COMPATIBILITY REQUIREMENTS FOR CARS IN FRONTAL AND SIDE IMPACT

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ABSTRACT

In support of the European Enhanced Vehicle-safety Committee (EEVC) research programme and through it, the International Harmonisation of Research Activities work on compatibility, TRL is investigating the compatibility of cars in frontal and side impact scenarios. Initial research has focused on identifying the major factors which influence compatibility and determining the extent to which they might influence injury outcome. Experimental crash test research is backed with Finite Element simulation modelling. For frontal impacts, full scale testing has been used to examine the influence of vehicle mass, stiffness, structural interaction and geometry. The modelling work has studied how non contact, deceleration related injuries might be minimised by optimising the deceleration pulse. For side impact, full car finite element models have been used for parametric studies to aid our understanding of the effects of the bullet vehicle mass, geometry and stiffness and to help predict more compatible designs. This has been backed by full scale crash testing, aimed at determining the ideal characteristics of interacting car front and side structures. All of this work is aimed at developing crash test requirements that are capable of assessing a car's compatibility.

INTRODUCTION

It is now generally accepted by the international vehicle safety community that the 'compatibility' of vehicles needs to be addressed. The development of vehicles which simply 'self protect' will no longer be acceptable and growing emphasis must be placed upon the protection of all road users.

Each subsystem within the global fleet may currently possess its own unique 'incompatibilities'. A large number of U.S. publications are written about side impact and the problem associated with 'aggressive' sport utility vehicles and their mismatch with the midsize family saloon. This problem is exacerbated by a road system that gives rise to a large number of side impacts. In European publications the emphasis is on frontal impact. However, it is well established that a compatible fleet can not be achieved through concentrating on a single impact scenario.

Research, guided by EEVC WG 15, which has been set up to study compatibility, will concentrate on both frontal and side car to car impacts. In addition it will also consider the requirements for car impacts with pedestrians, large vehicles and roadside obstacles. This should ensure that developments to improve car to car impacts do not reduce protection in these other types of impact.

In 1995 TRL commenced compatibility research stimulated by their findings from frontal and side impact testing. The programme is funded by the UK Department of Environment, Transport and the Regions and the European Commission. The work contributes to EEVC WG 15 and the International Harmonisation of Research Activities Compatibility Working Group.

The programme aims to identify how vehicle safety may be improved by developments to vehicle structures which are designed to interact better in an impact, and subsequently implement these changes in the vehicle fleet. This requires an understanding of the factors which influence compatibility and the development of new or modified legislative procedures to bring about greater compatibility. The project also aims to identify the potential benefits that could be obtained from improved compatibility.

This paper discusses those factors that are currently seen as having the greatest influence on compatibility and reports on the work carried out so far.

CAR TO CAR IMPACTS

Current activities are focusing on car to car frontal impact and car to car side impact separately, in order to understand more clearly the controlling factors. It is envisaged that conflicting requirements will exist and that a compromise will have to be established. It is possible that the current fleet differences between the U.S., Europe and other continents, will encourage researchers to reach differing compromises. It is for this reason that great importance must be placed upon international harmonisation in an attempt to reach a single, internationally appropriate, assessment procedure. One step towards compatibility may be the worldwide harmonisation of legislative and consumer testing.

CAR TO CAR FRONTAL IMPACT

This section discusses those parameters which are currently seen as having the greatest influence on compatibility in frontal impact and presents the results from some of the work carried out so far.

Collision Type

Collision type is the most important factor governing vehicle performance since it determines how the two car's structures will interact. For this reason, collision type will have a modifying influence on the effect of other parameters such as geometry and stiffness. For example, the effective global stiffness of a vehicle changes significantly from a full frontal impact to frontal offset impact, highlighting the importance of selecting the appropriate testing procedure (1). In this example, the differences in effective stiffness between the two collision types are attributed to the specific structures used for energy absorption and the modes by which they fail.

Geometry

Car structures are designed to perform a multitude of functions as well as good crashworthiness. As a consequence, car fronts often have local areas of stiff structure within a much larger area of weaker structure. Due to this lack of stiffness uniformity it is usual in a car to car frontal impact, for the stiff parts of one vehicle to penetrate the weaker parts of the other. This may result in penetration fork effect or over-ride. Since in these situations the vehicle's energy absorbing structure is ineffectively used, higher occupant compartment intrusions are often observed. In one reported offset frontal test between a small and medium sized car, the poor structural interaction masked the effect of mass and stiffness and dominated the outcome (2). In this test the structure of the small car over-rode that of the medium sized car.

More recently this test has been repeated. The aim was to select a vehicle which would remove the previously reported geometrical incompatibilities and permit the assessment of mass and stiffness influence. The chosen substitute was a car that performed structurally well in the European Offset Deformable Barrier (ODB) test, whilst being of a similar mass to the car it replaced. In this repeat test the medium sized car, possessing better frontal structure tie-up, virtually eliminated over-riding (Figure 1.).

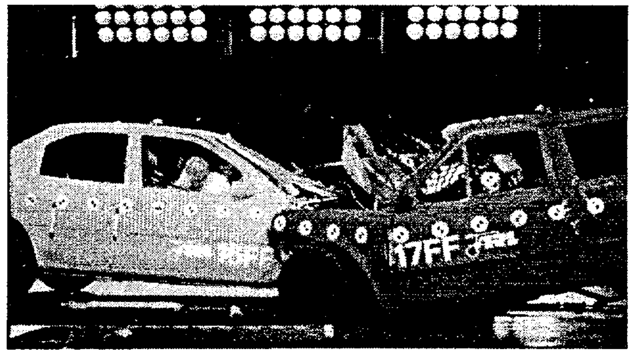

Figure 1. Medium size passenger car with well-connected front structures minimised over-ride by the small car.

Unfortunately, geometrical compatibility is not straight forward to assess. This was highlighted in an offset frontal car to car impact carried out between an off road vehicle and the medium size passenger car. It can be seen from the pre test photographs that there was a considerable difference in the vertical height of the significant structures of the two vehicles (Figure 2.).

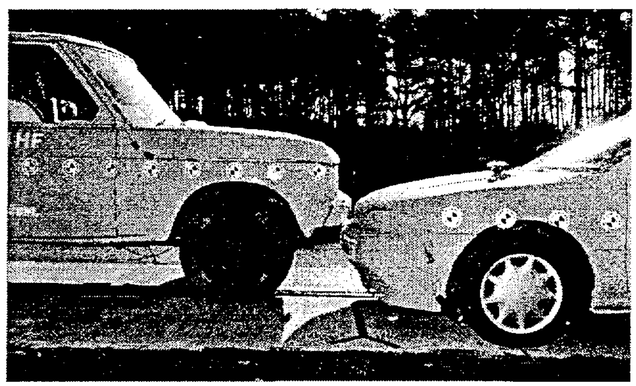

Figure 2. Geometrical differences between medium size passenger car and off road vehicle.

