 10 : 1. To compute the effectiveness when meeting the conditions of the main impact tests the results (1969, USA) of Cooke[4] which are differentiated according to injured and dead are used as a basis (refer to Table II). The table shows that when meeting the ESV tests an approximately 7-fold effectiveness with regard to frontal accidents as compared with rear accidents will result. Since the costs for the improvement of the front and the rear structure are about the same, 0.60 for the front structure are indicating a considerably higher benefit/cost factor than the rear structure at 0.11 (refer to Table III), in fact a ratio of 5.5:1. The pertinent ratio for a 30 mph side impact is 2.4:1.

Therefore the assumption can be made that the front and the side tests are equivalent as the result of a benefit/cost factor which is about the same. But the rear end impact test is much too sharp. The use of one cost unit in the rear end structure provides only a fraction of the effectiveness which would result when investing such a cost unit in the side or even in the front structure.

Since at 75 mph rear impact speed the cumulative-injuries speed curve according to the estimate of 95% saved is already in the asymptotic range, and since the costs rise at more than the square of the test speed, a reduction of the rear impact speed will result in an increase of the benefit/cost ratio and thereby in a more effective matching to the other test conditions.

TABLE I
Probability of Frontal and Rear End Impacts

	Share % Frontal Impact			Share % Rear Impact		
	Total	Injuries	Deaths	Total	Injuries	Deaths
USA [4] 2,056,000 Accidents 1969	60	60	51	4.7	4.8	1.1
USA [3] *Rural* 25,000 Accidents 1967	59	–	–	6.9	4.5	2.8
England [6] Rural, Urban 296 Accidents 1967-1968	–	–	45	–	–	0
England [2] Rural, Urban 656 Accidents 1965-1969	55	–	–	10	–	–
Sweden [5] Rural, Urban 28,000 Accidents 1965-1966	36	–	–	<8.7	–	–
Australia [7] Rural, Urban 408 Accidents 1963-1964	43	–	–	6.8	–	–

TABLE II
Benefits For ESV – Specifications

Impact Type	Deaths Total *Injur. Total* USA 1969 by Cooke [4]	Losses *Total*[1] 10^6 $	% Cases *Within Prot.* Speed Range by Cooke [4]	Total *Economic* Benefits 10^6 $
50 mph single vehicle frontal	5,660 296,000	244 } 894 650	35 66	85 } 515 430
75 mph multi vehicle frontal	15,780 745,000	680 } 2,320 1,640	25 60	170 } 990 820
30 mph multi vehicle side	8,220 214,000	354 } 824 470	38 55	135 } 393 258
75 mph multi vehicle rear	458 93,000	20 } 225 205	80 (est.) 95	16 } 211 195
50 mph multi vehicle rear	458 93,000	20 } 225 205	20 (est.) 50	4 } 107 103

[1] $43,000 loss per fatality [8]
[1] $ 2,200 loss per injury [8]

TABLE III
Benefit/Cost Ratios for ESV Specifications

Impact Type	Total Benefits 10^6 \$	Total[1] Costs 10^6 \$	Benefit/ Cost Ratio	Benefit/ Cost Ratio related to 75 mph rear
50 mph single vehicle frontal	515 } 1.505	2,500	0.60	5.5
75 mph multi vehicle frontal	990			
30 mph multi vehicle side	393	1,500	0.26	2.4
75 mph multi vehicle rear	211	2,000	0.11	1.0
50 mph multi vehicle rear	107	500	0.21	2.0

[1] $10 \cdot 10^6$ new cars per year [8]
\$250 improved front structure per car
\$150 improved side structure per car (estimated)
\$200 improved rear structure per car for 75 mph
\$ 50 improved rear structure per car for 50 mph

2. Reduced Rear Impact Speeds

For the derivation of a meaningful speed for the impact of the rear end against a fixed barrier within the scope of the other test conditions the following assumptions are made:

1. The 50 mph frontal impact against a fixed barrier is meaningful.
2. Modern front structures are designed to meet the SAE Recommended Practice J 850 a, that is a 30 mph frontal impact against a fixed barrier.
3. The additional costs for the improvement of the front and rear structure beyond the present condition are to be in the ratio of the losses for frontal and rear impacts, that is (894 + 2.320) / 225 = 14.3/1 (refer to Table II).
4. The costs for structures are proportional to the possible energy absorption.
5. During a rear impact of two equal sized cars the energy absorbed at the rear is to be similar to the energy absorbed at the front.