The results were contrary to expectations. The geometry of the two vehicles interacted well and there was no significant over-riding. In this particular test, the chassis rail of the off road vehicle engaged the suspension turret of the passenger car becoming retained by the engine mount attachment. A significant proportion of the loading was then transmitted through the engine and onto the firewall (Figure 3.).

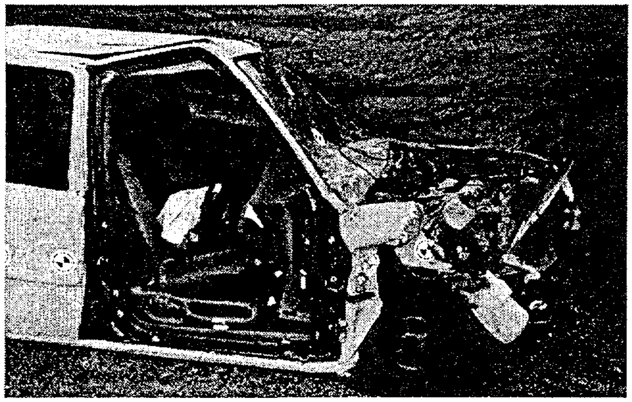

Figure 3. Good geometrical interaction resulted in no significant over-riding by impacting 1900 kg off road vehicle, note intrusion at facia level.

The resulting deformation of the car's front structure and door aperture were not dissimilar to the results obtained from a car to car impact between two of the same medium size vehicles (Figure 4.)

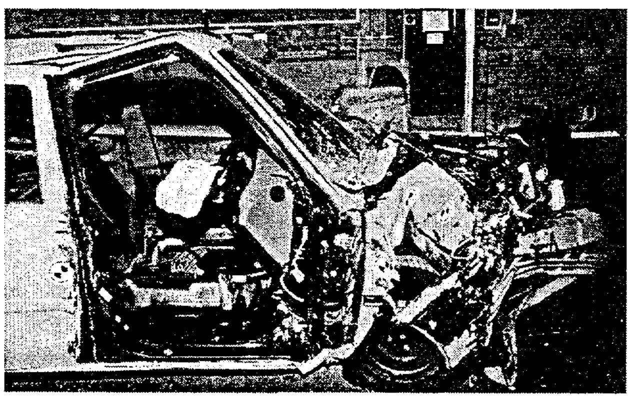

Figure 4. Deformation of medium sized car after offset frontal impact with identical vehicle.

One significant difference, however, was the intrusion at facia level. In the car impacted by the off road vehicle, the additional loading directed onto the fire wall resulted in high occupant compartment intrusion, and as a consequence caused a higher chest deflection on the dummy. An important observation to note from this test is that the structures of two apparently incompatible vehicles interacted in such a way as to virtually eliminate over-riding. Even though the good structural performance was not reflected in satisfactory dummy injury criteria (high chest deflection), this test goes part way to demonstrate that compatibility between dissimilar vehicles may be achievable.

In all of the tests commented on, it is a fair observation to make that geometry is not the only variable. Hence, in order to study the importance of structural compatibility, tests need to be conducted which have the sole variable of geometry.

A medium size, car to car offset test was performed using identical vehicles, at the same test weight, but having undergone modifications to their ride heights. The vertical difference in height was 100mm, well within the fleet variations for this segment of car (3).

In the lowered car, a large proportion of the loading was carried by the upper load path (Figure 5.). This resulted in excessive intrusion at facia rail level and to a higher degree than that observed in the car impacted by the off road vehicle.

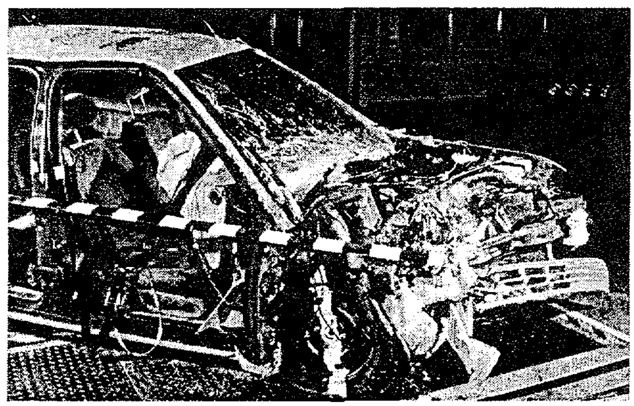

Figure 5. Lowered car, note 'A' pillar pushed down and high deformation at facia rail level.

In the lowered car the front profile was noticeably sloped back, and the 'A' pillar had been pushed downwards. The excessive deformation of the top load path meant the suspension turret had been displaced 480mm rearwards. By contrast, the suspension turret of the higher car moved rearwards by only 295mm. In this vehicle the 'A' pillar was pushed upwards increasing the deformation seen on the cant rail. More use was made of the lower load path and the frontal profile can be seen to be flatter (Figure 6.).

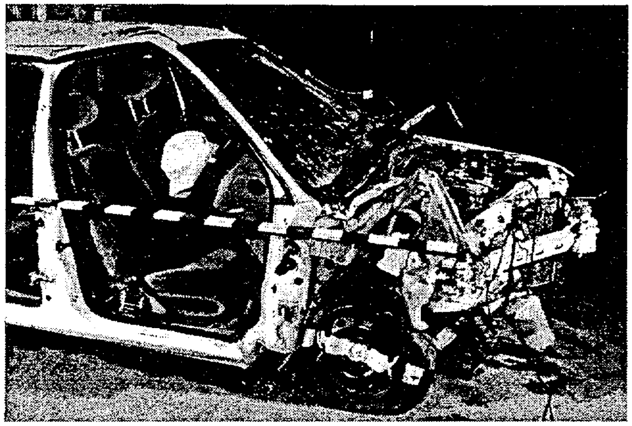

Figure 6. Raised car - note 'A' pillar pushed up and little deformation at facia rail level.

In this test the vehicle's mass, stiffness and occupant compartment strength were matched. However, the 100mm vertical height difference was sufficient to noticeably change the vehicles structural response. When considering that the vertical height difference of the two vehicles was less than the height of the bumper beam, these results become significant. Until vehicle designs enable structures to interact better in car to car impacts, any compatibility improvements in mass ratios, or stiffness matching, are unlikely to be fully realised. The authors suggest that in order to achieve a compatible fleet it will be necessary to firstly establish good geometrical compatibility.