As an example, a 1,000 kg passenger car of conventional design is used.

Present Costs For Front Structure:

$$A_{F1} = c \cdot \int F \cdot dx/_1 = c \cdot \tfrac{1}{2}m \cdot v_{F1}^2 = c \cdot 8.96 \cdot 10^4 \text{Nm}$$

in which

c = Proportionality factor
F = Force
x = Deformation Length
m = 1,000 kg mass of passenger car
v_{F1} = 30 mph

Costs Required For Front Structure:

$$A_{F2} = c \cdot \int F \cdot dx/_2 = c \cdot \tfrac{1}{2}mv_{F2}^2 = c \cdot 24.6 \cdot 10^4 \text{Nm}$$

in which
v_{F2} = 50 mph

Present Costs For Rear Structure:

$$A_{R1} \approx \tfrac{1}{2}A_{F1} = c \cdot 4.58 \cdot 10^4 \text{Nm (measured)}$$

Approach According To Assumption 3:

$$\frac{A_{F2} - A_{F1}}{A_{R2} - A_{R1}} \equiv \frac{\text{Losses Front}}{\text{Losses Rear}} \equiv \frac{14.3}{1}$$

Costs Required For Rear Structure:

$$A_{R2} = A_{R1} + \frac{1}{14.3}(A_{F2} - A_{F1})$$
$$= c \cdot 5.68 \cdot 10^4 \text{Nm}$$

The energy balances provide equivalent rear impact speeds:

1. **Impact rear against fixed barrier**

$$A_{R2} = c \cdot \tfrac{1}{2}mv_{R2}^2 \qquad V_{R2} = 23.9 \text{ mph}$$

2. Impact of moving barrier of similar weight against rear:

$$A_{R2} = c \cdot \tfrac{1}{2}\frac{m \cdot m}{m + m}\Delta v_B^2 \qquad \Delta v_B = 33.8 \text{ mph}$$

3. Impact of similar vehicle front against rear:

$$2 \cdot A_{R2} = c \cdot \tfrac{1}{2}\frac{m \cdot m}{m + m}\Delta v_c^2 \qquad \Delta v_c = 47.8 \text{ mph}$$

With the inclusion of approximately 4% safety the following rear end impact speeds are proposed:

Fixed barrier	v = 25 mph
Moving barrier	Δv = 35 mph
Vehicle-vehicle	Δv = 50 mph

To check the inclusion of the reduced rear impacts into the other specifications the benefit/cost factor is now computed. Figure 5 shows that with protection up to a relative speed of 50 mph rear impact speed approximately 50% of the losses will be saved. Tables II and III are showing computations or estimates of the absolute benefits and costs. The resulting benefit/cost factor of 0.21 is within the order of magnitude of the other specifications. The reduced rear impact test is therefore equivalent to the front and side impact tests.

3. References

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LIMITATION OF INTRUSION DURING SIDE IMPACT

Mr. William J. Wingenbach, AMF

Abstract

"Limitation of Intrusion During Side Impact"

One of the more difficult engineering problems encountered in providing vehicle crashworthiness is that of limitation of intrusion during side impact.

The work done on the AMF Experimental Safety Vehicle project has led to the evolution of a design concept which has as its basic element an aluminum honeycomb sandwich door panel. The sequential modes of behavior of the concept under increasing load levels are elastic beam action, plastic beam action, honeycomb crush, and finally, membrane stretching. As these behavioral modes progress, very large resistance to transverse loads develops even though transverse deflection remains small.

The concept has been modeled and analyzed for load deflection characteristics, and several evolutionary models have been built and tested under both static and dynamic loading, including full-scale vehicle crashes. Actual behavior has agreed very well with analytically predicted behavior enabling the side structure system to meet ESV design goals.

Introduction

The Advanced Systems Laboratory of AMF Incorporated at Santa Barbara, California, has been engaged in the development of an Experimental Safety Vehicle for the U.S. Department of Transportation. The scope of this project includes the complete span of activity from original conceptual design through developmental testing and evaluation, and culminates with the delivery of complete vehicles at the end of this year. To the extent possible, the project utilized a systems approach wherein each component was synthesized, analyzed, designed, and tested and evaluated with consideration of the interfaces with other components and of total vehicle objectives.

This paper is concerned specifically with the work associated with resistance of intrusion into the vehicle passenger compartment during side impacts. Included are an enumeration of technical objectives, descriptions of the system and method of analysis and a discussion of the results of developmental testing.

Objectives

The design of the AMF Experimental Safety Vehicle side structure was directed towards a set of objectives derived from explicit and implicit Department of Transportation goals. This set of objectives is as follows:

Passenger compartment intrusion is to be limited to three inches measured from a normal inside surface when struck on the side by the front bumper of a vehicle of equal mass. Impact velocity of 30 mph is normal to the side and at any point along the side. The impacting bumper structure is equivalent to the required ESV front bumper system which has the characteristics of providing override/underride protection over the range of 14 to 20 inches above ground, and a vehicle acceleration force which is velocity dependent. Maximum permissible vehicle acceleration versus impact velocity is shown in Figure 1.

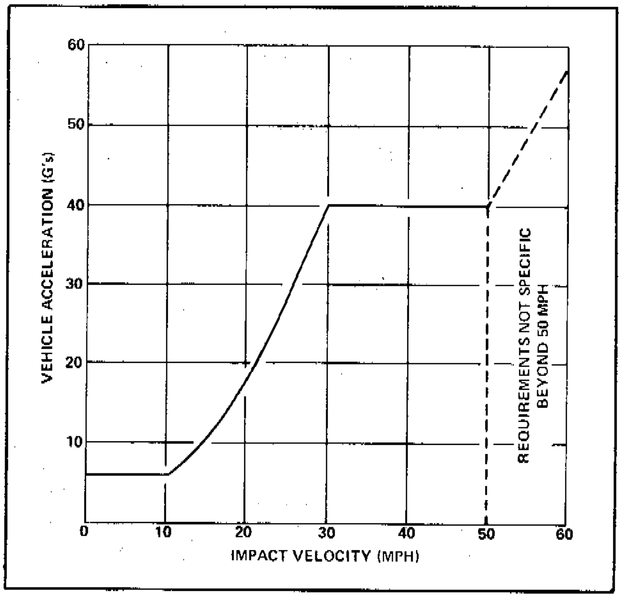


Figure 1

Passenger compartment intrusion is to be limited to three inches at the pillars and four inches at the longitudinal centerline of doors during impact into a fixed 14-inch pole. Impact velocity of 15 mph is normal to the side and at any point along the side.

Passenger compartment doors are to remain closed during any ESV specified crash condition. These conditions include front and rear impacts up to 50 mph and impact angles up to 45 degrees; rollover at 60 mph and side impacts. Impacts may occur with vehicle carrying five restrained or unrestrained occupants.

Passenger compartment doors are to remain operable after impact at any specified ESV crash condition.

Intrusion resistance is to be accomplished with minimum impact on overall vehicle cost and weight. Design concepts employed on the ESV should be susceptible to mass production.

Passenger compartment door systems are to provide ease of opening and closing, and ease of passenger ingress and egress equivalent to current production vehicles.

Side structure is to provide maximum driver field of view. There are to be no more than four pillars of minimum width within the driver's 270 degrees of forward view.

System Description

Major components of the intrusion-resistant side structure are the door panel, door retention hardware, and the passenger compartment side structure. A description of these components and their intended functions follows:

The door panel is an aluminum honeycomb sandwich consisting of an outer sheet, a honeycomb core, an inner sheet and a pair of vertical beams as shown in Figure 2. Door retention hardware along with the passenger compartment structure serve to provide the door panel with non-yielding pin supports fore and aft. Under transverse deflection of the panel, the pin supports develop a longitudinal tensile force which tends to stretch the door panel.