Mass and Structural Stiffness

The effect of mass and mass ratio on injury risk have been studied for many years and dominate the literature on compatibility. There is plenty of evidence that in car to car impacts the risk of injury in the heavier car is lower than that in the lighter car (4). However, whilst the authors do not dispute that mass has a significant effect on injury, it must be remembered that mass is a surrogate measure for other factors such as vehicle size, length of front structure and presence and quality of safety features. These factors could help to exaggerate the mass effect. It is also recognised that mass is a common parameter to record during accident investigation and as a consequence is available for data analysis.

In frontal accidents, occupants of lower mass vehicles often have higher injury risks due to both lower vehicle mass and stiffness (5,6). Ignoring crashworthiness, heavier cars are stiffer for other reasons, with structures often being stronger to take the higher engine and suspension loads (2). There is, however, no certainty that the stiffness differential of small and large cars will continue to exist at the current ratio. There is clear evidence that the stiffness of smaller cars is increasing as a consequence of new crash test requirements.

There is virtually universal agreement amongst independent accident investigators that passenger compartment intrusion is a major cause of fatal and serious injuries to restrained car occupants. This view is also supported by many accident investigators employed by car manufacturers (5). In order to limit and manage occupant compartment intrusion in car to car frontal impacts, the crush zones of the cars involved must be capable of absorbing the full energy of the impact. A number of simple concepts have been proposed to control the global stiffness of the vehicle and hence improve compatibility.

The first proposal to note is that of a 'semi-rigid' passenger compartment (7). This suggestion involves designing the vehicle in such a manner that the occupant compartment is sufficiently stiff so that it can resist the deformation force put on it by any colliding car. This ensures that the impact energy is absorbed by the front structures of both cars.

Another suggestion has be made in which a limit is placed on the maximum crush force that must not be exceeded in a given impact (8). This concept has been extended with suggestions being made that both maximum and minimum force requirements are needed or that force corridors should be defined. These proposals are effectively equivalent to controlling the vehicle's stiffness.

Two fundamental questions need to be answered before pursuing either of these logical proposals:
1. What test could be performed to ensure that the occupant compartment could withstand the maximum crush force level?
2. What should the crush force level(s) be?

Currently there is no internationally recognised or legislative test which is capable of addressing the first question. It would be necessary to introduce an additional test to make such an assessment. Two possibilities may be a quasi-static crush, or a frontal test at an elevated speed. Neither of these solutions are however without drawbacks.

In order to be in a position to address the second of these questions it is important to have an understanding of current interface force levels between the impacting cars, and to be able to measure them in both barrier and car to car impacts. The proposals which have been commented on in this paper are theoretical and the first step must be to establish if such requirements could be achieved.

Interface force can be obtained through the use of a load cell wall or the measurement of the deceleration of the constituent parts of the vehicle. Both of these methods have been used at TRL and for barrier tests have been shown to be in good agreement (Figure 7.).

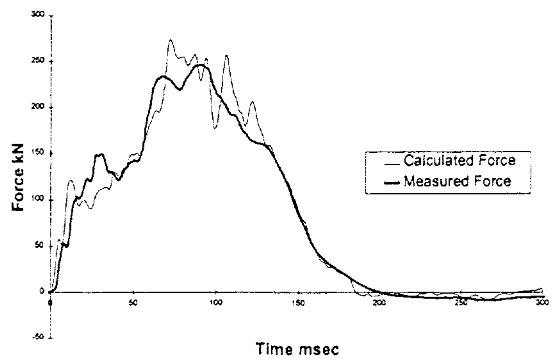

Figure 7. Interface force from frontal ODB test, note agreement between load cell wall data and interface force calculated from accelerometers.

In order to calculate the interface force from the car's deceleration, it is necessary to select and instrument discrete components and areas of structure. The selection is made based upon how it is believed these discrete areas of the vehicle will move and deform during the impact. It is assumed that the selected area will act as a lumped mass and the deceleration force for each lumped mass can be calculated from its associated deceleration. The summation of these forces gives the overall interface force (Figure 8.).

Differing views exist on how the total interface force could be limited or controlled. However, it should be possible to exercise control over vehicle local stiffness by limiting the interface force distribution. This type of control could be one way to improve geometrical compatibility.

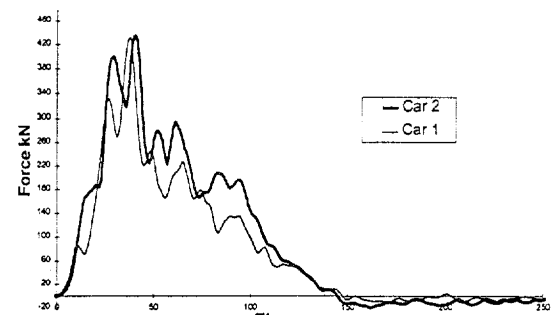

Figure 8. Interface force calculated in car to car offset impact.

With the semi rigid passenger compartment and the crush force limit theoretical concepts (7,8), it would be possible to achieve the requirements without controlling the shape of the deceleration pulse. It is the form of deceleration seen by the occupant compartment that drives the performance of the restraint system, and influences any restraint system induced injury. If we assume that our compatible vehicle has minimised intrusion due to the 'rigid' occupant cell, this type of injury will be dominant.

For this reason a study was initiated into the influence of the deceleration pulse on restraint system related injuries for frontal impact, with the aim of modifying the deceleration pulse to minimise this type of injury. Having identified the most desirable shape we will, in later work, assess if it is possible to achieve such a pulse in real cars.

<u>Computer Simulation</u> was selected as the most appropriate method by which to address this question. The MADYMO software package was used to simulate the deceleration of the occupant compartment. (Figure 9.).

Various simple shaped, analytical and experimental deceleration pulses were applied to the model in the fore aft direction. The Fourier Equivalent Wave (FEW) method (9) was used to generate a wide range of analytical deceleration pulses with predetermined ride down distances and times. This method sums three sine waves to produce a deceleration pulse :

$$a(t) = \sum_{N=1}^{3} a_n \sin(n\omega t) \qquad (1.)$$

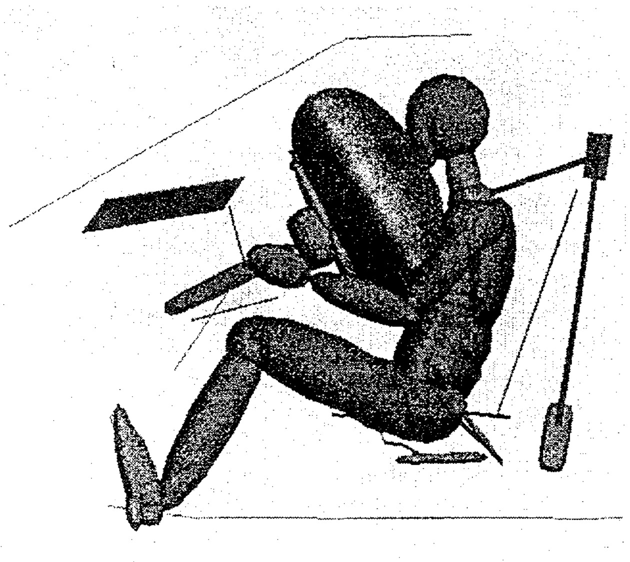

Figure 9. Occupant compartment model consisting of a HYBRID III dummy held by a typical restraint system.