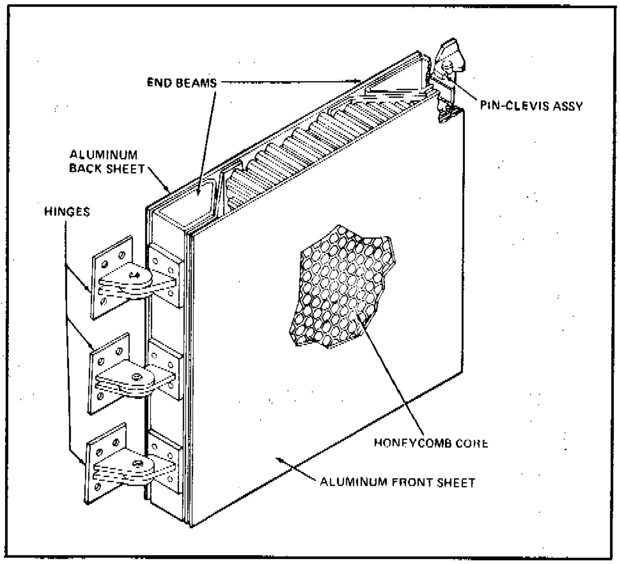


Figure 2

The sequence of behavior of the door panel under transverse loading is initially elastic and then plastic beam action during which the outer sheet is loaded in compression and the inner sheet is in tension. With increasing transverse load, the hoeycomb core begins to crush, decreasing the effectiveness of the panel as a beam. During this action, stress in the outer sheet reverses from compressive to tensile, while the inner sheet increases in tension. When the honeycomb is completely crushed both the inner and outer sheets are plastically stretched as membranes under tensile stress. This tensile stress in the sheets is reacted by the vertical beams which transmit the load to the door retention hardware. During deformation of the door panel, energy is absorbed in crush of the honeycomb and in plastic membrane stretching of the door sheets. In addition, a small amount of energy is stored by elastic deformation of the various structural elements.

Door retention hardware consists of three high-strength steel hinges, three pin clevis assemblies and a conventional door latching assembly. The hinges and pin clevis assemblies serve to provide the non-yielding pin supports for the door panel. As such, they provide the load path to the structure for the applied external transverse force and for the generated longitudinal tensile force. The conventional latch is adequate to resist internal transverse forces generated by an occupant striking the door during impact. The door retention hardware, and door panel along with a molded fiberglass inner and outer panel, window and window actuating mechanism, interior padding, etc., form the complete door assembly shown in Figure 3. Front and rear doors are similar in concept, although slightly different in shape. Rear doors also have fixed glazing.

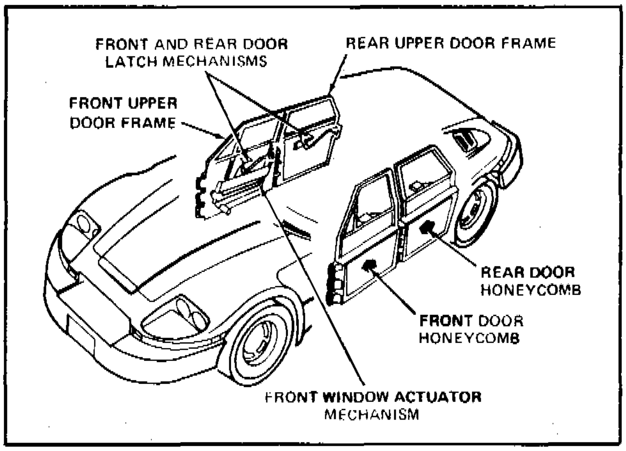


Figure 3

The portion of the passenger compartment structure which is involved in intrusion resistance to side impacts includes the A, B, and C posts and pillars, the perimeter frame, roof rails, and a honeycomb sandwich padding. In addition, there are several auxiliary transverse and longitudinal members which are employed to distribute loads throughout the vehicle. With the exception of the honeycomb sandwich padding, the structure is fabricated from high-strength sheet and tubing. The assembly is an all-welded integral structure shown in Figure 4. During impact by another vehicle, the side structure remains essentially elastic.

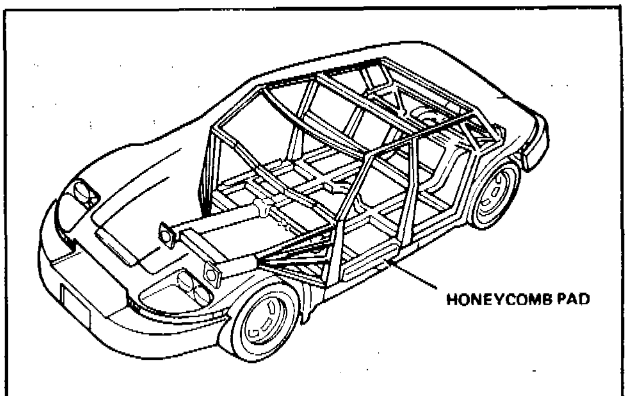


Figure 4. Space Frame Assembly

Figure 4

The aluminum honeycomb sandwich is installed outside of the perimeter frame in the region of the front door. During pole impact in this region, the honeycomb is crushed absorbing energy and distributing the load to the perimeter frame which remains essentially elastic. Pole impacts in the region of the shorter span rear door or at the posts will result in plastic deformation of the structure. A summary of properties of material utilized in the intrusion-resistant side structure is given in Table 1.

TABLE I
Properties of Material Utilized in Intrusion-Resistant Side Structure

	Material	Property
Door-Honeycomb sandwich, outer sheet	6061-0 AL	17 ksi-Ultimate
Door-Honeycomb sandwich, core	ACG AL	178 psi Crush
Door-Honeycomb sandwich, inner sheet	7075-T6 AL	76 ksi Ultimate
Door Vertical beams	6061-T6 AL	42 ksi-Ultimate
Hinges & Pin Clevis Assembly	AISI-4140 ST	200 ksi-Ultimate
Latch Assembly	American Motors	5000 lb.-Transverse
Passenger Compartment Structure	ASTM-517 ST	100 ksi-Yield
Frame-Honeycomb sandwich sheets	2024-T3 AL	65 ksi-Ultimate
Frame-Honeycomb sandwich core	5052-H39 AL	750 ksi-Crush

Method of Analysis

The general method of achieving a design is shown in Figure 5, and involves three separate analyses. The first is a deflection analysis to obtain the load-deflection characteristics of components of a structural system. Depending on complexity and the nature of the structure, this is accomplished through hand analysis or by use of a suitable computer model. The second analysis is that of determining the dynamic response loads of a structural system under various crash conditions. This is accomplished using a computer model, and the load-deflection data previously obtained. Finally, stress levels in the structure are calculated at the peak dynamic response. A preliminary design may be cycled several times through the analytical loop before acceptable results are obtained.

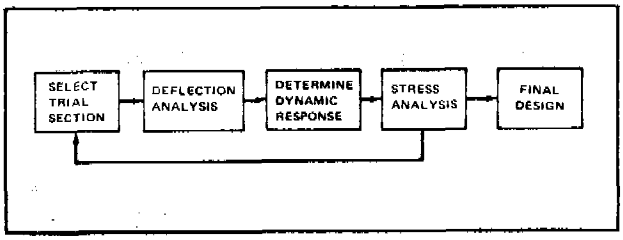


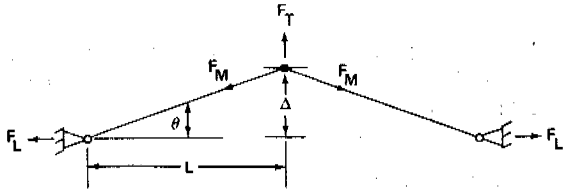
Figure 5

There are currently three mathematical models utilized in the analysis of impact problems. These are:

- SHOCK -- a nonlinear, lumped mass, dynamic response program for determining the behavior of systems under impulsive loadings.
- STRESS – a finite element beam program for the solution of space frame type structures.
- SAP – a general finite element program providing versatility in modeling three-dimensional structures.

The manner in which this approach was utilized in the specific problem of designing an intrusion-resistant side structure under the pole impact condition follows. The critical design condition was considered to be pole impact at the center of the front door.

STEP 1 Determine load deflection characteristics of door panel. The load-deflection behavior of the door panel was obtained by superposition of the various behavioral modes of the structure. Sequentially, the behavioral modes involved are: (a) elastic beam action; (b) plastic beam action, slight stretching of inner and outer sheet; (c) crush of honeycomb core, stretching of outer sheet, gradually increased stretching of inner sheet; (d) pure membrane stretching of inner and outer sheet. Membrane behavior was modeled and analyzed for load deflection characteristics using the truss model shown in Figure 6. This model is based on the assumption that a plastic hinge is initially formed at the load point and that the membrane behaves as a truss mechanism in post-yield behavior. The equations governing the load-deflection behavior of the mechanism in the fully plastic stage are presented in Figure 6. The accumulative load deflection curve for the panel is shown in Figure 7. Three regions under the curve are defined. Region "A" indicates the energy absorbed through beam action which also provides most of the panel stiffness at low deflections. Region "B" indicates energy absorbed by crush of the honeycomb core. Region "C", which accounts for the major energy absorption capacity of the panel structure is obtained through membrane stretching of the sheets.