Chest injury criteria were monitored as the chest is directly loaded by the seat belt. Peak chest compression was used as the measure of chest injury in the current study in preference to Viscous Criteria. Previous researchers who have studied the mechanism of impact induced soft tissue injury show that for a properly designed restraint system, the best indicator of injury is the peak chest compression (10).

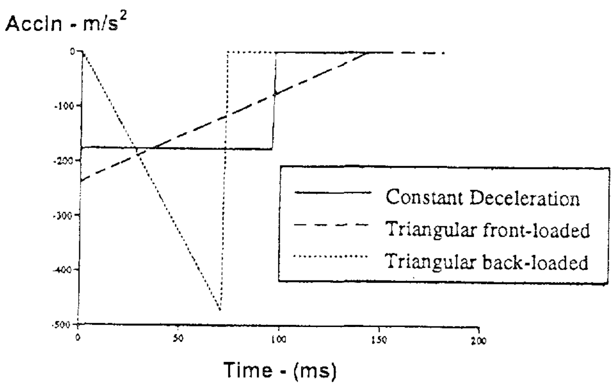

Figure 10. Comparison of simple shaped deceleration pulses for a ride down distance 0.8m.

Table 1.
Chest Injury; Simple Deceleration Pulse Shapes, Ride down Distance 0.8m, Initial Velocity 61km/hr.

Shape	Ride Down Time (msec)	Chest Injury	
		Peak Comp (mm)	Peak Accln (ms^{-2})
Constant	95	41 @62ms	359 @59ms
Triangular front loaded	142	44 @57ms	408 @55ms
Triangular back loaded	71	56 @78ms	618 @76ms

Investigations of the relationship between Hybrid III sternal deflection and thoracic injury severity on occupants have established that there is a 5 percent risk of injury greater or equal to AIS3 for a chest compression of 22mm. For a chest compression of 50mm the risk of injury greater or equal to AIS3 increases to 50 percent (11). Chest acceleration is also shown as a supplementary measure of injury.

The first parameter sweep applied three simple shaped deceleration pulses; constant, front loaded triangular and back loaded triangular, to the model to produce a ride down distance of 0.8 m (Figure 10.). This is a typical ride down for a 50 percent offset car to car impact. The results show that the constant pulse gives the lowest chest compression and the triangular back loaded the highest (Table 1.). An explanation for this emerges upon examination of the seat belt loads (Figure 11.).

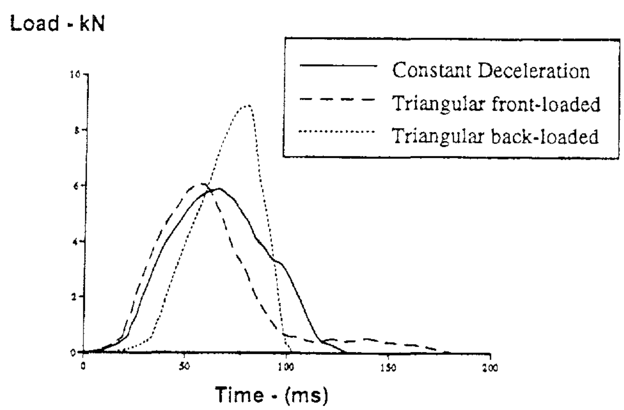

Figure 11. Comparison of seat belt loads for simple shaped deceleration pulses, ride down distance 0.8m.

The high deceleration at the beginning of the constant pulse, loads the occupant into the restraint system early. The constant deceleration, applied throughout the duration of the long ride down time, keeps the load applied over as long a time as possible to give a well rounded chest compression profile with a low peak (Figure 12.).

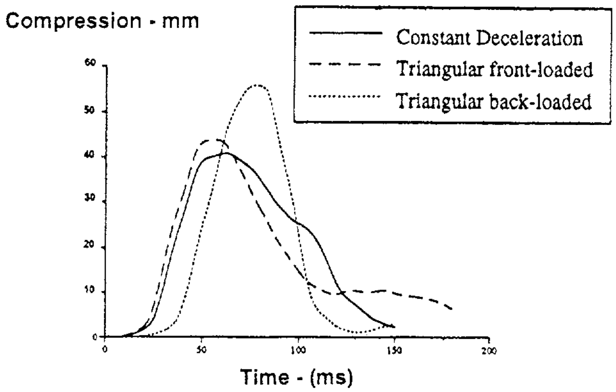

Figure 12. Comparison of chest compression for simple shaped deceleration pulses, ride down 0.8m.

In contrast, the low deceleration at the beginning of the back loaded triangular pulse loads the occupant into the restraint system late. As a consequence, a high restraining load is applied to the occupant for a short time, which results in a high peak chest compression. This also causes the peak chest compression to be at a much later time compared to the constant pulse.

A similar result is seen for a ride down distance of 1.2m which is a typical ride down for a car to ODB impact (Table 2.).

Table 2.
Comparison of Chest Injury; Simple Deceleration Pulse Shapes, Ride down Distance 1.2m, Initial Velocity 61km/hr.

Shape	Ride down Time / msec	Chest Injury	
		Peak Comp (mm)	Accln (ms^{-2})
constant	173	31 @67ms	226 @63ms
triangular front loaded	214	35 @63ms	266 @60ms
triangular back loaded	107	40 @104ms	404 @104ms

It is interesting to note that decreasing the ride down distance from 1.2 m to 0.8 m for a constant pulse has approximately the same effect on chest compression as keeping the ride down distance constant at 1.2 m and changing the shape of the pulse from a constant to a triangular back loaded pulse. These results are supported by previous work carried out at TRL (12).

Parameter sweeps were conducted using the analytically derived FEW deceleration pulses for various ride down distances. For this part of the study the airbag and steering wheel assembly was removed from the model. This eliminated the sensitivity of results to airbag trigger time. The double humped FEW pulse that gave the lowest chest compression for a ride down of 0.8m, is shown (Figure 13.), together with a graph of chest compression (Figure 14.).