LOAD-DEFLECTION EQUATIONS OF FULLY-YIELDED MEMBRANE

$F_T = 2F_M \sin\theta$

$F_L = F_M \cos\theta$

$\theta = \arctan \frac{\Delta}{L}$

$F_M = \sigma_y t h$

WHERE σ_y = YIELD STRESS OF MATERIAL
t = THICKNESS OF MEMBRANE
h = HEIGHT OF MEMBRANE

Figure 6

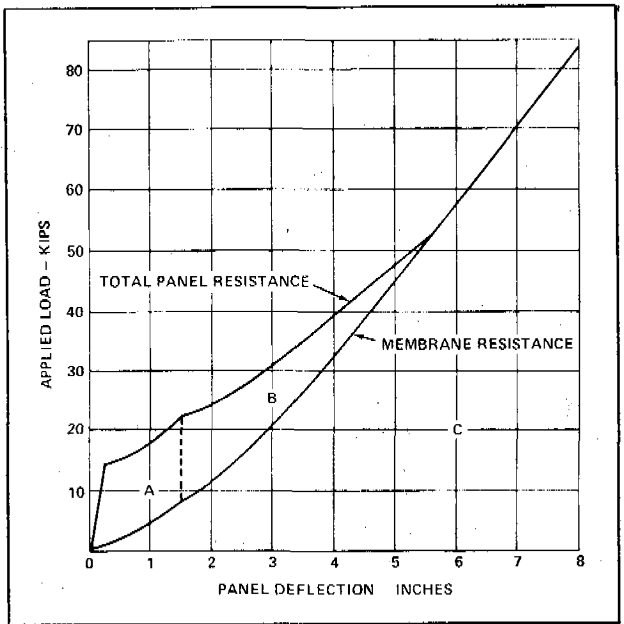


Figure 7

STEP 2 Determine load deflection characteristics of the frame honeycomb sandwich panel. The load deflection characteristics of this panel are shown in Figure 8.

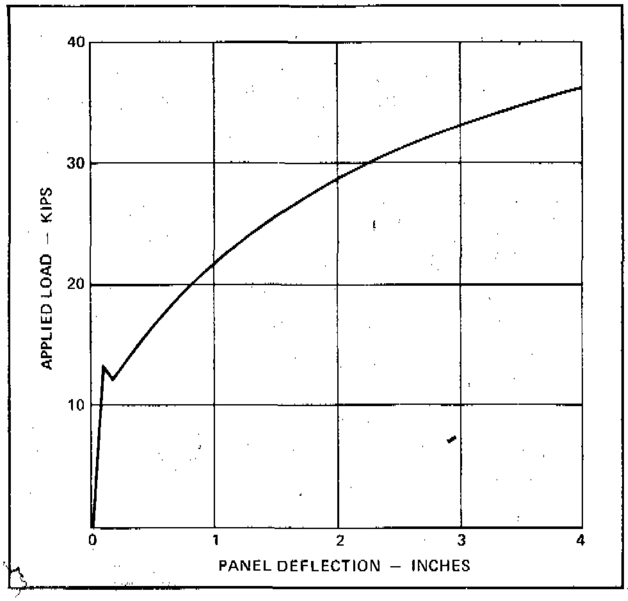


Figure 8

STEP 3 Analysis of the system under dynamic loading. The side structure system was modeled as shown in Figure 9. Included are the load-deflection characteristics of the door panel, frame honeycomb

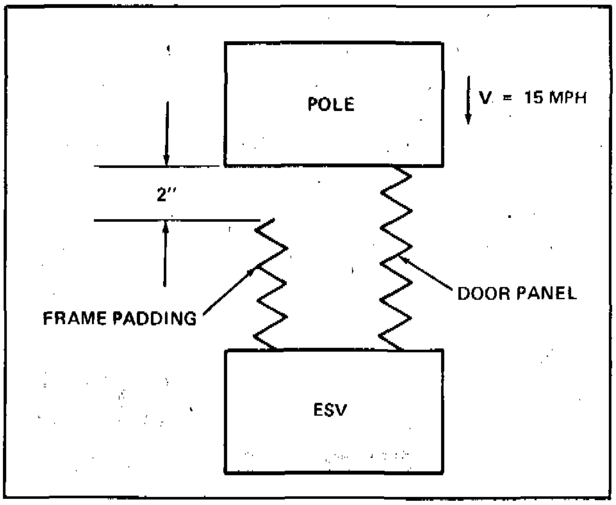


Figure 9

sandwich panel, mass of the vehicle and velocity discontinuity at the point of impact with the pole. A more complete model would include load deflection characteristics of the "A" and "B" posts and the perimeter frame, as well as rotational moments and vehicle moment of inertia. This model was exercised using the SHOCK computer code. The results of the analysis indicated that the pole would intrude into the striking vehicle about 7¼ inches measured from the point of initial contact with the door panel. This would result in an intrusion of the vehicle inside surface of approximately 3¼ inches compared to the allowable 4 inches. Door panel transverse load at this deflection is approximately 72 kips while the longitudinal load applied to the "A" and "B" posts is approximately 129 kips. The analysis indicated that the frame honeycomb sandwich panel would be completely crushed allowing direct pole contact with the perimeter frame resulting in a short duration load spike.

STEP 4 Design of structural elements. The final step in the analysis is that of sizing structural members using the dynamic response loads developed previously. In general, this step is an iterative process in which element dimensions are selected, analyzed and new dimensions chosen until a satisfactory design is achieved.

The passenger compartment side structure was designed using the STRESS computer code. The structure was modeled as shown in Figure 10, and analyzed for the loading conditions given in Table 2. Loads given in Table 2 are derived from the dynamic response analysis while the applied moments are those developed by the door retention hardware. A summary of member cross-sections and calculated peak axial and bending stress in each member is given in Table 3. These results were considered satisfactory, and the structure was fabricated for testing as modeled.

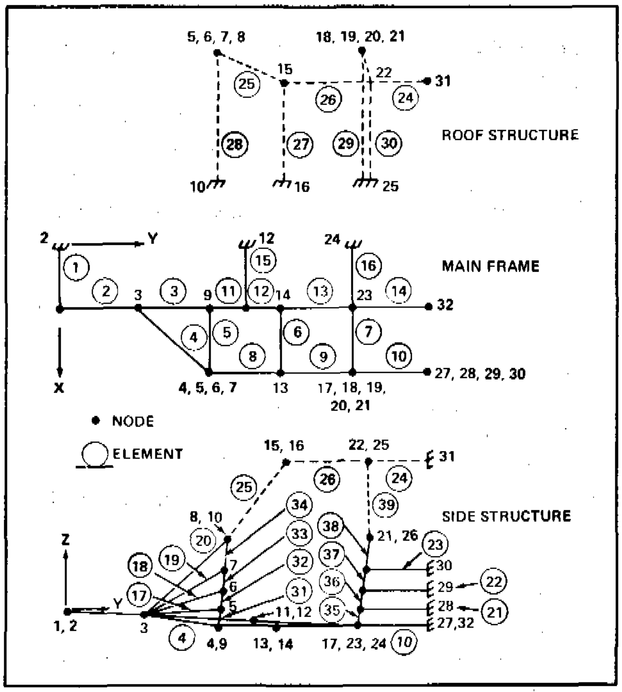


Figure 10

TABLE II
Applied Loads During Pole Impact

Node	Type	Direction	Magnitude
5	Force	X	-12.0 k
5	Force	Y	43.0 k
5	Moment	Z	150 in k
6	Force	X	-12.0 k
6	Force	Y	43.0 k
6	Moment	Z	150 in k
7	Force	X	-12.0 k
7	Force	Y	43.0 k
7	Moment	Z	150 in k
18	Force	X	-12.0 k
18	Force	Y	-43.0 k
18	Moment	Z	-150 in k
19	Force	X	-12.0 k
19	Force	Y	-43.0 k
19	Moment	Z	-150 in k
20	Force	X	-12.0 k
20	Force	Y	-43.0 k
20	Moment	Z	-150 in k
13	Force	X	-100 k

Test Results

The test program supporting the development of the intrusion-resistant side structure was conducted in two phases. The first phase involved component development in which several evolutionary versions of the aluminum honeycomb panel and door retention hardware were built and tested until satisfactory results were obtained. The second phase involved the construction and testing of a complete structural vehicle.