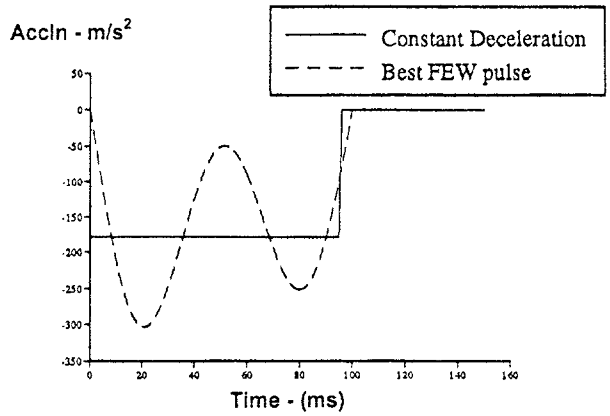

Figure 13. Comparison of doubled humped FEW and constant deceleration pulses.

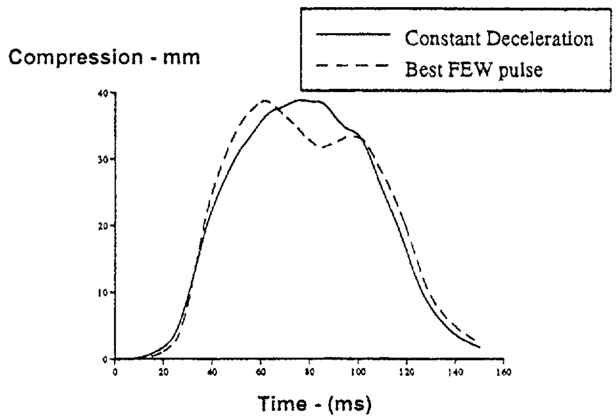

Figure 14. Comparison of chest compression for double humped FEW and constant deceleration pulses.

If these graphs are examined in detail it is seen that the double humped FEW pulse gives a low chest compression because of a similar behaviour to the constant pulse, i.e. the high deceleration at the beginning of the pulse loads the occupant into the restraint system early and the high deceleration at the end keeps the load applied for as long as possible.

Summarising, the results from this study indicate that :

1. Peak chest compression is minimised by maximising ride down distance.
2. Peak chest compression is minimised by having a passenger compartment deceleration pulse profile that is constant in shape as opposed to triangular back loaded.

The next part of the study abstracted the fore aft occupant compartment deceleration pulses from the following crash tests:

1. Car to car frontal impact with 50 percent overlap at an initial velocity of 56 km/hr.
2. European ODB test with 40 percent overlap but at an initial velocity of 61 km/hr.
3. Light car impacted by heavy car (mass ratio 1.4) with 50 percent overlap at an initial velocity of 56 km/hr.
4. Heavy car impacted by light car (mass ratio 1.4) with 50 percent overlap at an initial velocity of 56 km/hr.
5. Car to rigid barrier test with 100 percent overlap at an initial velocity of 56 km/hr.

It should be noted that in all of these crash tests the same model of mid sized family car was used. These pulses were applied to the model, in order to determine occupant response without any passenger compartment intrusion. Two restraint systems were modelled; the first was a seatbelt only system, and the second a seatbelt and airbag system. The results of applying these pulses to the model for both restraint systems are shown below (Table 3.).

Firstly, we will discuss the differences between the model response for the car to car 50 percent overlap and the ODB deceleration pulses. A comparison of the deceleration pulses is shown (Figure 15.).

For the seatbelt only system, it is seen that the chest compression is higher for the ODB deceleration pulse (Figure 16.). This shows that the advantage of having a longer ride down distance is out-weighed by the increase in velocity and the change in the shape of the pulse, i.e. to a more triangular back loaded form.

Table 3.
Chest injury model results obtained by applying deceleration pulses abstracted from crash tests.

Deceleration Pulse Description	ΔV (km/hr)	Ride-down Distance (m)	Seatbelt restraint system		Seatbelt and airbag restraint system	
			Chest Comp (mm)	Chest Accln (ms^{-2})	Chest Comp (mm)	Chest Accln (ms^{-2})
1. Car to car 50% overlap	56	1.0	37	305	38	327
2. Offset Deformable Barrier 40% overlap	61	1.2	41	335	35	341
3. Light car impacted by heavy car 50% overlap	65	0.8	42	370	44	422
4. Heavy car impacted by light car 50% overlap	47	1.2	33	252	33	257
5. Full frontal rigid barrier 100% overlap	56	0.7	43	365	43	405

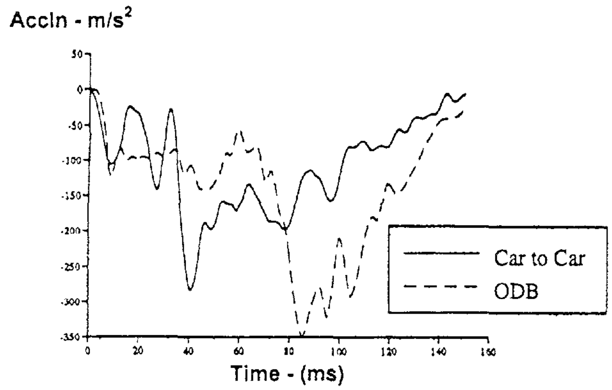

Figure 15. Comparison of deceleration pulse shape for ODB and 50 percent car to car.

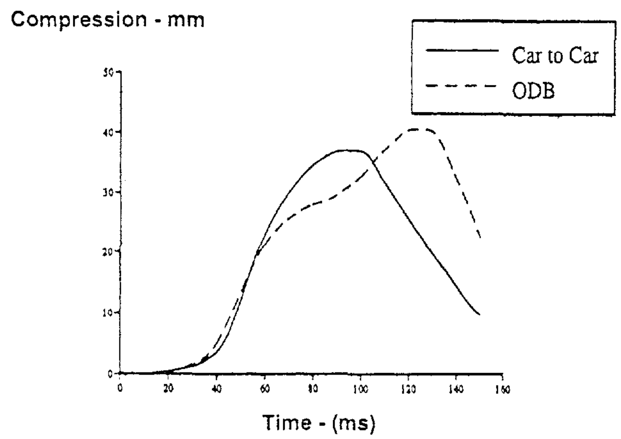

Figure 16. Comparison of chest compression for car to car and ODB impact with seatbelt restraint system.

For the seatbelt and airbag system it is seen that the chest compression is higher for the car to car deceleration pulse (Figure 17.). The reason for this is that the higher deceleration at the beginning of the car to car pulse causes the occupant to contact the airbag with a higher velocity giving a high sternum to airbag load (Figure 18.), so causing higher chest compression. One possible explanation for this is that the restraint system is 'tuned' to perform well in the ODB test. Manufacturers should be aware that the additional deceleration at the beginning of a car to car pulse compared to an ODB pulse can cause the occupant to contact the airbag with a greater velocity hence causing higher chest loads.