TABLE III
Side Structure Cross-Sections & Peak Stresses

Element	Name	Section	Axial	Bending
1	Front Housing Stabilizer	3x3x.120	6.8 (ksi)	63.2 (ksi)
2	Front Housing	3x5x.188	1.3	46.3
3	Front Housing	5x5x.188	7.3	40.2
4	Lower A Post Support	2x2x.062	5 4	133.0
5	A Post Lateral	4x4x.120	22.4	116,2
6	Front Door Lateral	3x3x.100	85.9	40.7
7	B Post Lateral	4x4x.120	13.5	91.5
8	Floor Sill--A to Center	5x4 to 3x3x.120	41.5	118.6
9	Floor Sill–Center to B	5x4 to 3x3x.120	66.4	64.3
10	Floor Sill--B Aft	3x4x.120	29.3	33.1
11	Front Housing	5x5x.188	1.2	85.9
12	Main Frame	4x4x.120	4.5	131.2
13	Main Frame	4x4x.120	5.0	127.2
14	Main Frame	4x4x.120	9,7	21.2
15	Torsion Bar Frame	3x4x.120	56.4	58.4
16	Center Cross Frame	4x4x.120	32.0	34.3
17	Lower A Hinge Support	1¼x1¼x.250	18.5	20.8
18	Mid. A Hinge Support	1¼x1¼x.250	35.8	24.8
19	Upper A Hinge Support	1¼x1¼x.250	18.7	11.4
20	Upper A Post Support	1¼x1¼x.250	14.7	135.3
21	Rear Panel Simulation	1 3/8x.095		
22	Rear Panel Simulation	1 3/8x.095		
23	Rear Panel Simulation	1 3/8x.095		
24	Roof Sill Aft	3x2x.188	15.5	37.0
25	A Pillar	2½x2½x.120	26.5	115.5
26	Roof Sill -Fwd	3x2x.188	14.9	128.9
27	A Roll Bar	2x2x.120	10.7	109.0
28	A Cross Frame	2½x2½x.120	35.5	122.5
29	B Cross Frame	3x4½x.188	22.9	112.1
30	B Roll Bar	2x2x.120	1.6	38.0
31	A Post	3x8 to 3x3x.120	12.5	101.5
32	A Post	Same as 31	5.2	112.8
33	A Post	Same as 31	11.4	139.5
34	A Post	Same as 31	11.1	139.5
35	B Post	3x8 to 3x3x.120	1.8	57.3
36	B Post	Same as 35	2.3	151.5
37	B Post	Same as 35	2.3	151.5
38	B Post	Same as 35	3.3	129.1
39	B Pillar	2x2x.120	1.3	143.2

The first phase – component development test program – was conducted using the AMF crash simulator. This facility, which has been utilized to support a variety of automobile research programs, has the following features:

- Volumetric capacity in excess of a full size automobile
- Static or dynamic load capacity
- Tri-directional loading capacity
- Load application capacity of 100,000 pounds, 25,000 pounds, and 15,000 pounds along orthogonal directions
- Ability to apply loads separately, sequentially or simultaneously
- Ability to simulate crash load pulses including control of onset rate, pulse magnitude and pulse duration
- Fixturing to mount and provide controlled retention of a wide range of component size, shape and configuration

The result of a dynamic test of the door panel and door retention hardware is shown in Figure 11, along with analytically predicted behavior. Up to the point at which a hinge attachment failed, the behavior was reasonably close to expected behavior and was considered to be satisfactory. The failure in a weld at the hinge attachment was attributed to a manufacturing deficiency and not to the design. Observations made using high-speed photography and strain gages confirm the occurrence of expected sequential modes of behavior. Prior to failure, the panel had exhibited both elastic and plastic beam action, stress reversal in the front sheet and honeycomb core crush through more than half of its thickness. Peak loads were 39.6 kips, transverse, and 70.1 kips, longitudinal at a ram stroke of 4.25 inches. Post-test observation disclosed a vertical break through the front sheet at the point of contact. This was attributed to the occurrence of buckling while stressed in compression and subsequent failure in tension at the crease formed by buckling.

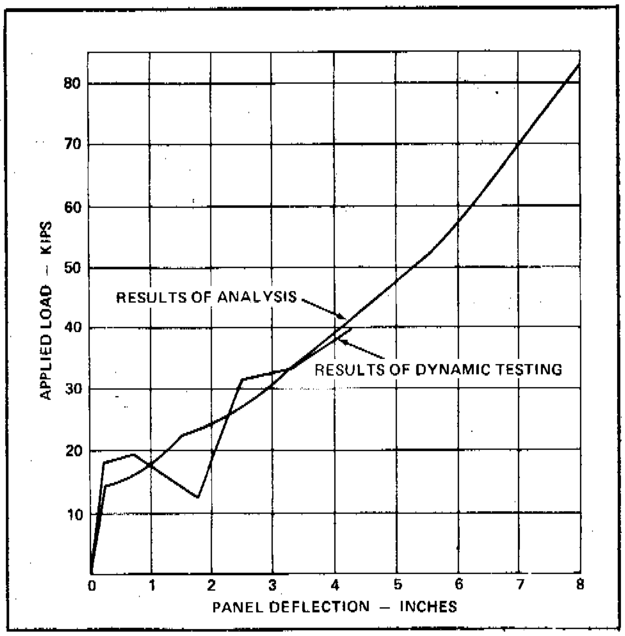


Figure 11

A door panel of the same design was installed on a structural vehicle. The assembly was impacted against a