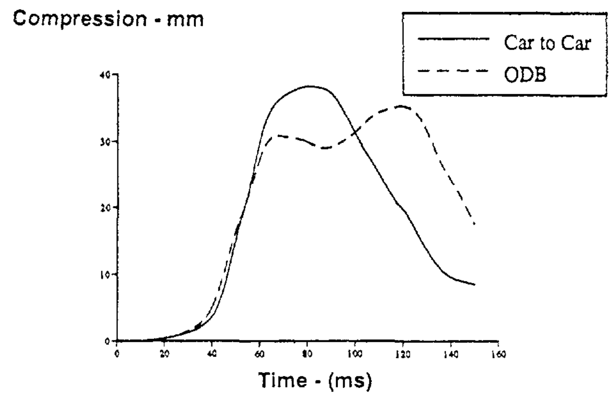

Figure 17. Comparison of chest compression for car to car and ODB impacts with seatbelt and airbag restraint system.

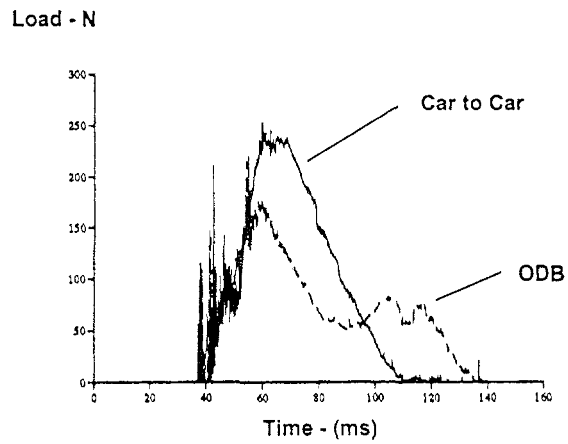

Figure 18. Comparison of sternum load from airbag for car to car and ODB impacts.

This would be difficult to allow for by adjusting the airbag trigger time, as the beginning of the pulses are very similar.

Secondly, we will compare the differences in the model response by applying the deceleration pulse from the car to car test with the pulses from the cars in the light to heavy car test (Figure 19.). The deceleration pulse from the light car has a more triangular back loaded shape, whilst the pulse from the heavy car has a more constant form. In addition the ride down is less and the change in velocity greater for the lighter car (Table 3.). It should be noted again that these cars are the same model and hence have the same stiffness

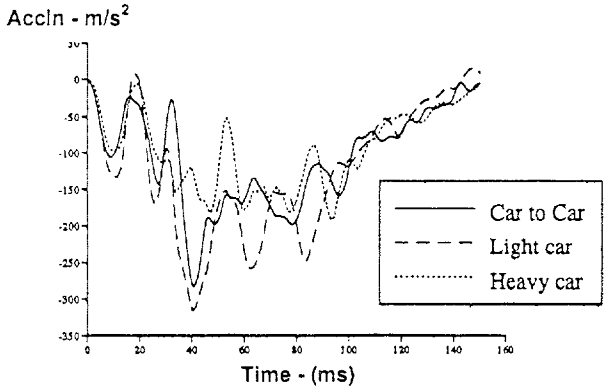

Figure 19. Comparison of deceleration pulse shapes for car to car and light to heavy car impacts.

In summary, in a car to car impact with cars of a similar stiffness, the occupant in the heavier car is subjected to a deceleration pulse that causes least injury in terms of ride down distance and shape. The converse is true for the occupant of the lighter car (Figure 20.).

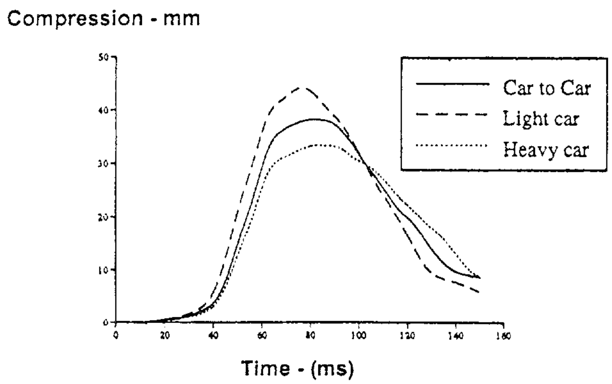

Figure 20. Comparison of chest compression for car to car and light to heavy car impacts.

Finally, we will consider the results obtained by applying the deceleration pulse from the full frontal rigid barrier test to the model (Figure 21.). The ride down distance for this deceleration pulse is the lowest (0.7 m) and hence considering this factor alone would be expected to give a high peak chest compression. In fact, the chest compression is not particularly high (Figure 22.). The low chest compression is a consequence of the near constant deceleration profile. This profile is most likely the result of manufacturers designing cars to meet and perform well in full frontal rigid wall legislative and consumer tests.

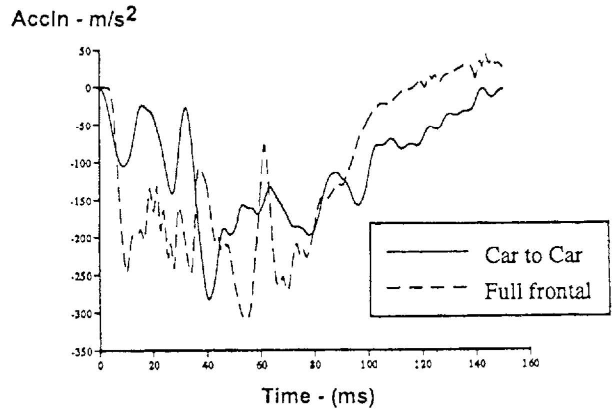

Figure 21. Comparison of deceleration pulse shapes for car to car and full frontal rigid wall impacts.

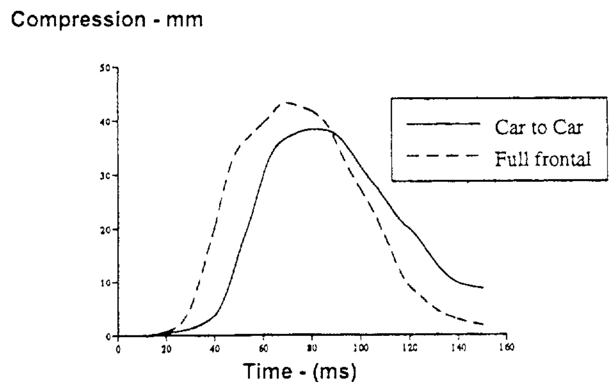

Figure 22. Comparison of chest compression for car to car and full frontal rigid wall impacts.

This study has indicated that in order to minimise chest injury the passenger compartment deceleration pulse should have a constant profile as opposed to a triangular back loaded profile, and ride down distance should be maximised. This has also been demonstrated with pulses from experimental car crashes.

The authors recognise that this study has used some theoretical deceleration pulses which may be impractical. However, a comprehensive range of pulses was investigated in order to quantify possible benefits before the study was restricted with practical considerations. In the future this study will be extended to help quantify the potential benefits of controlling the stiffness and deceleration pulse shape of vehicles in a 'compatible fleet'.

CAR TO CAR SIDE IMPACT

For side impact, parametric studies have been carried out using full car finite element models. The aim of this work was to aid our understanding of the effects of the bullet vehicle's mass, geometry and stiffness on the impacted car's structure and occupants response. This will help us to identify car structure characteristics which will improve compatibility. The modelling has been supported by full scale crash testing.

Finite Element Modelling

The purpose of this study was to understand the effect of changing bullet vehicle parameters on the impacted car's structure and dummy response. In order to undertake such a parametric study, the European Mobile Deformable Barrier (MDB) was chosen as the bullet vehicle. It was assumed that changing the MDB characteristics would indicate trends similar to those from changing the characteristics of an impacting car.

The FE model of the small four door car, EUROSID and MDB, used for the study is shown below (Figure 23.).

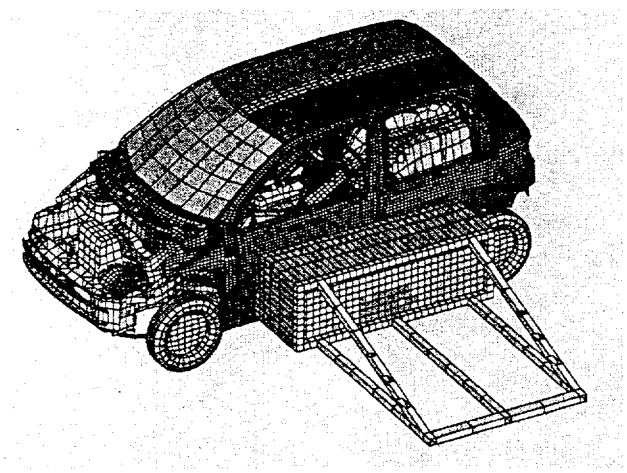

Figure 23. Small car FE model.

The model was validated for an European side impact test and shown to give reasonable agreement. The resulting vertical intrusion profile from this test is an indicator of a good structural response expected from a well designed modern car (13). In order to understand the results from the study it is important to understand the structural interaction between the barrier, the car and EUROSID. For this purpose the position of the barrier relative to EUROSID and the main side structure of the car is shown. It is seen that the bottom of the barrier just interacts with the sill and that the bottom stiffer half of the barrier is just in line with lower part of the EUROSID pelvis. There is limited contact with the 'A' and 'C' pillars (Figure 24.).

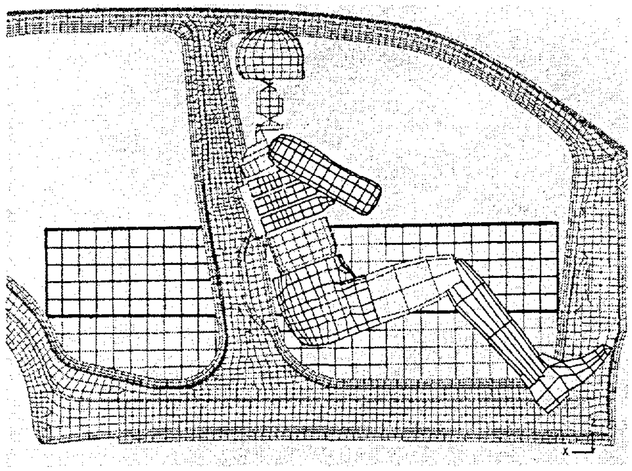

Figure 24. Relative positions of MDB, EUROSID and car structure in European side impact test.

A number of parameter sweeps were performed changing the following barrier characteristics:
1. Barrier centre impact point.
2. Barrier mass.
3. Barrier front face geometry.
4. Barrier stiffness.

Firstly, we will consider the effect of changing the point of impact of the barrier centre. For a standard European side impact test this is 550 mm above ground level in line with the R-point. This gives a barrier ground clearance of 300 mm (Table 4).

The form of these results can be explained in terms of the load paths into the car and its subsequent structural response. The two paths that we are considering are the load path through the door into the occupant and through the car's structure to its distributed mass. Ideally, we would like to reduce the load through the door into the occupant by putting more load directly into the car's structure. Whilst achieving this, the vertical intrusion profile should be maintained and any unnecessary delay in the occupant's acceleration should be avoided.

Table 4.
Injury Parameters with Varying Barrier Impact Point - European Side Impact Test as Reference.

Barrier centre impact point	Reference (Euro)	Lowered 100 mm	Raised 100 mm	Fore 200 mm	Aft 200 mm
Chest Compression (mm)					
Top rib	35	24	47	31	37
Middle rib	35	21	49	32	37
Bottom rib	32	20	44	30	32
Viscous Criteria (m/s)					
Top rib	0.58	0.22	0.74	0.50	0.59
Middle rib	0.63	0.26	0.93	0.55	0.63
Bottom rib	0.53	0.20	1.0	0.48	0.53
Abdomen Total					
Force (kN)	1.9	1.6	2.2	2.1	2.0
Pubic Symphysis					
Force (kN)	3.7	3.2	4.1	3.6	4.4

Lowering the barrier achieves this by giving better structural engagement with the sill. Raising the barrier has the opposite effect and changes the intrusion profile to put additional load on the chest. Moving the barrier fore and aft does not have such a large effect on the injury criteria as moving the barrier up and down. This is an expected result as the amount of structural engagement does not change greatly. Any additional interaction with the 'A' and 'C' pillars is with the weak edges of the barrier. These parameter sweeps indicate that more structural engagement with the sill can result in lower injury. However, a test conducted at TRL in which good structural engagement with the sill was achieved caused significant amounts of roll on the target car (Figure 25.). This may not be desirable as excessive roll may lead to head impacts on the cant rail.

Figure 25. Good Structural Engagement Inducing Significant Role in Target Vehicle.

Secondly, we will consider the effect of changing the barrier mass (Table 5.). The mass of the car modelled was about 800 kg. The effects on the dummy injury criteria are not as great as one might expect, considering the large changes in mass. The reason for this is that the most of the injury criteria peak before 40 ms whereas the momentum transfer is not complete until 80 - 100 msec. However, this effect is larger than that observed in a previous test conducted at TRL where no significant change in injury criteria was seen when changing the barrier mass from 950 kg to 1350 kg, with a car mass of 1080 kg (14). A possible explanation for this is that the stiffness of cars in side impact has increased, hence the momentum transfer is earlier, so mass can have a greater effect on injury criteria.

Table 5.
Injury Parameters with Varying Barrier Mass - European Side Impact Barrier as Reference.

Barrier mass	950 kg (Euro)	500 kg	1500 kg
Chest Compression (mm)			
Top rib	35	32	37
Middle rib	35	32	36
Bottom rib	32	27	33
Viscous Criteria (m/s)			
Top rib	0.58	0.51	0.79
Middle rib	0.63	0.50	0.73
Bottom rib	0.53	0.37	0.58
Abdomen Total			
Force (kN)	1.9	1.3	2.4
Pubic Symphysis			
Force (kN)	3.7	2.9	5.1

Thirdly, we will consider two simple geometry changes. The European side impact barrier has a bottom half which is 60 mm deeper than the top half (Figure 26.).

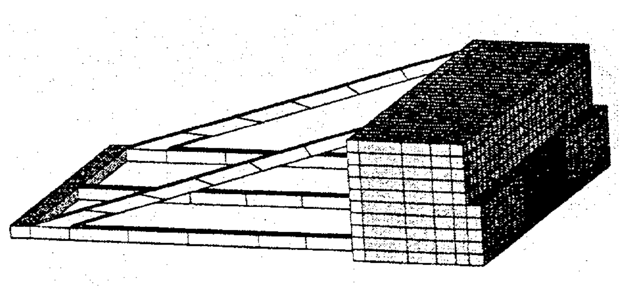

Figure 26. FE Model of European side impact barrier.

The first change was to extend the top half of the barrier 60 mm forward to give the barrier a planar front. The second change was to shorten the top half of the barrier by 60 mm so that the barrier bottom half was 120 mm deeper than the top half. The certification test results for the standard barrier model are near the centre of the specified corridor. Even with these changes in geometry, the barrier model still gives a total force deflection characteristic within the corridor. This shows that the effect of the geometry change on overall barrier stiffness was minimal. The effect of the geometry changes on the EUROSID injury criteria, were also small (Table 6.). Some slight improvement is seen from shortening the top half of the barrier. The most likely cause of this is the transfer of load from the load path into the occupant to the load path directly into the car.

Finally, we will consider the results of the stiffness parameter sweeps (Table 7). It is seen that stiffening the whole of the barrier increases all of the injury parameters, stiffening just the top of the barrier causes an even larger increase, but stiffening just the bottom of the barrier reduces the chest injury.

Table 6.
Injury Parameters with Varying Barrier Geometry-European Side Impact Barrier as Reference.

Barrier Geometry	Reference (Euro)	Planar Front	Bottom half 120mm deeper than top
Chest Compression (mm)			
Top rib	35	33	29
Middle rib	35	34	32
Bottom rib	32	31	30
Viscous Criteria (m/s)			
Top rib	0.58	0.57	0.52
Middle rib	0.63	0.65	0.56
Bottom rib	0.53	0.57	0.57
Abdomen Total			
Force (kN)	1.9	2.1	2.0
Pubic Symphysis			
Force (kN)	3.7	3.5	3.2

Table 7
Injury Parameters with Varying Barrier Stiffness.

Barrier Stiffness	Reference (EU)	Stiffness 2X	Stiffness 4X	Stiffness Top 2X	Stiffness Top 4X	Stiffness Bot 2X	Stiffness Bot 4X	Stiffness 2X	Stiffness Bot 2X
Geometry Changes:	None							Bottom half 120mm deeper than top	
Chest Compression (mm)									
Top rib	35	36	37	40	41	30	30	26	22
Middle rib	35	38	39	41	44	32	29	28	25
Bottom rib	32	36	37	36	42	30	27	28	26
Viscous Criteria (m/s)									
Top rib	0.58	0.54	0.51	0.60	0.64	0.56	0.44	0.25	0.23
Middle rib	0.63	0.65	0.64	0.60	0.68	0.64	0.54	0.40	0.38
Bottom rib	0.53	0.54	0.58	0.60	0.67	0.67	0.56	0.49	0.46
Abdomen Total									
Force (kN)	1.9	2.9	2.9	2.2	3.0	1.9	2.0	1.9	1.8
Pubic Symphysis									
Force (kN)	3.7	4.0	5.1	4.3	4.7	3.7	4.2	3.1	3.0
Barrier Peak Loads (kN)									
Barrier Bottom	85	101	115	79	71	106	120	111	111
Barrier Top	31	37	41	47	64	23	18	28	19
Barrier Total	112	128	149	124	133	126	136	127	119

If we combine this with a geometry change, i.e. shorten the top half of the barrier by 60 mm, a further reduction in the chest injury is achieved.

These results can be explained in terms of the load share between the load path into the occupant and the load path directly into the car. Stiffening the top half of the barrier increases the load through the door into the occupant and hence increases injury. In contrast, stiffening the bottom half decreases the load through the door into the occupant, by transferring load directly into the car, so reducing chest injury. For this particular car, as the middle of the barrier is approximately in line with the bottom of the pelvis, the ratio of the barrier top and bottom loads can be used as an indicator of the load share between the two major load paths (Table 7). It should be noted that in the case of stiffening the top of the barrier, there is an additional factor which could increase chest injury. This is the change in the intrusion profile from vertical to one that preferentially loads the chest.

In summary, the results of this modelling study indicate that in order to improve compatibility for side impact, the bullet vehicle should be designed such that it engages the structure of the target vehicle more effectively, through improved geometrical interaction. However, this should be achieved without compromising the intrusion profile or causing excessive roll in the target car. Stiffening of the bullet car's upper load path, without stiffening the lower path should be avoided as this will lead to increased occupant injury. It should be noted that these conclusions have yet to be validated by experimental test.

DISCUSSION

In both frontal and side impact, it has been shown that good geometrical interaction is fundamental to compatibility. In frontal impact the effect of geometry can mask the effect of mass and stiffness and dominate the outcome. In side impact, it is seen that better structural interaction can result in lower injury criteria by transferring loads from the occupant load path directly to one directly into the car's structure.

For frontal impact it is envisaged that in order to achieve compatibility between vehicles, the frontal structure will need to have a more uniform stiffness, with better structural tie-up. This may mean that the upper load path stiffness will increase. At a first glance this would appear to be in conflict with the requirements for side impact, but this does not have to be the case. In side impact the crush distance is small compared to frontal impact. Hence, in order to resolve the possible conflict, the upper load path of the car could have a stiffness profile that is soft for the initial crush and stiff for the remainder, satisfying both side and frontal requirements. This is not in conflict with current design, since the initial crush of cars is generally soft so that stiffer members are not deformed in low speed impacts, to meet insurance company repair ratings.

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